EXPERIMENTAL INVESTIGATION AND OPTIMIZATION OF SHELL AND TUBE HEAT EXCHANGER USING GA

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ABSTRACT

The primary goal in designing a heat exchanger is to determine the necessary heat transfer area for a given heat duty, as it directly impacts the overall cost of the heat exchanger. There are multiple design options available, involving various parameters such as outer diameter, pitch, tube length, tube passes, baffles, baffle spacing, and baffle cut. In our experiment, we focused on designing and evaluating a heat exchanger using experimental data. Specifically, we conducted the experiment using the HT30X Heat Exchanger Service unit and utilized the HT33 Shell and Tube software in conjunction with it. Previous research has extensively explored the optimization of shell and tube heat exchangers from an economic standpoint using different techniques and algorithms. However, optimization involving multiple objectives is relatively uncommon. In our work, we aimed to optimize the heat exchanger design based on two objectives: a) minimizing the length of the heat exchanger, and b) minimizing the overall cost. We have outlined a clear methodology for our study and conducted case studies to demonstrate how one objective affects the other. To facilitate the optimization process, we employed the Multi-Objective Optimization solver, which utilizes a genetic algorithm (gamultiobj). This solver is available in the optimization toolbox of the MATLAB software. Our experimental setup included a HT33 shell and tube heat exchanger with one shell, seven tubes, one pass, and two baffles. We performed iterations on the variables, including the shell diameter, outer tube diameter, and baffle spacing, to identify the optimal solution using the optimization solver.

Keywords
Optimization,multi-objective, Genetic algorithm, shell andtubeheat exchanger, Cost

INTRODUCTION

A shell and tube heat exchanger is a commonly used design in various industries, particularly in oil refineries and large chemical processes, where higher pressure applications are required. This type of heat exchanger consists of a shell, which is a large pressure vessel, housing a bundle of tubes. One fluid flow through the tubes while another fluid passes over the tubes, transferring heat between the two fluids. The bundle of tubes is known as a tube bundle and can be composed of different types of tubes, such as plain or longitudinally finned tubes. In our experiment, we utilized the HT30X Heat Exchanger Service unit and employed the HT33 Shell and Tube software in conjunction with it. We collected data, including inlet and outlet temperatures of both the shell and tube fluids, and measured the geometry of the heat exchanger unit. Using a genetic algorithm, a search-based optimization technique inspired by genetics and natural selection, we performed multi-objective optimization on the design parameters of the shell and tube heat exchanger. The variables we considered were the tube's outer diameter (do), the shell's inner diameter (Ds), and the baffle spacing (B). Our objectives were to minimize the length of the heat exchanger and reduce the overall cost, encompassing the initial investment and annual operating expenses. Genetic algorithms, developed by John Holland and his colleagues at the University of Michigan, have proven to be successful in optimizing various problems, including our heat exchanger design.

LITERATURE REVIEW

JunDarChenSingTsu Tsai [1]: The authors have developed a computation model based on analytical derivation for multi-tubepass crossflow type heat exchangers. The model focuses on finding the mean temperature difference and explores various commonly used configurations. The results provide practical references in terms of a temperature correction factor. Antonio C. Caputo, Marcello P. Pelagagge, Paolo Salini [2]: This paper emphasizes the importance of cost minimization in heat exchangers for both designers and users. Traditional design approaches rely on iterative procedures, gradually adjusting design parameters until a satisfactory solution is achieved. The authors propose a new optimization design approach, considering the minimization of annual total cost as the objective. Jiangfeng Guo, Lin Cheng, Mingtian Xu [3]: The authors present a new optimization design approach for shell-and-tube heat exchangers. Their method utilizes the dimensionless entropy generation rate as the objective function, scaling it with the ratio of the heat transfer rate to the cold fluid's inlet temperature. M. Saffar-Avval, and E. Damangir [4]: The authors utilize an optimization program to calculate the optimum baffle spacing and number of sealing strips for various types of shell and tube heat exchangers. They provide a set of correlations for determining the optimal baffle spacing, which complements the recommendations of the Heat Exchanger Design Handbook (HEDH). M.S. Abd-Elhady C. C. M. Rindt& A. A. van Steenhoven [5]: This experimental study investigates the influence of flow direction with respect to gravity on particulate fouling of heat exchangers. Four flow orientations (horizontal, upward, downward, and at a 45° angle) are examined to determine the optimal flow direction for minimizing fouling. C. P. Hedderich, M. D. Kelleher and G. N. Vanderplaats [6]: The authors have developed a computer code for analyzing air-cooled heat exchangers and coupled it with a numerical optimization program. This combination allows for an automated design and optimization procedure for air-cooled heat exchangers. The analysis method used calculates the mean overall heat-transfer coefficient and the overall pressure drop for different flow arrangements. A. Unuvar, and S. Kargici [7]: The authors propose an approach for the optimum design of heat exchangers based on the minimization of annual total cost. They compare their method to alternative optimum design approaches, such as the Lagrange multiplier method, highlighting the advantages of their approach in terms of time and efficiency.

Dr.S.ElizabethAmudhini Stephen, Joe Ajay.A, Karthiyayini.K [8]: The authors focus on cost minimization of shell and tube heat exchangers and develop a new optimization design approach using four nontraditional optimization algorithms: Genetic algorithm, Simulated Annealing, Pattern search, and fmincon algorithm. Millie Pant, Radha Thangaraj and V. P. Singh [9]: This paper introduces the Improved Constraint Differential Evolution (ICDE) algorithm for solving constrained optimization problems in heat exchanger design. The authors highlight the capabilities of Differential Evolution (DE), an evolutionary approach known for handling non-differentiable, non-linear, and multi-modal objective functions.Ming-Hua Lin, C. Jung-Fa Tsai, and Chian-Son Yu [10]: The authors discuss the theoretical and algorithmic contributions to optimization in practical applications. They classify the approaches for treating optimization problems into deterministic and heuristic methods.P.D. Chauduri, U. Diwekar, and J. Logsdon [11]: This paper presents an efficient strategy for the optimal design of heat exchangers using simulated annealing (SA). The authors demonstrate how an algorithmic procedure for large-scale optimization problems is well-suited for addressing the discrete optimization problem in heat exchanger design. They develop a command procedure to run the SA algorithm iteratively with the design program.R. Selbas, O. Kizilkan, and M. Reppich [12]: The authors focus on the optimal design of shell-and-tube heat exchangers, considering it as the main objective of their study. They highlight the significance of heat exchangers in various systems and propose formulating the design as an optimization problem solved with particle swarm optimization.Mauro A. S. S. Ravagnani, Aline P. Silva, Evaristo C. Biscaia Jr., and Jose A. Caballero [13]: The authors formulate the design of shell and tube heat exchangers as an optimization problem. They employ particle swarm optimization to minimize the global cost, considering area cost and pumping cost or area minimization, while adhering to the standards of the Tubular Exchanger Manufacturers Association and respecting pressure drops and fouling limits.Nairen Diao, Qinyun Li, Zhaohong Fang [14]: The authors establish an equation of conduction-advection for heat transfer in porous media to estimate the impact of groundwater flow on the performance of geothermal heat exchangers in heat pump systems. They provide an analytical and transient method for a heat source in an infinite medium using green function analysis.B. Bansal and H. Muller Steinhagen [15]. This study investigates the crystallization and also the fouling of calcium sulfatein a heat exchanger. The effects of flow velocityand wall temperature, and CaSO4 concentration on fouling rates are examined. W.J. Marner, A.E. Bergles, and J.M. Chenoweth [16]: The authors address the challenges faced by designers in comparing and evaluating the performance of various augmented surfaces in heat exchangers. They emphasize the need for a standardized presentation of test data in the literature, as the production of augmented surfaces increases.

METHODOLOGY

We experimented on HT30X Heat Exchanger Service unit. The software HT33 Shell and Tube will be used with the HT30X Heat Exchanger Service unit.The shell and tube heat exchanger is commonly used in the food and chemical process industries. This type of exchanger mainly has a tubes in parallel which are enclosed in a cylindrical shell. Heat is transferred between one fluid flowing through the tubes and another fluid flowing through the cylindrical shell around the tubes

Equipment and its size

Exchanger contains a shell and seven tubes.

Two transverse baffles.

The exchanger is mounted on a PVC base plate which incorporate four holes.

Tubes made from stainless steel.

The outer annulus, caps and baffles are constructed from clear acrylic.

The end housings incorporate the necessary fittings for sensors.

There are four thermocouples for temperature measurement.

Heat exchanger service unit “HT30X”.

Heat transmission length = 1.008 m

Heat transmission length of each tube = 0.144 m

Tube inside diameter = 0.00515 m

Tube outside diameter = 0.00635 m

Shell inside diameter = 5.1 cm

Energy Balance Equations

Mass flowrate(qm) = Volume flowrate(qv)*Density of fluid(ρ) (kg/sec)

Heat Power(Q) = Mass flowrate(qm)*specific heat(Cp)*change in temperature(ΔT) (W)

Heat power emitted from hot fluid0 Qe = q*mh*Cph(T1-T2) (W)

Heat power absorbed by cold fluid Qa= q*mc*Cpc(T4-T3) (W)

LMTD = (ΔT1 – ΔT2) / ln (ΔT1 /ΔT2)

Counter Current Operation

When the heat exchanger is connected for counter current operation the hot and cold fluid streams flow in opposite directions across the heat transfer surface.

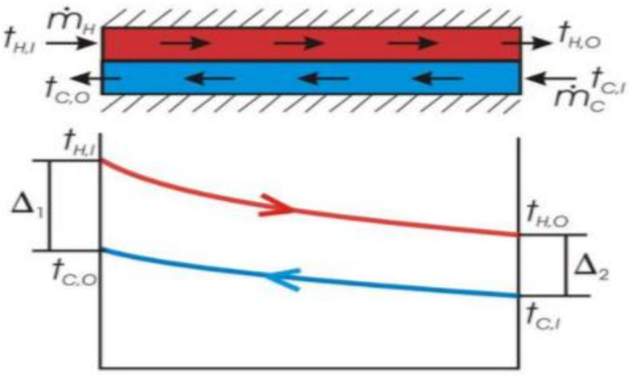

Figure 1: Temperature profiles in C

$t_{H,I}$ = T1 , $t_{C,I}$ = T3

$t_{H,O}$ = T2 , $t_{C,O}$ = T4

1. Reduction in hot fluid temperature ΔThc
2. Increase in cold fluid temperature ΔTco
3. Heat power emitted from hot fluid Qe =
4. Temperature efficiency for hot fluid η h = (T1 – T2)/(T1 – T3)*100 (%)
5. Temperature efficiency for cold fluid η c = (T4 – T3)/(T1 – T3)*100 (%)
6. Mean Temperature efficiency η m = (η h + η c)/2 (%)

Co-current operation

When the heat exchanger is connected for co-current operation the hot and the cold fluid streams flow in the same direction across the heat transfer surface.

$t_{H,I}$ = T2 , $t_{C,I}$ = T3

$t_{H,O}$ = T1 , $t_{C,O}$ = T4

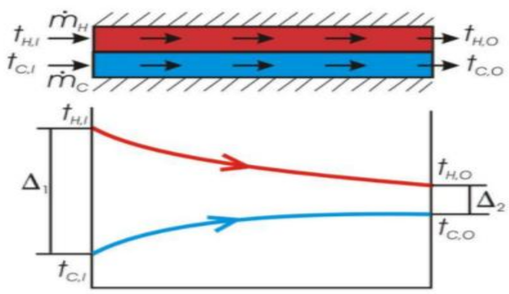

Figure 2: Temperature profiles in Co-current flow

1. Reduction in hot fluid temperature ΔThot =T2 – T1 (°C)
2. Increase in cold fluid temperatureΔTcol =T4 – T3 (°C)
3. Heat power emitted from hot fluid Qe = q*m_h*C_{ph}(T_2-T_1) (W)
4. Temperature efficiency for hot fluid η h = (T2 – T1)/(T2 – T3)*100 (%)
5. Temperature efficiency for cold fluid η c = (T4 – T3)/(T2 – T3)*100 (%)
6. Mean Temperature efficiency η m = (η h + η c)/2 (%)

Experiment

After install setup, we took several readings those are based on temperature and flowrate of cold water and hot water. First, we started counter-current then co-current flowrate readings.

We start the setup, here a heater is also installed, with this heater we heated water, that water flow in tube and that tube water cools with the help of cold water in shell.

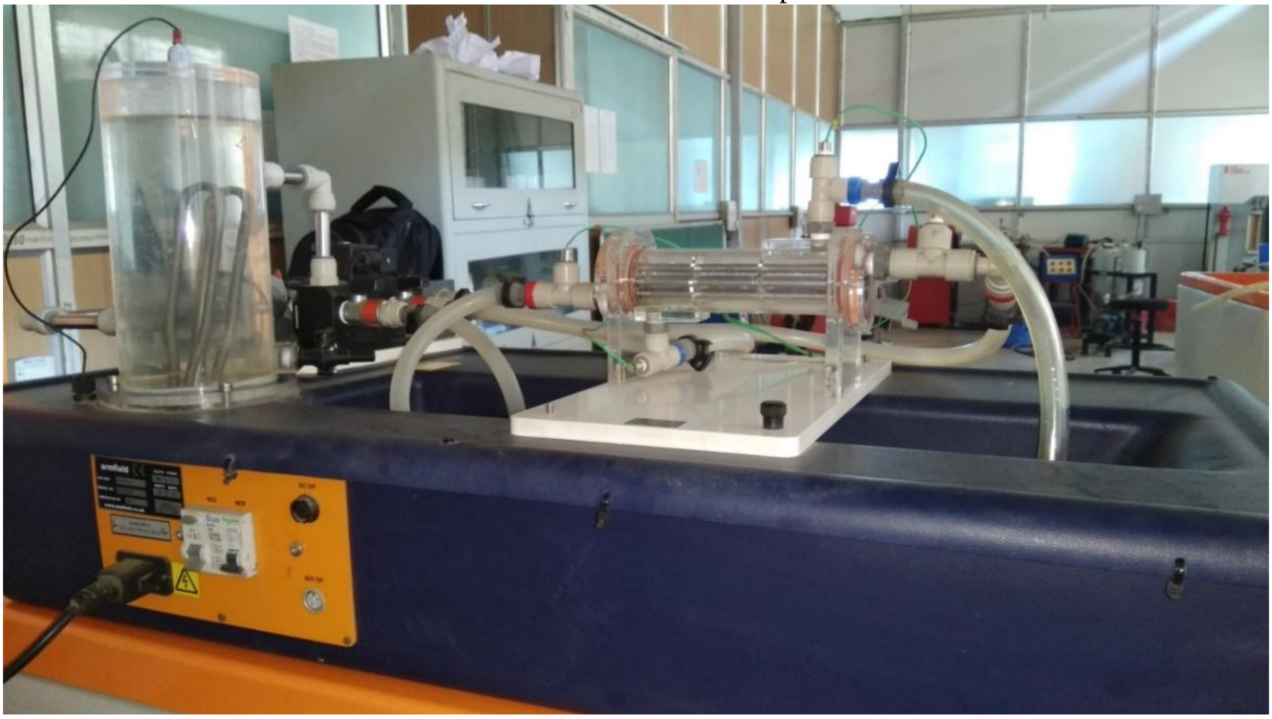

Figure 3: Equipment setup

Counter-Current Flowrate

We take reading in every 15 second, after completed 20 reading, we set new hot temperature.

First, we set shell side flowrate constant then varying tube side flowrate.

Table1: List of counter current readings of different temperature and flowrate.

No.	Cold Water (shell side) Flowrate (L/min)	Hot Water (tube side) Flowrate (L/min)	Temperature (at which readings are noted.)
1	1.55	1.00	40,45,50,55,60,65,70
2	1.55	1.55	40,45,50,55,60,65,70
3	1.55	2.00	40,45,50,55,60,65,70
4	1.55	2.55	45,50,55,60,65
5	2.00	1.00	45,50,55,60,65,70
6	2.00	1.50	45,50,55,60,65
7	2.00	2.00	45,50,55,60,65
8	2.00	2.50	50,55,60,65

Co-Current Flowrate

We take reading in every 15 second, after completed 20 reading, we set new hot temperature.

First, we set shell side flowrate constant then varying tube side flowrate.

Table 2: List of Co-Current readings of different temperature and flowrate.

No.	Cold Water (shell side) Flowrate	Hot Water (tube side) Flowrate	Temperature (at which readings are noted.)

	(L/min)	(L/min)	
1	1.00	1.00	50,55,60,65,70
2	1.00	1.50	50,55,60,65,70
3	1.00	2.00	50,55,60,65,70
4	1.00	2.50	55,60,65,70
5	1.00	3.00	55,60,65
6	1.00	3.50	55
7	2.00	1.00	50,55,60,65,70
8	2.00	1.50	55,60,65,70
9	2.00	2.00	55,60,65,70

Genetic Algorithms

For multi-objective optimization of shell and tube type heat exchanger, we will use GA in MATLAB.

Two objective functions are simultaneously optimized to obtain a set of solutions that yield the best values for both functions

1. The cost which consists of the initial investment and the annual cost of operation (Ctot)
2. Length of the heat exchanger (L)

The three design variables are:-

1. Tube outside diameter (d0)
2. Shell inside diameter (Ds)
3. Baffle spacing (B)

The optimization procedure was implemented by a multiobjective genetic algorithm (GA). In this case, a heat exchanger of specific configuration with the design specifications, the GA was used to optimize. 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.

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: 1000
Time Limit: Default (Infinite)
Fitness Limit: Default (Infinite)
Stall Generations: Default (100)
Function Tolerance: 1e-4.

Result and Discussion

After install set-up we took several readings with different temperature and flowrate conditions. There are many thermocouples set in our setup and we can see the temperature of that part where thermocouple is installed.With the help of inlet and outlet temperature of shell and tube heat exchanger and flowrate of cold and hot water, we can easily find the LMTD and Overall heat transfer coefficient.At a constant temperature, flowrate we took several readings by using sample configuration, we took 30 second as sample interval.

1	65	70.1	32.1	38.1	30	2.5	100	2.04
2	65.2	70.2	32	38.1	30	2.5	100	2.06
3	65.1	70	31.9	38	30	2.5	100	2.08
4	65	69.9	32	38	30	2.5	100	2.09
5	65.1	70	32	37.9	30	2.5	100	2.08
6	65.1	70.1	32	38	31	2.5	100	2.08
7	65.1	70	32	38	31	2.5	100	2.08
8	65.2	70.1	32	38	31	2.4	100	2.08
9	65.3	70	32	38.1	32	2.4	100	2.08
10	65.2	69.9	32	38.1	32	2.4	100	1.99
11	65.3	70	32	38.1	32	2.6	100	2.01
12	65.3	70.1	32	38.2	32	2.5	100	2.09
13	65.3	70.2	32	38.2	32	2.5	100	2.05
14	65.5	70.2	31.9	38.2	32	2.5	100	2.04
15	65.4	70	31.9	38.2	33	2.4	100	2.03
16	65.3	70	31.9	38.2	35	2.4	100	2.06
17	65.5	70.1	31.9	38.2	35	2.5	100	2.05
18	65.5	70	31.9	38.3	35	2.5	100	2.04
19	65.4	69.8	31.8	38.3	35	2.5	100	2.06
20	65.4	69.8	31.8	38.2	34	2.5	100	2.05
21	65.4	69.9	31.8	38.2	35	2.5	100	2.05
22	65.5	70	31.8	38.2	35	2.4	100	2.08
23	65.5	70	31.8	38.2	35	2.5	100	2.08
24	65.4	69.9	31.8	38.2	35	2.6	100	2.05
25	65.4	69.9	31.7	38.2	35	2.5	100	2.05
26	65.5	70.1	31.7	38.1	35	2.5	100	2.04
27	65.6	70.2	31.7	38.1	35	2.5	100	2.05
28	65.5	70.1	31.7	38.1	34	[illegible]		

Figure 4: Hot and cold-water temperature and flowrate

Flow Orientation	Notes	Specific Heat hot Fluid cph [kj/kg K]	Specific Heat cold Fluid cpc [kj/kg K]	Hot Fluid Avg. Temp. [˚C]	Density Hot Fluid [kg/m^3]	Cold Fluid Avg Temp [˚C]
Cocurrent		4.188	4.178	67.55	979.2	35.05
Cocurrent		4.188	4.178	67.56	979.1	35.05
Cocurrent		4.188	4.178	67.55	979.2	34.97
Cocurrent		4.188	4.178	67.47	979.2	34.95
Cocurrent		4.188	4.178	67.55	979.2	34.97
Cocurrent		4.188	4.178	67.63	979.1	34.98
Cocurrent		4.188	4.178	67.58	979.1	34.98
Cocurrent		4.188	4.178	67.52	979.1	35
Cocurrent		4.188	4.178	67.53	979.1	35.03
Cocurrent		4.188	4.178	67.58	979.1	35.03
Cocurrent		4.188	4.178	67.55	979.1	35.03
Cocurrent		4.188	4.178	67.73	979.1	35.05
Cocurrent		4.188	4.178	67.75	979.1	35.05
Cocurrent		4.188	4.178	67.83	979	35.03
Cocurrent		4.188	4.178	67.7	979	35.03
Cocurrent		4.188	4.178	67.66	979.1	35.03
Cocurrent		4.188	4.178	67.81	979.1	35.03
Cocurrent		4.188	4.178	67.73	979	35.05
Cocurrent		4.188	4.178	67.6	979.1	35.03
Cocurrent		4.188	4.178	67.82	979.1	35
Cocurrent		4.188	4.178	67.88	979.1	35
Cocurrent		4.188	4.178	67.78	979.1	34.98
Cocurrent		4.188	4.178	67.78	979	34.98
Cocurrent		4.188	4.178	67.85	979	34.97
Cocurrent		4.188	4.178	67.83	979.1	34.93
Cocurrent		4.188	4.178	67.81	979	34.89
Cocurrent		4.188	4.178	67.91	979	34.89
Cocurrent		4.188	4.178	67.81	979	34.9
Cocurrent		4.188	4.178	67.73	979.1	34.82
Cocurrent		4.188	4.178	67.55	979.1	34.89

Figure 5: Specific heat and density of hot and cold water

Overall efficiency [%]	Temp. Efficiency of hot fluid [%]	Temp. Efficiency of cold fluid [%]	Mean Temp Efficiency [%]	LMTD	Oveall Heat Transfer cofficient U	Question Scored [%]
				32.16	1422.6	
				32.32	1397.04	
				32.27	1392.57	
				32.21	1352.38	
Hidden	Hidden	Hidden	Hidden	32.26	1371.82	
				32.34	1405.25	
				32.29	1353.26	
				32.31	1319.16	
				32.32	1297.03	
				32.25	1282.02	
				32.31	1362.19	
				32.37	1336.19	
				32.38	1324.93	
				32.49	1319.76	
				32.36	1224.41	
				32.33	1232.01	
				32.46	1263.33	
				32.36	1264.57	
				32.26	1237.95	
				32.31	1245.22	
				32.37	1236.46	
				32.47	1223.1	
				32.49	1256.51	
				32.38	1266.81	
				32.39	1232.97	
				32.62	1257.12	
				32.72	1271.72	
				32.6	1270.15	
				32.61	1291.34	
				32.67	1291.92	

Figure 6: Temperature, LMTD and overall heat transfer coefficient

Genetic Algorithms

Here we observe that the minimum of f(1)(=7999.991) is obtained at the point (do=0.00899, B=0.09994, Ds=0.3621). But that point also has the maximum value for length (=4.0687) among all optimal points. Analogously the least value of f(2) of 0.0442 is at the point (d0=0.001, B=0.053, Ds=0.4994), but with the maximum value(=8000.0592) of the cost function in the list of optimal points. The list of all feasible solutions with their functional values and the pareto front with length of the heat exchanger have been illustrated before.

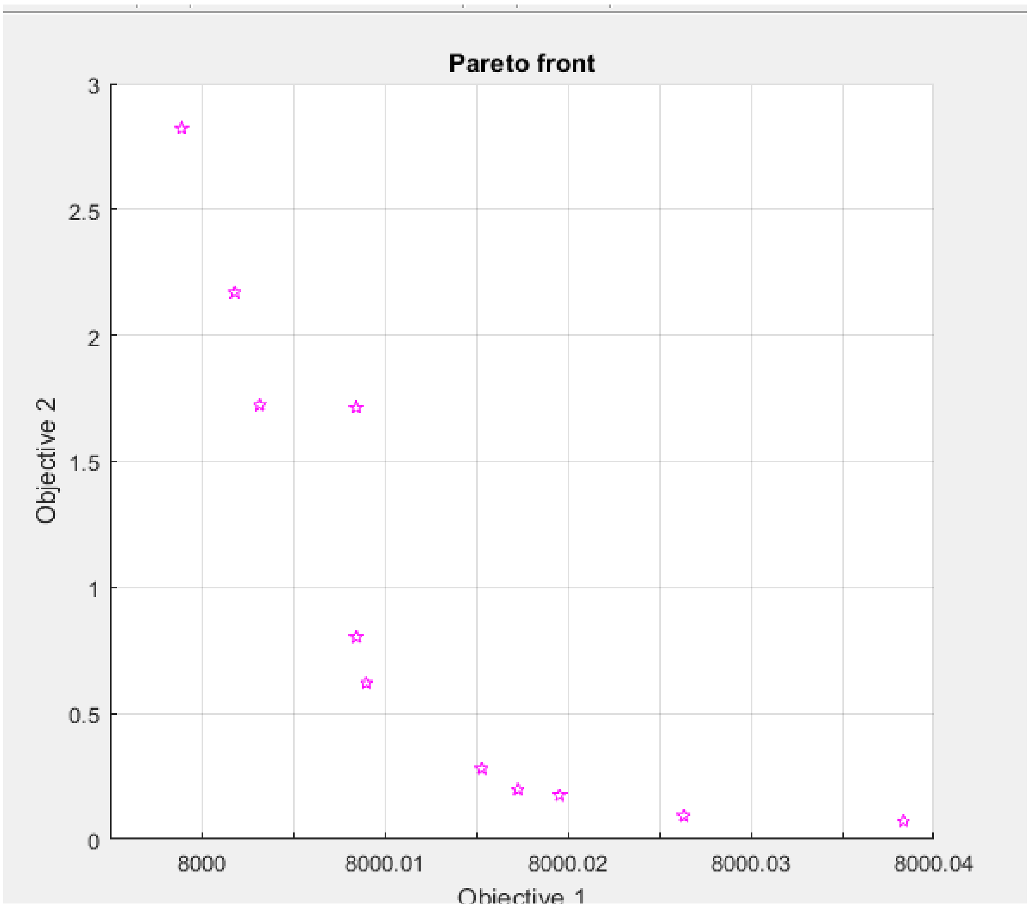

Figure 7: Pareto front depicting the optimal points

It can be seen that, dimensions close to [do=0.00899, B=0.09994] favour lower cost function values. On the contrary, when dimensions are close to [B=0.053, Ds=0.4994], we tend to get lesser length values. Thus, in general for lower annual cost values we need the baffle spacing to be approximately 0.1. But for smaller lengths we want baffle space and inner shell diameter to be close to 0.053 and 0.5 respectively.

Conclusion

We worked on Heat exchanger and find different data. Data includes inlet and outlet temperature of shell side and tube side, flow rate of hot and cold fluid, density of hot and cold fluid, specific Heat of hot and cold fluid, mass flow rate of hot and cold fluid, LMTD, overall heat transfer coefficient.Then we worked on Matlab, our work is multiobjective optimization of shell and tube type heat exchanger. Our variable are shell diameter, tube diameter and baffle spacing. Our objective are minimization of length of heat exchanger and overall cost. With the help of optimtool we completed our task and make a perfect optimization. Annual cost and length of heat exchanger are opposing entities; i.e. increase in one, invariable produces reduction in the other. This could be attributed to the fact that the daily operating cost is inversely proportional to average duration of operation per day. When the length of the heat exchanger increases, so does the net surface area available for heat exchange, due to which the overall efficiency of heat transfer for a heat exchanger of a given duty cycle and given temperature difference between working fluids increases. Hence it needs to work for ewer hours.

Appendix A
Code of Genetic Algorithm

```
function Y=bs(x)
```

```
a1 = 8000 ;
a2 = 259.2 ;
a3 = 0.91  ;

%cold = shell
%hot = tube

mc= 0.034;
%mh = 0.041;
ms = 0.034 ;
mt = 0.041 ;

Cpc= 4.178 ;
%Cph = 4.188 ;
Cps = 4.178 ;
Cpt = 4.188 ;

Tci = 32.1 ;
Tco = 38.1;
Thi = 70.1 ;
Tho = 65.0 ;

ks = 0.62 ;
kt = 0.62 ;

mus = 0.000720 ;
mut = 0.000416 ;

muwts = 0.000720 ;
muwt  = 0.000416 ;

Rs =0.00017 ;
Rt =0.00017 ;

rhos = 994 ;
rhot = 979.2 ;

sigmas=0.07035 ;
sigmat=0.06517 ;

n=1;
eta=0.85 ;
p=4;

CE=0.12 ;
H=500 ;
bo=0.72 ;

C= 0.249;
n1 = 2.207 ;

%Q = mh*Cph*(Thi-Tho) =mc*Cpc*(Tco-Tci) ;.249   2.207
Q=mc*Cpc*(Tco-Tci) ;
%Q = mh*Cph*(Thi-Tho);

% x(1)=do= Tube outside diameter ;
% x(2)=Ds= Shell inside diameter ;
% x(3)=B=  Baffle spacing ;
di= 0.8*x(1) ;
dt=(x(1)+ di)/2 ;
St=3.14*dt ;
```

```
% Shell hydraulic diameter
de =4*(0.43*(St.^2)-((0.5*3.14*x(1).^2)/4))/(0.5*3.14*x(1)) ;

% Cross Section area normal to flow direction
As =x(2)*(-x(3))*(1-(x(1)/St)) ;

% Shell side Reynolds number
Res=(ms*de)/(As*mus);

% Shell side Prandtl number
Prs=(mus*Cps)/ks ;

% Shell side heat transfer coefficient
hs   =(0.36*kt*(Res^0.55)*(Prs^0.33)*(mus^0.14))/((de)*(muwts^0.14))
;

% Number of tubes
Nt = C*(x(2)/x(1))^(n1) ;

% Tube side Flow velocity
vt = (4*mt*n)/(3.14*(dt^2)*rhot*Nt) ;

% Tube side Reynolds number
Ret = (rhot*vt*di)/mut ;

% Tube side Prandtl number
Prt = (mut*Cpt)/kt ;

% Tube-side heat transfer coefficient
ht   =   (0.027*kt*(Ret^0.8)*(Prt^0.33)*(mut^0.14))/(x(1)*(muwt^0.14))
;

% The overall heat transfer coefficient
U =1/((1/hs) + Rs + (Rt+(1/ht))*(x(1)/di));

% Logarithmic mean temperature difference
LMTD=((Thi-Tco)-(Tho-Tci))/2.303*log((Thi-Tco)/(Tho-Tci));

% The correction coefficient
R =(Thi-Tho)/(Tco-Tci) ;

% The efficiency
P =(Tco-Tci)/(Thi-Tci) ;

% The correction factor
F      =(((R^2+1)^0.5)*log((1-P)/(1-P*R)))/((R-1)^0.5)*log((2-P*R+1-
((R^2+1)^0.5))/(2-P*R+1+((R^2+1)^0.5))) ;

% Surface area of heat exchanger
S=Q/U*LMTD*F;

% The capital investment
Ci=a1+a2*S^a3 ;

% A=the heat exchanger surface area
A=Q/(U*F*LMTD) ;

% Y(2)=L= Length of the heat exchanger
Y(2)=A/(3.14*x(1)*Nt) ;

% Darcy tube side friction factor
ft= (1.82*(log(10^Ret))-1.64)^(-2) ;
```

```
% Tube side pressure drop
delPt = rhot*(vt^2)*0.5*n*((Y(2)*ft/di)+p);

% Shell side friction factor
fs=2*bo*Res^(-0.15) ;

% Shell side flow velocity
vs=ms/rhos*As;

% shell side pressure drop
delPs=(fs*rhos*(vs^2)*Y(2)*x(2))/(2*x(3)*de);

% pumping power
PP= ((mt*delPt/sigmat)+(ms*delPs/sigmas))/eta ;
Co=PP*CE*H ;

% The total discounted operating cost related to pumping power
Cod=(Co/11) + (Co/11^2) +(Co/11^3) ;

% Y(1)=Ctot=The cost which consists of the initial investment and the annual cost of operation
Y(1)=Ci+Cod ;

end
```

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