MULTI-OBJECTIVE OPTIMIZATION OF THE DESIGN PARAMETERS OF SHELL AND TUBE TYPE HEAT EXCHANGER BASED ON ECONOMIC AND SIZE CONSIDERATION

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

BACHELOR IN TECHNOLOGY
IN
MECHANICAL ENGINEERING

Department of Mechanical Engineering

National Institute of Technology, Rourkela

Rourkela 769008

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Mechanical engineering

By

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C E R T I F I C A T E

This is to certify that the work in this thesis entitled "Multi-Objective Optimization of the Design Parameters of a Shell and Tube Type Heat Exchanger Based on Economic and Size Consideration" by ***Sampreeti Jena***, has been carried out under my supervision in partial fulfillment of the requirements for the degree of *Bachelor of Technology in Mechanical Engineering* during session 2012-2013 in the *Department of Mechanical Engineering*, *National Institute of Technology*, *Rourkela*.

To the best of my knowledge, this work has not been submitted to any other University/Institute for the award of any degree or diploma.

Date: 08/05/2013

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CONTENTS

ABSTRACT

A heat exchanger is a device that is used to transfer heat between two or more fluids that are at different temperatures. These are essential elements in a wide range of systems, including the human body, automobiles, computers, power plants and comfort heating /cooling equipment. The most commonly used type is the shell and tube type heat exchanger.

Owing to their wide utilization, their cost minimization is an important target and instead of traditional iterative procedures, we implement a software-based (MATLAB), genetic algorithm in order to achieve the minimization of the total cost of equipment including capital investment and the sum of discounted annual energy expenditures including pumping. Simultaneously, the minimization of length of the heat exchanger is also targeted.

The multi-objective algorithm searches for the optimal values of design variables such as outer tube diameter, outer shell diameter and baffle spacing, for two types of tube layout arrangement (triangular and square) with the number of tube passes being two or four.

NOMENCLATURE

- C_{ps} → Shell side specific heat
- C_{pt} → Tube side specific heat
- $R_{foul,shell}$ → Shell side fouling resistance
- $R_{foul,tube}$ → Shell side fouling resistance
- ρ_t → Tube side Fluid Density
- ρ_s → Shell side Fluid Density
- μ_s → Viscosity at shell wall
- μ_t → Viscosity at tubewall
- T_{ci} → Cold fluid inlet temperature
- T_{co} → Cold fluid outlet temperature
- T_{hi} → Hot fluid inlet temperature
- T_{ho} → Hot fluid outlet temperature
- k_t → Tube side conductive heat transfer coefficient
- k_s → Tube side conductive heat transfer coefficient
- m_s → Shell side mass flow rate
- m_t → Tubeside mass flow rate
- b_0, k_1, n_1, p → Numerical constants
- d_i → Inner Tube diameter
- Pt → Tube Pitch
- ny → Equipment life
- i → Interest rate
- Ce → Energy Cost
- H → Annual operating time
- n → Number of tube passes
- Re_s → Shell side Reynolds Number
- Re_t → Tube side mass flow rate
- Pr_s → Shell side Prandtl Number
- Pr_t → Shell side Prandtl Number
- h_s → Shell side convective heat transfer coefficient

- h_t→ Tube side convective heat transfer coefficient
- U→ Overall heat transfer coefficient
- F→ Temperature Difference corrective factor
- P→ Pumping power
- Q→ Heat Duty
- L→ Length of heat exchanger
- S→ Surace area of heat exchanger
- ΔT_{ML}→ Logarithmic mean temperature difference
- ΔP_s→ Sheell side pressure drop
- ΔP_t→ Tube side pressure drop
- Cl→ Clearance
- C_{OD}→ Annual Operating Cost
- C_i→ Initial Investment
- C_{tot}→ Total annual cost
- N_t→ Number of tubes
- f_t→ Darcy tube side friction actor
- f_s→ hell side riction factor
- η→ Pump efficiency
- ν_s→ Shell side flow velocity
- ν_t→ Tube side flow velocity
- d_e→ Equivalent shell diameter

INTRODUCTION

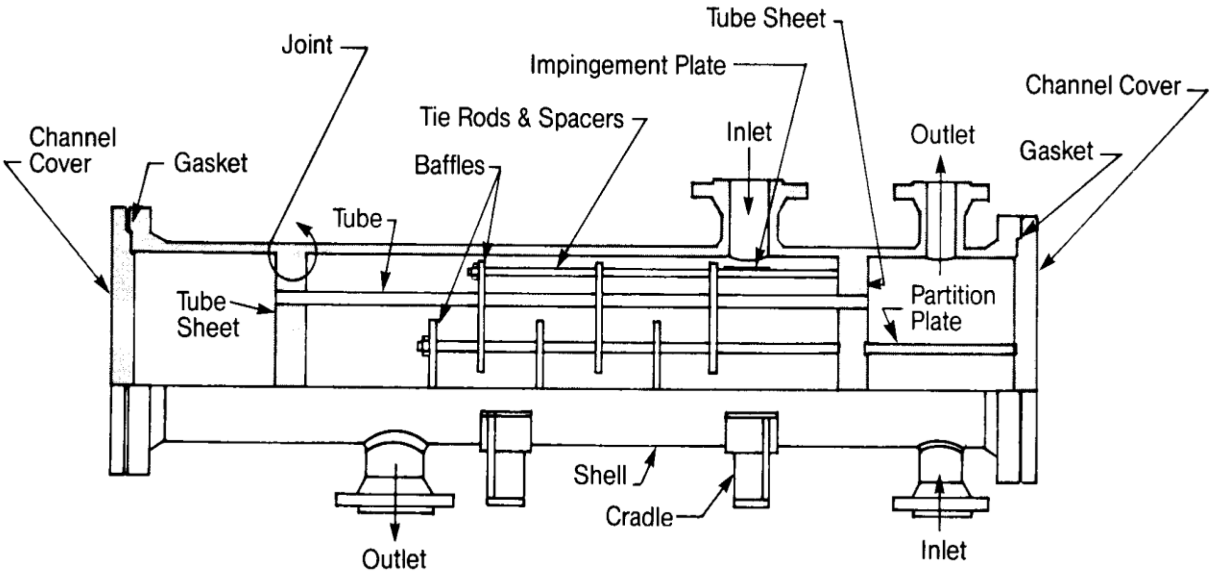
Shell-and-tube heat exchangers are possibly the most widely used type of heat exchangers owing to their flexibility of operating temperatures and pressures. They have much more favorable ratios of heat transfer surface to volume than double-pipe heat exchangers. They are also manufactured in diverse variety of sizes and flow configurations. They can operate at relatively higher pressures, and their structure enables disassembly for periodic maintenance and cleaning purposes. Shell-and-tube heat exchangers have extensive applications in refrigeration, power generation, heating and air conditioning, chemical processes, manufacturing, and medical.

A shell-and-tube heat exchanger is a reincarnation of the double-pipe configuration. Instead of a single pipe within a larger pipe, it comprises of a bunch of pipes or tubes encased within a cylindrical shell. One fluid flows through the tubes, and a second fluid flows within the space between the tubes and the shell. Baffles are installed along the tube bundle to direct the fluid between the tubes and shell, across the tubes. The turbulence induced by this flow configuration results in higher heat-transfer coefficients than flow parallel to the tubes. Heat is exchanged between the fluids through the tube walls. The fluids can be either liquids or gases on either the shell or the tube side. The use of many tubes increases the effective heat transfer area without resulting in an unrealistic heat exchanger length.

Heat exchangers with identical phases phase (liquid or gas) on each side can be called one-phase or single-phase heat exchangers. Two-phase heat exchangers can be used in vaporization of liquid into a gas(called boilers), or in condensing vapor into liquid (called condensers), with the phase change usually occurring on the shell side. Tube materials selected must be strong, thermally-conductive, corrosion-resistant, high quality with the ability to endure high thermal stresses and stresses due to fluid flow, typically metals, including copper alloy, stainless steel, carbon steel, non-ferrous copper alloy, Inconel, nickel, Hastelloy and titanium. Wrong choice of tube material could result in a leak through a tube between the shell and tube sides leading to fluid cross-contamination and possibly loss of pressure.

Pressure drop and heat transfer rates are inter-related entities and both influence the capital and operating costs of a heat exchanging system in a decisive way. The sum total of the pressure drops across the shell side and tube side determines the pumping power required and hence the annual operating cost. On the other hand the heat transfer rate determines the daily duration of operation for a given heat duty and hence equipment life, thereby, also controlling the annual operation cost. Hence it is necessary to suggest dimensions of apparatus that yield both optimal heat transfer and optimal pressure drop. Also the surface area needs to be in check to minimize the initial capital investment and the length cannot be allowed to exceed a certain practical limit.

Many variations are possible in the shell and tube design. Typically, the ends of each tube are connected to plenums (sometimes called water boxes) through holes in tubesheets. The tubes may be straight or bent in the shape of a U, called U-tubes. In nuclear power plants called pressurized water reactors, steam generators that are two-phase, shell-and-tube heat exchangers typically employ U-tubes. Most shell-and-tube heat exchangers are 1, 2, or 4 pass designs on the tube side which indicates the number of times the fluid in the tubes passes through the fluid in the shell. Surface condensers in power plants are often 1-pass straight-tube heat exchangers (see Surface condenser for diagram). Two and four pass designs are common as the fluid can enter and exit on the same side thereby, effectively simplifying the construction. Counter current heat exchangers allow the highest log mean temperature difference between the hot and cold streams. Many companies however do not use single pass heat exchangers because they are fragile, besides being costly to build. Often multiple heat exchangers can be used to simulate the counter current flow of a single large exchanger.



Shell & Tube Heat Exchanger

The design of shell-and-tube heat exchangers including thermodynamic and fluid dynamic design, strength calculations, cost estimation and optimization represents a complex process containing an integrated whole of design rules, calculating methods and empirical knowledge of various fields. At present various commercial programs such as HTRI, HTFS, THERM and CC-Therm are available. These tools allow designing and rating of tubular heat exchangers however, they do not consist of any optimization strategies that are needed from industries' point of view. The application of optimization methods, when designing heat exchangers, leads to the most cost-effective variant of apparatus. The design of heat exchangers requires knowledge of the allowable pressure drops of the streams that can be fully used. Information about allowable pressure drops are also required for input data of the above mentioned software packages. Setting the allowable pressure drop from experience or technical intuition can lead to a final solution far from the optimal design.

In the computer science field of artificial intelligence, a genetic algorithm (GA) is a search heuristic that imitates the process of natural evolution. This heuristic (also sometimes called a metaheuristic) is commonly used to generate solutions to optimize and search problems. Genetic algorithms belong to evolutionary algorithms (EA), which generate solutions to optimization problems using techniques inspired by natural evolution, such as inheritance, mutation, selection, and crossover. Genetic algorithms are useful in bioinformatics, phylogenetics, computational science, engineering, economics, chemistry, manufacturing, mathematics, physics, pharmacometrics and other fields.

In a genetic algorithm, a population of candidate solutions (called individuals, creatures, or phenotypes) to an optimization problem is improved toward better solutions. Each candidate solution has a set of properties (its chromosomes or genotype) which can be mutated and altered.

The evolution starts from a population of randomly generated individuals and is an iterative process, with a new population, in each iteration, called a generation. In each generation, the fitness of every individual in the population is computed; the fitness being the value of the objective function. The more fit individuals are stochastically elected from the current population, and each individual's genome is modified (recombined and possibly randomly mutated) to advance to a new generation. The new generation of candidate solutions is then utilized in the next iteration of the algorithm. Commonly, the algorithm terminates when either a maximum number of generations has been produced, or a satisfactory fitness level has been reached for the population.

A standard representation of each candidate solution is as an array of bits while arrays of other types and structures can be used to advance to the next generation. These genetic representations are convenient as their parts are easily juxtaposed due to their fixed size during crossover operations. Variable length representations make crossover implementation more complex. Tree-like representations are explored in genetic programming and graph-form representations are explored in evolutionary programming; a mix of both linear chromosomes and trees is explored in gene expression programming.

Problems which are characteristically viable for genetic algorithms include timetabling and scheduling problems. GAs has been applied to engineering to solve global optimization problems. As a general rule genetic algorithms might be useful in problem domains that have a complex fitness landscape as mixing, i.e., mutation in combination with crossover, to move the population away from local optima that a conventional hill climbing algorithm is prone to. Examples of problems solved by genetic algorithms include: mirrors designed to funnel sunlight to a solar collector, antennae designed to pick up radio signals in space, and walking methods for computer figures.

LITERATURE SURVEY

A Genetic Algorithm for Multi-objective Structural Optimization

Rodrigo E. Castro, Helio J. C. Barbosa

A genetic algorithm for multi-objective optimization has been implemented to evolve a uniformly distributed set of solutions in a Pareto set by: (i) ranking the population according to non-domination properties; (ii) defining a filter to retain Pareto set solutions and (iii) using suitable operators: exclusion, addition and single-objective operator which improves the individuals from the current filter to obtain a better Pareto set.

PROCEDURE:

PMOGA proposed here makes use of a filter to retain Pareto set. In order to improve the results of the MOOP, three new operators – the exclusion, the addition and the single-objective operator – are introduced here. A strategy of ranking based in non-domination properties is used to rank the solutions, after which, a tournament selection is used and the evolution of the population continues. The exclusion operator introduced derives the closest solution in the current filter to be removed for better solutions. This exclusion procedure is repeated till the current filter reaches the specified size. The metric used can be written as:

$$d_{j,k} = \sum_{i=1}^{N.fo} \frac{100.\left| fo_i(j) - fo_i(k) \right|}{0,5.[fo_i(j) + fo_i(k)]}$$

The addition operator selects the two more distant (in the objective function space) solutions and recombines those *n* times, where the value of *n* is usually between 2 and 6. This operator is to find solutions that complete the space between the two far away solutions. Finally, the single-objective operator finds the two extreme solutions, on the basis of each objective, to perform *n* recombinations between them. The purpose is to conduct a superior search for the optimum of each individual objective, thus obtaining a better distribution of the solutions.

To illustrate the use of the proposed PMOGA, two engineering design examples from the literature were given. The recombination operator adopted was the uniform crossover applied with a probability of 0.85. The mutation rate was fixed at 0.05 and the selection procedure used was a tournament based on the non-domination ranking. In both examples the PMOGA was run once, starting from a randomly generated initial population.

Example 1: Design of an I-beam

In this problem it is required to find the dimensions of the beam, that satisfy geometric and strength constraints and minimize the following conflicting objectives:

- cross section area of the beam;
- static deflection of the beam, under the vertical load P

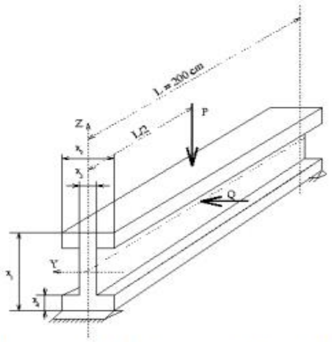


Figure 1: The simply supported I-beam [Coello and Christiansen, 1997].

The strength constraint is: $$\frac{M_y}{W_y} + \frac{M_z}{W_z} \le k_g$$

Where My and Mz are maximal bending moments in the Y and Z directions; Wy and Wz are section module in the Y and Z directions.

The section modules can be expressed as follows:

$$W_y = \frac{x_3(x_1 - 2x_4)^3 + 2x_2x_4\left[4x_4^2 + 3x_1(x_1 - 2x_4)\right]}{6x_1}$$

$$W_z = \frac{(x_1 - 2x_4)x_3^3 + 2x_4x_2^3}{6x_2}$$

Thus the strength constraint is:

$$16 - \frac{180000x_1}{x_3(x_1 - 2x_4)^3 + 2x_2x_4\left[4x_4^2 + 3x_1(x_1 - 2x_4)\right]} - \frac{15000x_2}{(x_1 - 2x_4)x_3^3 + 2x_4x_2^3} \ge 0$$

The objective functions are:

- Cross section area

$f_1(\mathbf{x}) = 2x_2x_4 + x_3(x_1 - 2x_4)\ \text{cm}^2$

- Static deflection

$$f_2(x) = \frac{PL^3}{48EI} cm$$

Where I is the moment of inertia which can be calculated from

$$I = \frac{x_3(x_1 - 2x_4)^3 + 2x_2x_4\left[4x_4^2 + 3x_1(x_1 - 2x_4)\right]}{12}$$

After substitutions, the second objective function is:

$$f_2(x) = \frac{60000}{x_3(x_1 - 2x_4)^3 + 2x_2x_4\left[4x_4^2 + 3x_1(x_1 - 2x_4)\right]} cm$$

Example 2: Welded Beam Design

The problem involves a beam subjected to a force F in its end and that needs to be welded to another structural component satisfying stability conditions and project limitations. The design variables - weld thickness (*h*); length of weld (*l*); width of the beam (*t*) and thickness of the beam (*b*) - are indicated in the Figure 3.

The two objectives for this problem are:

- the cost of the beam
- the deflection at the end of the beam

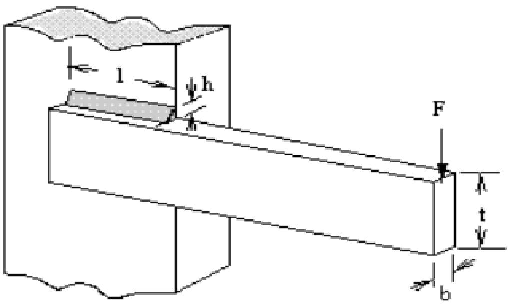


Figure 3: The welded beam [Deb, 1998].

There are five strength constraints. The two first constraints ensure that the shear stress and normal stress developed at the support location of the beam are lesser than the allowable shear strength (τ_{max}) and yield strength (σ_{max}) of the material respectively. The third constraint takes care that the permissible buckling load (along *t* direction) of the beam is greater than the applied load *F*. The fourth is a maximum limit (*umax*) for the displacement at the end of the beam. The fifth constraint checks that the thickness of the beam is not smaller than the weld thickness. There are other geometric constraints on the decision variables:

Min. $f1$ (**x**) = 1.10471$h2l$ + 0.04811tb (14+l)

Min. $f2$ (**x**) = 2.1952 / ($t3b$)

Subject to

$g_1(\mathbf{x}) \rightarrow \tau(\mathbf{x}) - \tau_{max} \leq 0$

$g_2(\mathbf{x}) \rightarrow \sigma(\mathbf{x}) - \sigma_{max} \leq 0$ (12)

$g_3(\mathbf{x}) \rightarrow F - P_c(\mathbf{x}) \leq 0$

$g_4(\mathbf{x}) \rightarrow 2.1952 / (t_3 b) - u_{max} \leq 0$

$g_5(\mathbf{x}) \rightarrow h - b \leq 0$

$0.125 \leq h, b \leq 5.0$

$0.1 \leq l, t \leq 10.0$

The stress and buckling terms are given as follows:

$$\tau(x) = \sqrt{\tau'^2 + \tau''^2 + l\tau'\tau'' / \sqrt{0.25\left[l^2 + (h+t)^2\right]}}$$

$$\tau' = \frac{6000}{\sqrt{2}hl}$$

$$\tau'' = \frac{6000(14 + 0.5l)\sqrt{0.25\left[l^2 + (h+t)^2\right]}}{2\left\{0.707hl\left[l^2/12 + 0.25(h+t)^2\right]\right\}}$$

$$\sigma(x) = \frac{504000}{t^2 b}$$

$$P_c(x) = 64746.022(1 - 0.0282346t)tb^3$$

Conclusion:-

The proposed algorithm could obtain Pareto-optimal solutions in a single run, even without the introduction of sensitive parameters for the resolution of the problem. This demonstrates that a good estimation of the Pareto set can be achieved in practice.

Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique

V.K. Patel, R.V. Rao

The concerned study peruses the use of a non-traditional optimization technique; called particle swarm optimization (PSO), for design optimization of shell-and-tube heat exchangers from economic perspective. Minimization of total annual cost is taken to be the objective function. Three design variables, shell internal diameter, outer tube diameter and baffle spacing are considered for optimization. Two tube layouts viz. triangle and square are also considered for optimization. Four different case studies are presented to establish the efficacy and precision of the proposed algorithm. The results of optimization using PSO technique are compared with those obtained by using genetic algorithm (GA).

Particle swarm optimization (PSO) is another evolutionary computation technique which exhibits common evolutionary computation features including initialization with a population of random candidates and searching for optima by upgrading generations. Potential solutions, called ‘birds’ or ‘particles’, are then “flown” through the problem space by following the current optimum particles. Each particle remembers its coordinates in the problem space, which are linked with the best solution (fitness) it has attained so far. This value is called ‘pBest’. Another “best” value that is explored by the global version of the particle swarm optimization is the overall best value and its location obtained by any particle in the population. This location is called ‘gBest’.

The particle swarm optimization concept consists of, at each iteration, diverting the velocity (i.e. accelerating) of each particle toward its ‘pBest’ and ‘gBest’ locations (global version of PSO).

Acceleration is weighted by a random term with separate random numbers being used for acceleration toward ‘pBest’ and ‘gBest’ locations. A new velocity is computed for each particle (potential solution) based on its previous velocity, the best location it has reached (‘pBest’) so far, and the global best location (‘gBest’), the population has achieved.

Particle's velocities on each dimension are limited by a maximum velocity parameter V_{max}, as specified by the user. Also if X_{i+1} exceeds maximum value for the corresponding design variable then it is set to the maximum value for that design variable and if X_{i+1} is less than the corresponding minimum design variable then it is brought to the minimum value for that design variable. The large inertia weights allow wide velocity updates enabling global exploration of the design space while small inertia weights concentrate the velocity updates to neighboring regions of the design space.

The effectiveness of the present approach using PSO is assessed by assessing four case studies. The first three case studies were studied by Caputo et al. using GA approach and taken from literature. The fourth case study was analyzed by Selbas et al. using GA approach.

Case 1: 4.34(MW) duty, methanol-brackish water exchanger

Results shows that a significant increase in the number of tubes reduces the tube side flow velocity which consecutively reduces the tube side heat transfer coefficient by 1.08%. The reduction in shell diameter increases the shell side flow velocity which consecutively increases the shell side heat transfer coefficient by 12.1%. The overall effect of these higher shell side heat transfer coefficient cause 8.2% increase in overall heat transfer coefficient, which leads to 7.45% decrease in heat exchanger area and 7.81% reduction in exchanger length, as compared to GA approach considered by Caputo et al. The capital investment also decreases corresponding to 5.7% because of reduction in heat exchanger area.

Case 2: 1.44(MW) duty, kerosene-crude oil exchanger.

It is seen that in this case higher tube side flow velocity increases the tube side heat transfer coefficient by 3.2%. Similarly, high shell side flow velocity increases the shell side heat transfer coefficient by 24.5%. An 8.85% increment in overall heat transfer coefficient is observed in the present case due to the combined increment in tube side and shell side heat transfer coefficient. As a result of high overall heat transfer coefficient, a decrease of 10.2% in heat exchanger area and reduction of 27.5% in heat exchanger length is observed compared to GA approach considered by Caputo et al. The capital investment is decreased by 5.1%. The higher tube side and shell side flow velocity increases the tube side and shell side pressure drop.

Case 3: 0.46(MW) duty, distilled water-raw water exchanger

In this case also a reduction in the heat exchanger area is observed (about 5.36%) because of higher overall heat transfer coefficient. The capital investment reduces by 2.86% compared to GA approach considered by Caputo et al. But, the increment in total pressure losses (about 14.4%) results in 1.5% increment in annual operating expense. Hence, a combined effect of reduction in capital investment and increment in operating expense led to a reduction of the total cost of about 2.51% compared to GA approach considered by Caputo et al.

Case 4: 2.09(MW) duty, water-water exchanger

Results show that higher tube side and shell side flow velocity increases the tube side and shell side heat transfer coefficient (20.2% on tube side and 39.6% on shell side) which in turn results in 11.24% increment in overall heat transfer coefficient in the current approach. The higher overall heat transfer coefficient results in 10.1% reduction in heat exchanger area and 17.5% reduction in heat exchanger length compared to GA approach considered by Selbas et al. The capital investment is decreased by 6.6%. The higher tube side and shell side flow velocity along with the smaller tube and shell size increases the tube side and shell side pressure drop (46.8% on tube side and 65% on shell side).

Heat exchanger design based on economic optimization:

Antonio C. Caputo, Pacifico M. Pelagagge, Paolo Salini

In this paper a process for optimal design of shell and tube heat exchangers is proposed, which implements a genetic algorithm to minimize the total cost of the equipment including capital investment and the sum of discounted annual energy expenditures related to pumping. In order to ascertain the accuracy of the proposed plan, three case studies are also presented.

PROCEDURE:-

The procedure for optimal heat exchanger design includes the following steps:

• approximation of the exchanger heat transfer area based on the required duty and other design specifications by assuming a random set of design variables values;

• computation of the capital investment, operating cost, and the objective function;

• utilization of the optimization algorithm to select a new set of values for the design variables;

• repetition of the previous steps till a minimum of the objective function is found.

Design specification indicate the heat duty of the exchanger, and are specified by imposing five of the following six parameters: the mass flow rates of the two fluids, as well as the inlet and outlet temperatures of the fluids shellside T_{is}, T_{os}, and tubeside, T_{it}, T_{ot}, the rest being determined by energy balance. Fixed parameters provided by the user are the tubesheet patterns (triangular or square) and pitch, the number of tube-side passages (1, 2, 4 . . .), the fouling resistances $R_{foul,shell}$ and $R_{foul,tube}$, and the thermo-physical properties of both fluids. The optimization variables, with values updated iteratively by the algorithm, are the shell inside diameter Ds, tube outside diameter do, and baffles spacing B.

Based on the actual values of the design specifications, the fixed parameters, and the present values of the optimization variables, the exchanger design routine evaluates the values of the shell-side and tube-side heat exchange coefficients h_s, h_t, the overall heat exchange area S, the number of tubes N_t, the shell, tube length L and the tube-side and shell-side flow velocities υ_s and υ_t, thus outlining all necessary details of the exchanger satisfying the assigned thermal duty specifications. The computed values of flow velocities and the constructive details of the exchanger structure are then used to evaluate the best value of the objective function.

The low chart depicts the sequence of procedures in the implementation of the algorithm.

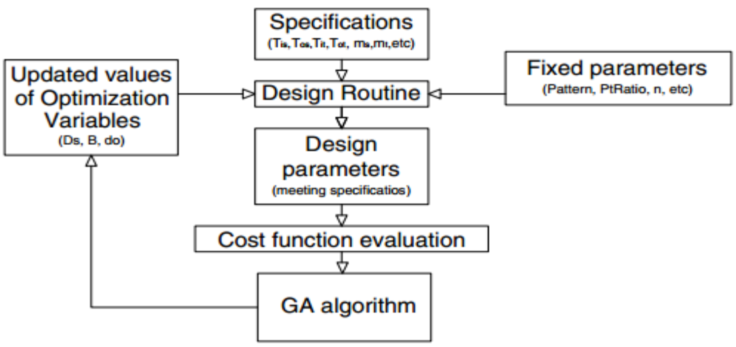


Fig. 1. Proposed optimisation algorithm.

RESULTS:-

The following three different test cases, representative of a wide range of possible applications, were considered:

Case 1: methanol–brackish water exchanger:

A slight reduction of heat exchange area resulted owing to a reduction of the exchanger length, even if the number of tubes increased considerably and the tubes diameter was also decreased. The capital investment decreased analogously (_4.4%). The higher number of tubes and the shorter shell were instrumental in reducing both the shellside and tubeside flow velocity leading to a marked decrease of pressure losses. So, the annual pumping cost decreased markedly (55.14%). Overall, the combined reduction of capital investment and operating costs led to a reduction of the total cost of about 14.5%.

Case 2: kerosene–crude oil exchanger:

In this case a medium decrease of the heat exchange area was noted (about 13%). This was the combined result of a rise in shell diameter coupled with a marked increase in the number of tubes, combined with a significant decrease of both diameter and length of the tubes. The capital investment, hence decreased by 7.4%. Also the marked reduction of flow velocities enabled a saving of 66.2% in the annual operating expenses, leading to a net reduction of the total cost which decreased by about 24.8% compared to the original solution.

Case 3: distilled water–raw water exchanger:

In this case, a significant increase of about 34% of the heat exchange area was observed with a corresponding increase of capital investment (+15.8%). This was due to a significant increase of shell diameter as well as the number of tubes, not neutralized by the strong reduction of the length of the tubes and by a slight decrease in the diameter of the tubes. Conversely, a very high reduction of flow velocities and pressure drops allowed to drastically cancel about 93.9% of annual operating costs.

METHODOLOGY USED

The optimization procedure was implemented by a multi-objective genetic algorithm (GA).Starting from an initial population of randomly created individuals representing candidate solutions, in this scenario, a heat exchanger of specific configuration and conforming with the design specifications, the GA uses the concept of survival of the fittest to produce more desirable individuals in subsequent evolutionary generations of the population. The cost value of each candidate solution denotes the fitness function of the individual which is a measure of its quality relative to the entire population.

According to GA theory, each new generation is composed of:-

1. The best individual(s) advanced from the previous generation (the so called Elite Count).

2. New child individuals obtained by crossover recombination of the genes from a pair of selected parents of the current generation.
3. Mutant individuals.
4. Migrant individuals from past generations.

The proposed design optimization procedure was implemented on a personal computer resorting to the Genetic Algorithm toolbox of the scientific computing environment MATLAB. Following an experimental trials, the following setting parameters for the GA were chosen. Each generation was made of 15 individuals as the added computational time required to analyze larger populations was not offset by the smaller number of generations required to attain convergence considering that always the
same best individual was obtained.

The maximum number of generations was set at 1000. However, in the tests convergence was always obtained within about 500 generations. The number of best performing individuals of a generation which are transferred to the next one (Elite count) was set at 2.

The adopted ‘‘Crossover Fraction’’ parameter, i.e. the percentage of individuals of each generation, excluding the Elite Count individuals, which are generated through a crossover recombination of selected individuals of the previous generation was 0.5. The selection algorithm used for picking the parent individuals was the selection-roulette in which parents are picked with a probability proportional to their fitness function. The crossover method utilized is the so-called crossover-scattered, where a random binary vector is created having a number of bits equal to the number of genes of an individual. Then, the genes where the value is 1 are copied from the first parent, while the genes where the value is 0 are copied from the second parent. The obtained genes are then combined to form the child.

The percentage of individuals who undergo a mutation derives instead from the previously defined parameters in that all individuals who are not bred and are not part of the elite count are subject to mutation. Finally, migration of individuals from previous generations is allowed each third generation. The number of transferred individuals is (PopulationSize _ EliteCount) * (MigrationFraction) where the migration fraction was set at 0.5. Those migrant individuals substitute the worst individuals of the current generation.

GOVERNING EQUATIONS

Mean logarithm temperature difference:

$$S = \frac{Q}{U \Delta T_{\mathrm{ML}} F}, \qquad (1)$$

F being the temperature difference corrective.

The heat transfer coefficient is computed through the following equations:

$$U = \frac{1}{\frac{1}{h_s} + R_{\mathrm{foul,shell}} + \frac{d_o}{d_i} \cdot \left(R_{\mathrm{foul,tube}} + \frac{1}{h_t}\right)}$$

$$d_i = 0.8 d_o.$$

The tube-side heat transfer coefficient h_t is computed, according to the flow regime, resorting to the following correlations:

$$h_t = 0.027 \frac{k_t}{d_0} \mathrm{Re}_t^{0.8} \mathrm{Pr}_t^{1/3} \left(\frac{\mu_t}{\mu_{wt}}\right)^{0.14} \qquad (5)$$

(for $\mathrm{Re}_t > 10{,}000$ [9])

where, f_t is the Darcy friction factor [5] given as,

$$f_t = \left(1.82 \log 10^{\mathrm{Re}_t} - 1.64\right)^{-2} \qquad (6)$$

Re_t is the tube side Reynolds number and given by,

$$\mathrm{Re}_t = \frac{\rho_t v_t d_i}{\mu_t} \qquad (7)$$

Flow velocity for tube side is found by,

$$v_t = \frac{m_t}{(\pi/4) d_t^2 \rho_t} \left(\frac{n}{N_t}\right) \qquad (8)$$

N_t is the number of tubes and n is the number of tube passes which can be found approximately from the following equation [4,6,30,31],

$$N_t = C \left(\frac{D_s}{d_0}\right)^{n_1} \qquad (9)$$

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.

Pr_t is the tube side Prandtl number and given by,

$$\mathrm{Pr}_t = \frac{\mu_t C_{pt}}{k_t} \qquad (10)$$

also, $d_i = 0.8 d_o$

Kern's formulation for segmental baffle shell-and-tube exchanger is used for computing shell side heat transfer coefficient h_s [3],

$$h_s = 0.36\frac{k_t}{d_e}\mathrm{Re}_s^{0.55}\mathrm{Pr}_s^{1/3}\left(\frac{\mu_s}{\mu_{wts}}\right)^{0.14} \quad (11)$$

where, d_e is the shell hydraulic diameter and computed as [3,30],

$$d_e = \frac{4\left(S_t^2 - \left(\pi d_o^2/4\right)\right)}{\pi d_o} \quad (12)$$

(for square pitch)

$$d_e = \frac{4\left(0.43S_t^2 - \left(0.5\pi d_o^2/4\right)\right)}{0.5\pi d_o} \quad (13)$$

(for triangular pitch)

Cross section area normal to flow direction is determined by [2],

$$A_s = D_s B\left(1 - \frac{d_o}{S_t}\right) \quad (14)$$

$$v_s = \frac{m_s}{\rho_s A_s} \quad (15)$$

Reynolds number for shell side follows,

$$\mathrm{Re}_s = \frac{m_s d_e}{A_s \mu_s} \quad (16)$$

Prandtl number for shell side follows,

$$\mathrm{Pr}_s = \frac{\mu_s C_{ps}}{k_s} \quad (17)$$

The overall heat transfer coefficient (U) depends on both the tube side and shell side heat transfer coefficients and fouling resistances are given by [2],

$$U = \frac{1}{(1/h_s) + R_{fs} + (d_o/d_i)\left(R_{ft} + (1/h_t)\right)} \quad (18)$$

Considering the cross flow between adjacent baffle, the logarithmic mean temperature difference (LMTD) is determined by,

$$\mathrm{LMTD} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln((T_{hi} - T_{co})/(T_{ho} - T_{ci}))} \quad (19)$$

The correction factor F for the flow configuration involved is found as a function of dimensionless temperature ratio for most flow configuration of interest [32,33].

$$F = \sqrt{\frac{R^2+1}{R-1}}\frac{\ln((1-P)/(1-PR))}{\ln\left(\left(2-PR+1-\sqrt{R^2+1}\right)/\left(2-PR+1+\sqrt{R^2+1}\right)\right)} \quad (20)$$

where R is the correction coefficient given by,

$$R = \frac{(T_{hi} - T_{ho})}{(T_{co} - T_{ci})} \tag{21}$$

P is the efficiency given by,

$$P = \frac{(T_{co} - T_{ci})}{(T_{hi} - T_{ci})} \tag{22}$$

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

$$A = \frac{Q}{UFLMTD} \tag{23}$$

For sensible heat transfer, the heat transfer rate is given by,

$$Q = m_h C_{ph}(T_{hi} - T_{ho}) = m_c C_{pc}(T_{co} - T_{ci}) \tag{24}$$

Based on total heat exchanger surface area (A), the necessary tube length (L) is,

$$L = \frac{A}{\pi d_o N_t} \tag{25}$$

$$\Delta P_t = \Delta P_{\text{tube length}} + \Delta P_{\text{tube elbow}} \tag{26}$$

$$\Delta P_t = \frac{\rho_t v_t^2}{2}\left(\frac{L}{d_i} f_t + p\right) n \tag{27}$$

Different values of constant p are considered by different authors. Kern [3] assumed $p = 4$, while Sinnot et al. [30] assumed $p = 2.5$.

The shell side pressure drop is,

$$\Delta P_s = f_s \left(\frac{\rho_s v_s^2}{2}\right)\left(\frac{L}{B}\right)\left(\frac{D_s}{d_e}\right) \tag{28}$$

where,

$$f_s = 2 b_o \mathrm{Re}_s^{-0.15} \tag{29}$$

and $b_o = 0.72$ [34] valid for $\mathrm{Re}_s < 40{,}000$.

Considering pumping efficiency (η), pumping power computed by,

$$P = \frac{1}{\eta}\left(\frac{m_t}{\sigma_t}\Delta P_t + \frac{m_s}{\sigma_s}\Delta P_s\right) \tag{30}$$

The objective function has been assumed as the total present cost Ctot

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

The capital investment C_i is computed as a function of the exchanger surface adopting Hall's correlation

$C_i = a_1 + a_2 S_{a3}$

Where a_1 = 8000, a_2 = 259.2 and a_3 = 0.91 for exchangers made with stainless steel for both shells and tubes

The total discounted operating cost related to pumping power to overcome friction losses is instead computed from the following equations:

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

$$C_o = P \cdot C_E \cdot H,$$

$$P = \frac{1}{\eta}\left(\frac{m_t}{\rho_t} \cdot \Delta P_t + \frac{m_s}{\rho_s} \cdot \Delta P_s\right)$$

PROCEDURE

- A total of 4 design variations are considered for the optimization.
- Case 1: n= 2, Square tube layout
- Case 2: n = 2, Triangular tube layout
- Case 3: n = 4, Square tube layout
- Case 4: n = 4, Triangular tube layout

- The three design variables are
  - Tube outside diameter (d_0)
  - Shell inside diameter (D_s)
  - Baffle spacing (B)

- 2 objective functions are simultaneously optimized to obtain a set of solutions that yield the best values for both functions.
  - The cost which consists of the initial investment and the annual cost of operation
  - Length of the heat exchanger

- The optimization toolbox in MATLAB is used for implementation of multi-objective optimization using Genetic Algorithm. The solver used is “gamultobj” and the settings are fixed as following:-
  - Population Type: Double Vector
  - Creation Function: Constraint Dependent
  - Population Size: Default (15*No of variables)
  - Initial Population: Default
  - Initial Scores: Default

- Selection Function: Tournament
- Tournament size: Default (2)
- Crossover Fraction: Default (0.8)
- Mutation Function: Constraint Dependent
- Crossover Function: Intermediate
- Crossover Ratio: Default (1)
- Migration Direction: Forward
- Migration Fraction: Default (0.2)
- Migration Interval: Default (20)
- Distance Measure Function: Default @ distancecrowding
- Pareto Front Population Fraction: Default (0.35)
- Hybrid Function: fgoalattain
- Maximum Generations: 200*No of Variables
- Time Limit: Default (Infinite)
- Fitness Limit: Default (Infinite)
- Stall Generations: Default (100)
- Function Tolerance: 1e-4

➢ The Pareto Front and Average Pareto Spread with an interval of 1 generation.

EMPIRICAL AND STATISTICAL DATA

The shell side fluid and tube side fluids are distilled water and raw water respectively.

- $C_{ph} = C_{pt} = C_{ps}$ = 4.18 KJ/Kg K
- $R_{foul,shell} = R_{foul,tube}$ = 0.00017m^{2}K/W
- ρ_t = 999 kg/m^3
- ρ_s = 995 k/m^3
- μ_s = 0.008 pa s
- μ_t = 0.00092 pa s
- T_{ci} = 23.9^{0}C
- T_{co} = 26.7^{0}C
- T_{hi} = 33.9^{0}C
- T_{ho} = 29.4^{0}C
- K_t = 0.62 W/mK
- K_s = 0.62 W/mK
- $m_h = m_s$ = 22.07 kg/s

- m_t = 35.31 kg/s
- b_0 = 0.72
- d_i = 0.8 d_0
- Pt = 1.25 d_0
- ny = 10 years
- i = 10%
- Ce = 0.12 units/KWh
- H = 7000 yr/hr

Assumptions:

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

Table 1 TRIANGULAR TUBE PITCH

No. of passes	K_1	n_1
2	0.249	2.207
4	0.175	2.285

Table 2 SQUARE TUBE PITCH

No of passes	K_1	n_1
2	0.156	2.291
4	0.158	2.263

Variable Bounds:

- $0.015 \leq d0 \leq 0.051$
- $0.05 \leq B \leq 0.5$
- $\leq Ds \leq 1.5$

RESULTS

CASE 1:

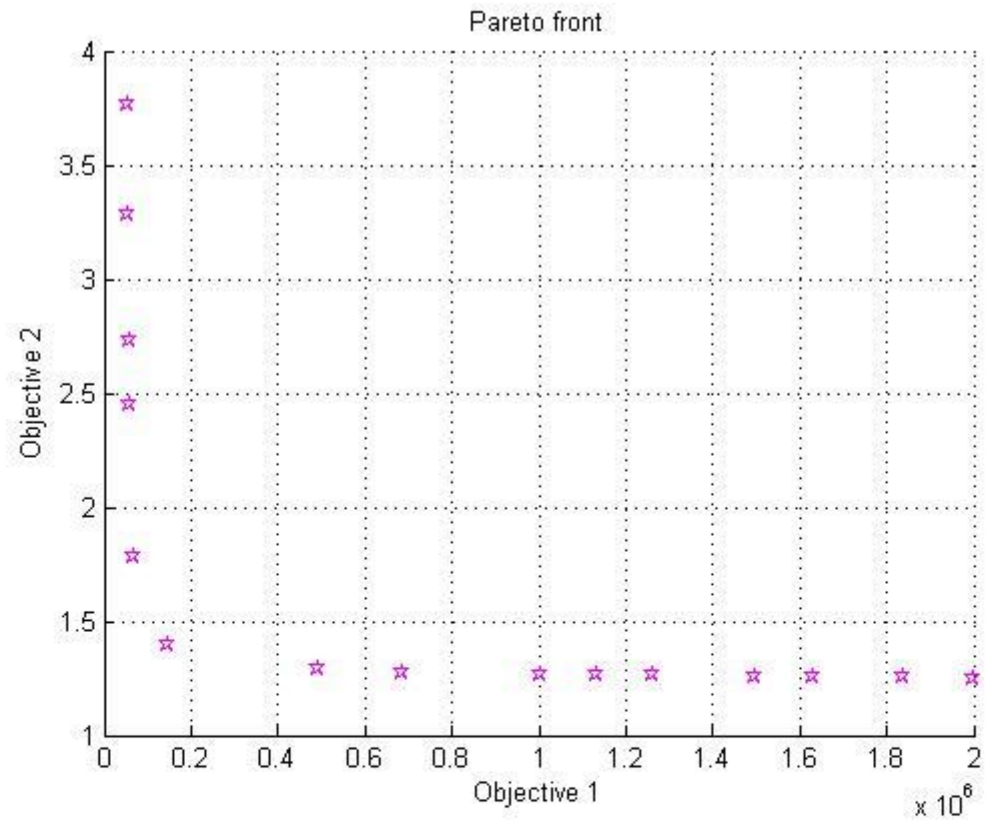
- Final Form of The Objective Function:

f(1)=
((162823.372*(x(1)^1.323)*(x(2)^0.55))/(x(3)^6.323)+(16756.278*(x(1)^0.873))/(x(3)^6.873)+(21509.566*(x(1)^0.8492))/(x(3)^5.0402))*((8.36+log10((2.36825*(x(1)^2.35))/(x(3)^4.17)))^(-2))+
(323.79*(x(1)^0.582))/(x(3)^4.582)+(3991.27*(x(1)^0.5917))/((x(2)^2.3)*(x(3)^2.5917))+(410.744*(x(1)^0.1417))/((x(2)^2.85)*(x(3)^3.1417))+(527.268*(x(1)^0.1179))/((x(2)^2.85)*(x(3)^0.2254))+8000+259*((788*(x(1)^0.45)*(x(2)^0.55)*(x(3)^0.55)+81.12+(104.13*(x(3)^1.8328))/(x(1)^0.0238))^0.91);

f(2)=
(1609.29*(x(1)^1.741)*(x(2)^0.55))/(x(3)^1.741)+(165.605*(x(1)^1.291))/(x(3)^2.291)+(212.58*(x(1)^1.2672))/(x(3)^0.4582);

Table 1 Pareto solution sets for Case 1

Pareto Solution Set	f(1)	f(2)	X(1)	X(2)	X(3)
1	1995992	1.253852	0.015005	0.050006	1.5
2	53242.4	3.767434	0.015005	0.499994	0.7557
3	1626856	1.258168	0.015005	0.053912	1.5
4	682542.7	1.280339	0.015006	0.074948	1.498931
5	1497419	1.260781	0.015005	0.055601	1.49912
6	1838280	1.256146	0.015008	0.051543	1.499676
7	53242.4	3.767434	0.015005	0.499994	0.7557
8	1003543	1.270012	0.015007	0.064594	1.499441
9	143647.8	1.400892	0.01502	0.157123	1.44308
10	66804.41	1.787831	0.015005	0.438604	1.280073
11	489623.6	1.292702	0.015008	0.085657	1.496248
12	59118.51	2.449016	0.015007	0.423265	0.99767
13	55659.61	2.729828	0.015009	0.495414	0.939288
14	1132994	1.266791	0.015005	0.061698	1.499506
15	1261123	1.264304	0.015006	0.059266	1.499671
16	54120.71	3.281879	0.015007	0.485797	0.825147



Pareto front (case I)

Case 2:

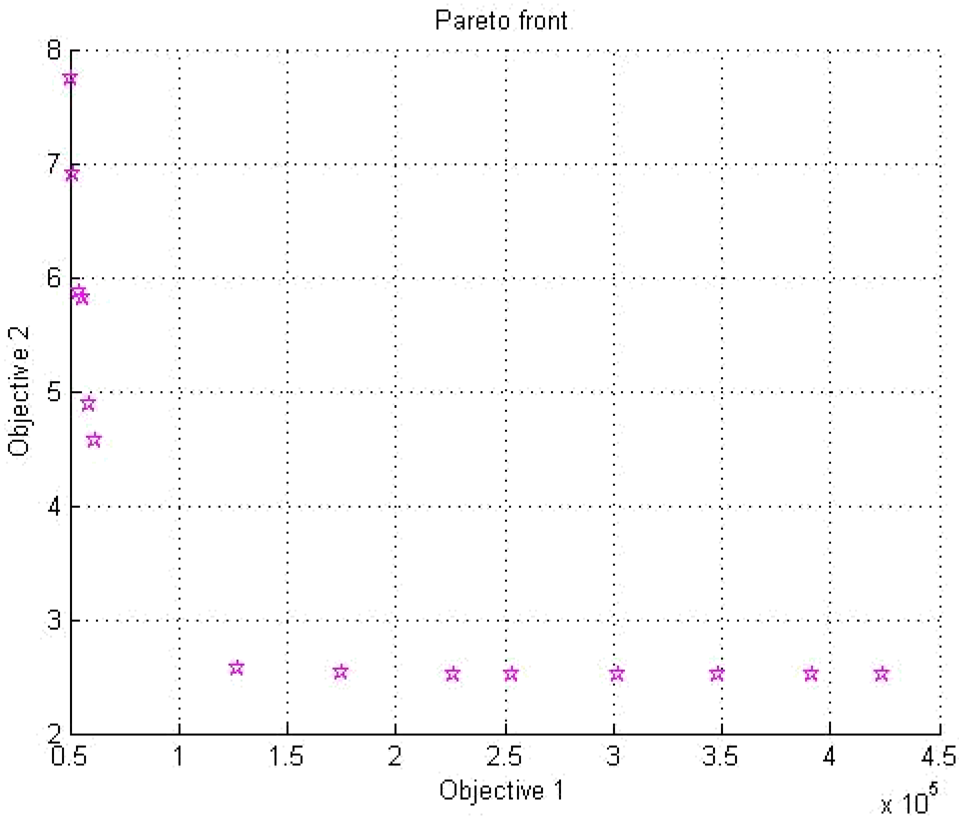
- Final form of the objective function

f(1)=
((138.75*(x(1)^1.071)*(x(2)^0.55))/(x(3)^6.071)+(25.236*(x(1)^0.621))/(x(3)^6.621)+(105.233*(x(1)^0.6554))/(x(3)^4.8554))*(((0.1424*(log10((314167.22*(x(1)^1.207))/(x(3)^2.207))))-0.23355)^(-2)) + (127.127*(x(1)^0.414)) /(x(3)^4.414) +(247.5*(x(1)^0.507)) /((x(2)^2.3)*(x(3)^2.507))+(45.017*(x(1)^0.057))/((x(2)^2.85)*(x(3)^3.057))+(187.71*(x(1)^0.0914))/((x(2)^2.85)*(x(3)^1.2914))+8000+259*((446*(x(1)^0.45)*(x(2)^0.55)*(x(3)^0.55)+81.12+338.256*(x(1)^0.0344)*(x(3)^1.7656))^0.91);

f(2)=
(570.434*(x(1)^1.657)*(x(2)^0.55))/(x(3)^1.657)+(103.752*(x(1)^1.207))/(x(3)^2.207)+(432.63*(x(1)^1.2414))/(x(3)^0.4414);

Table 2 Pareto solution sets for Case 2

Pareto Solution Set	F(1)	F(2)	X(1)	X(2)	X(3)
1	50203.7	7.750892	0.01661	0.487396	0.530987
2	174555.3	2.535273	0.016133	0.099044	1.498579
3	253185.9	2.522927	0.016133	0.074852	1.498398
4	302366.2	2.518645	0.016133	0.067478	1.498587
5	55813.66	5.817907	0.016574	0.373452	0.65661
6	58633.08	4.88362	0.016274	0.411127	0.756528
7	423595.5	2.512074	0.016132	0.056833	1.498633
8	226041.5	2.525847	0.016133	0.080585	1.498575
9	391206.8	2.513735	0.016134	0.059053	1.49857
10	126586.7	2.562349	0.016133	0.161544	1.498579
11	51471.08	6.901998	0.016423	0.444906	0.571023
12	50203.7	7.750892	0.01661	0.487396	0.530987
13	61680.07	4.57523	0.016362	0.39078	0.80761
14	54451	5.87219	0.016218	0.39287	0.638104
15	347773.9	2.515787	0.016133	0.062638	1.498588
16	423595.5	2.512074	0.016132	0.056833	1.498633



Pareto front (case 2)

Case 3:

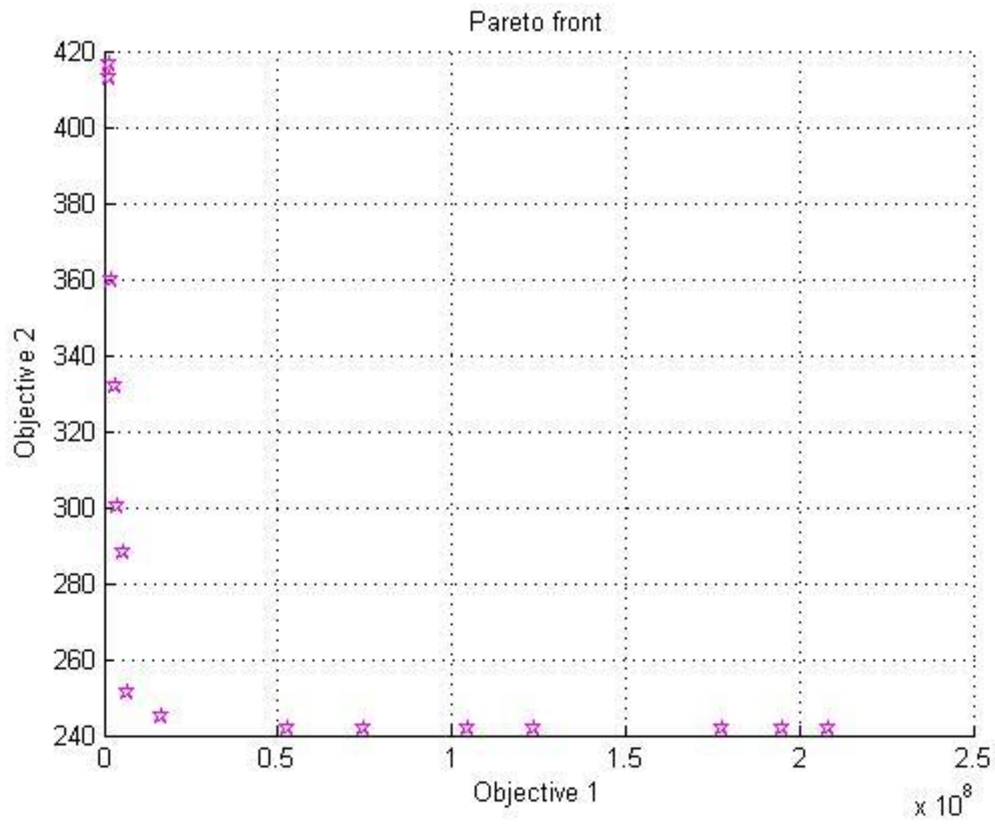
- Final form of the objective function

```
f(1)=
((1254079.3*(x(1)^1.239)*(x(2)^0.55))/(x(3)^6.239)+(129066.24*(x(1)^0.789))/(x(3)^6.789)+661639.48/((x(1)^0.8492)*(x(3)^5.0402)))*((8.36+log10((8.17*(x(1)^2.29866))/(x(3)^4.1186)))^(-2)) + (2525.93*(x(1)^2.789)) /(x(3)^6.789) + (3312.4057*(x(1)^0.563)) / ((x(2)^2.3)*(x(3)^2.563))+(340.9035*(x(1)^0.113))/((x(2)^2.85)*(x(3)^3.113))+1747.59/((x(1)^0.8974)*(x(2)^2.85)*(x(3)^1.3026))+8000+259*((788.207*(x(1)^0.45)*(x(2)^0.55)*(x(3)^0.55)+81.12+(415.85*(x(3)^1.8104))/(x(1)^1.0104))^0.91);
```

```
f(2)=
(1588.74*(x(1)^1.713)*(x(2)^0.55))/(x(3)^1.713)+(163.5087*(x(1)^1.263))/(x(3)^2.263)+(838.204*(x(1)^0.2526))/(x(3)^0.4526);
```

Table 3 Pareto solution sets for Case 3

Pareto Solution Set	F(1)	F(2)	X(1)	X(2)	X(3)
1	1382230	416.4354	0.051	0.5	0.948281
2	2.08E+08	241.9545	0.015001	0.052165	1.499999
3	7.44E+07	241.9939	0.015004	0.076317	1.49999
4	1.63E+07	244.8743	0.015663	0.145053	1.498017
5	1382230	416.4354	0.051	0.5	0.948281
6	1.24E+08	241.9692	0.015001	0.063086	1.499992
7	3146567	331.8706	0.050071	0.217882	1.491477
8	5393604	288.0674	0.02912	0.208352	1.489072
9	1.78E+08	241.959	0.015001	0.055233	1.499996
10	1.95E+08	241.9562	0.015001	0.053439	1.499999
11	1.04E+08	241.9732	0.015001	0.067203	1.499996
12	3975894	300.0257	0.020535	0.363654	1.122275
13	6431416	251.141	0.01609	0.319452	1.440522
14	5.24E+07	242.0047	0.015003	0.087425	1.499964
15	1755124	359.6938	0.049805	0.442119	1.262807
16	1383275	413.1487	0.051	0.499999	0.963652



Pareto front (case 3)

Case 4:

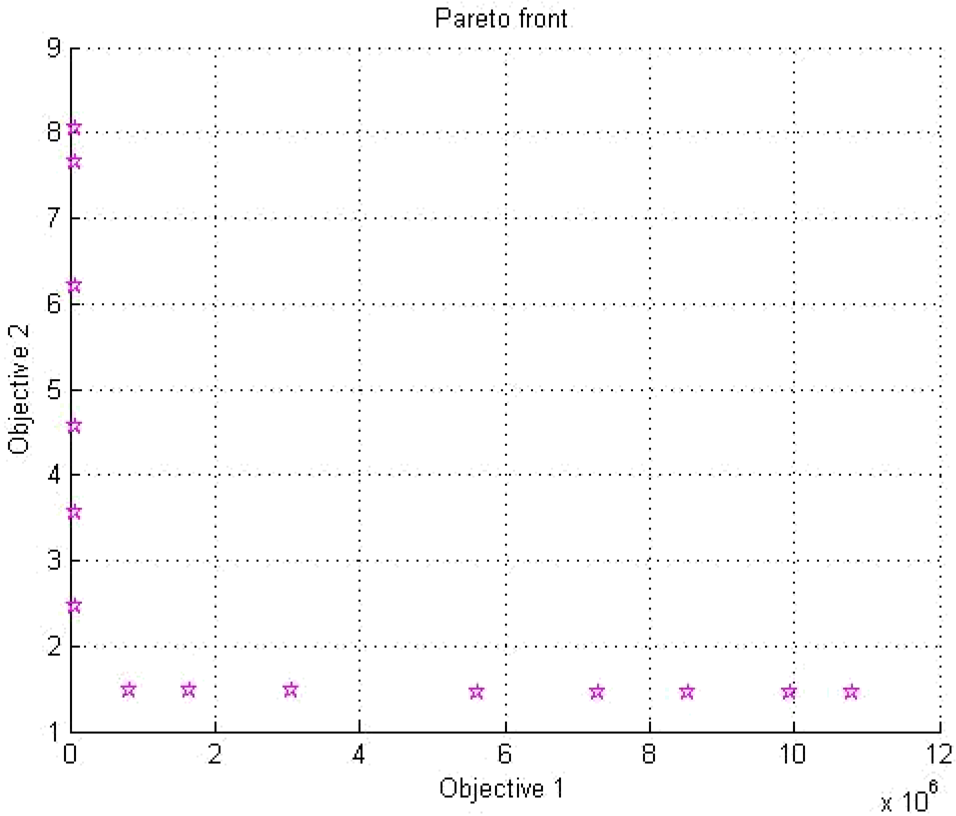
- Final form of the objective function

f(1)=
((811.63*(x(1)^1.305)*(x(2)^0.55))/(x(3)^6.305)+(147.62*(x(1)^0.855))/(x(3)^6.855)+(266.64*(x(1)^0.827))/(x(3)^5.027))*(((0.07178*(log10((57687.5*(x(1)^1.285))/(x(3)^2.285))))-3.725)^(-2))+ (2058.76*(x(1)^0.57)) /(x(3)^4.57) +(352.156*(x(1)^0.585)) / ((x(2)^2.3)*(x(3)^2.585))+(64.05*(x(1)^0.135))/((x(2)^2.85)*(x(3)^3.135))+115.69/((x(1)^0.893)*(x(2)^2.85)*(x(3)^1.3))+8000+259*((446*(x(1)^0.45)*(x(2)^0.55)*(x(3)^0.55)+81.12+(146.52*(x(3)^1.828))/(x(3)^1.7656))^0.91);

f(2)=
(811.63*(x(1)^1.735)*(x(2)^0.55))/(x(3)^1.735)+(147.62*(x(1)^1.285))/(x(3)^2.285)+(266.64*(x(1)^1.257))/(x(3)^0.457);

Table 4 Pareto solution set For Case 4

Pareto Solution Set	F(1)	F(2)	X(1)	X(2)	X(3)
1	66155.38	7.775957	0.049461	0.49999	1.499961
2	66155.38	7.775957	0.049461	0.49999	1.499961
3	1.28E+07	1.449332	0.015005	0.052773	1.499951
4	1765662	1.527018	0.015415	0.105802	1.49979
5	1.28E+07	1.449332	0.015005	0.052773	1.499951
6	69219.47	2.829932	0.023279	0.497548	1.499825
7	3115350	1.467734	0.015013	0.087028	1.49981
8	66324.57	6.098001	0.041308	0.499978	1.499843
9	9175549	1.453202	0.015005	0.059365	1.49972
10	465306	1.571534	0.015555	0.17355	1.499823
11	69090.95	4.231669	0.031607	0.466695	1.499853
12	4799939	1.461389	0.015009	0.07464	1.499806
13	7701611	1.455654	0.015009	0.063145	1.499722
14	1.02E+07	1.451919	0.015005	0.05719	1.499824
15	66195.73	7.030898	0.045909	0.499808	1.499911
16	71460.67	2.043988	0.018211	0.49999	1.499961



Pareto front (case 4)

DISCUSSIONS

In the first case we observe that the minimum of f(1)(=53242.4) is obtained at the point (d_0=0.015005, B=0.49999, D_s=0.7557). But that point also has the maximum value for length (=3.7674) among all optimal points. Analogously the least value of f(2) of 1.2538 is at the point (d_0=0.15005, B=0.050005, D_s=1.5), but with the maximum value(=1992995) of the cost function in the list of optimal points. The point at d_0=0.015005, B=0.4386 and D_s=1.28, has moderately acceptable values for both cost (=66804.41) and length (=1.7878).

In the second case we notice a similar trend. The minima for the first objective function occurs at (d_0=0.01661, B=0.487396, D_s=0.530987) with a cost value of 50203.7, but at the same time with largest length (=7.750892) among all optimal points. On the other hand, at (d_0=0.016132, B=0.056833, D_s=1.498633), we obtain the minimum possible length (=2.512074) but also the largest cost value (=423595.5) among the points. Like the previous case, at the point (d_0=0.016274, B=0.411127, D_s=0.756528), we can see that a balance has been struck between the cost (=58633.08) and the length (=4.88362).

In the third case we observe that the minimum of f(1)(= 1382230) is obtained at the point (d_0=0.051, B=0. 5, D_s=0.948281). But that point also has the maximum value for length (=416.4354) among all optimal points. Analogously the least value of f(2) of 241.9545is at the point (d_0=0.015001, B=0.052165, D_s=1.499999), but with the maximum value(=2.08E+08) of the cost function in the list of optimal points. The point at d_0=0.020535, B=0.363654 and D_s=1.122275, has moderately acceptable values for both cost (=3975894) and length (=300.0257).

In the fourth case we observe that the minimum of f(1)(= 66155.38) is obtained at the point (d_0=0.049461, B=0.49999, D_s=1.499961). But that point also has the maximum value for length (=7.775957) among all optimal points. Analogously the least value of f(2) of 1.449332is at the point (d_0=0.015005, B=0.052773, D_s=1.499951), but with the maximum value(=1.28E+07) of the cost function in the list of optimal points. The point at d_0=0.031607, B=0.466695and D_s=1.499853, has moderately acceptable values for both cost (=69090.95) and length (=4.231669).

CONCLUSION

- Annual cost and length of heat exchanger are competing and opposing entities; i.e. increase in one, invariable produces reduction in the other.

- When the number of tube passes is two, the obtain overall lower values of the cost unction in case of the square pitch and conversely, lower values of length in case of the triangular pitch.

- When the number of tube passes is four we obtain overall lower values for both cost and length functions in case of the triangular pitch.

- When the number of tube passes is increased from two to four there is a substantial rise in the cost function values for the square layout a moderate increase in case of the triangular layout.

- On the other hand, when the number of tubes is increased, there is a tremendous rise in the overall length values for the square layout. But we also note a marked decrease for length in case of the triangular pitch.

- It can be seen that when the number of tubes is two, dimensions close to [d_0=0.015, B=0.5] favour lower cost function values. On the contrary, when dimensions are close to [B=0.05, D_s=1.5], we tend to get lesser length values.

- It can be seen that when the number of tubes is two, dimensions close to [d0=0.05, B=0.5] favour lower cost function values. On the contrary, when dimensions are close to [d_0=0.015, B=0.05, D_s=1.5], we tend to get lesser length values.

- Thus, in general for lower annual cost values we need the baffle spacing to be approximately 0.5. But for smaller lengths we want baffle space and inner shell diameter to be close to 0.05 and 1.5 respectively.

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