Steam Surface Condenser Design based on Cost Optimization using Genetic Algorithm

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By

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Declaration

This is to certify that

1. The thesis comprises my original work towards the degree of Master of Technology in Mechanical Engineering (Thermal Engineering) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgement has been made in the text to all other material used.

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Certificate

This is to certify that the Major project entitled "Steam Surface Condenser Design based on cost optimization using Genetic Algorithm" submitted by **Nirmalkumar P Bhatt (09MMET02),** towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (Thermal Engineering) of Nirma University, Ahmedabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The results embodied in the major project, to the best of our knowledge, haven't been submitted to any other university or institution for award of any degree or diploma.

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Abstract

Steam surface condenser is a shell and tube heat exchanger used in power plant to condense the steam. Cost minimization of this condenser design is a key objective for both designer and users. In the design of condenser, one need to check the thermal performance which satisfies process condition, while rating depends on number of parameters like tube geometry, tube size, tube layout, and working fluid condition. In the present work, rating of surface condenser has been carried out using appropriate condensation correlation for tube bundle and tubeside flow. The variation of thermo-physical properties of water and steam with temperature are also considered to evaluate the overall heat transfer coefficient for surface condenser. A Computer program has been prepared to solve the problem efficiently. The results of overall heat transfer coefficient calculated by the program are compared and found to be within 3.33% of deviation with the results obtained from HTRI software. The Results are also compared with the experimental results of HEI standard. These also shown good agreements with available experimental result for lower tubeside velocity (<2.3 m/s) and the deviation found is within 11.256%.The present study also explores the use of a non-traditional optimization technique; Genetic Algorithm (GA), for optimization of steam surface condenser. The code for GA is developed and successfully applied for the optimization of same by varying the design variables such as shell internal diameter, tube outer diameter, tube thickness and tube material. The two different ranges of design variables are used. One is Siemens range (dimension range of earlier parameters are used by Siemens Ltd., Baroda) and second is TEMA Range (dimension range of same parameters taken from TEMA standard). The objective function for capital cost and total cost is derived and same is used for optimization. The optimized results obtained from selecting these range are compared and it was found that by widening the design variables range using total cost as an objective function , the efficiency of GA for optimization improves.

Keywords: Surface condenser, Optimization, Genetic algorithm

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Nomenclature

a_1	numerical constant ($\$$)
a_2	numerical constant ($\$/m^2$)
a_3	numerical constant
A	heat transfer area (m^2)
B	baffles spacing (m)
B_c	baffles cut (%)
c	flow constant
C_i	capital investment ($\$$)
c_l	clearence (m)
C_N	inundation correction factor
C_{od}	total discounted operating cost ($\$$)
C_o	annual operating cost ($\$/yr$)
c_p	specific heat at constant pressure (kJ/kgK)
C_{tot}	total cost ($\$/yr$)
C_{ug}	shear correction factor
d	diameter (m)
D_e	equivalent shell diameter (m)
D_s	Shell inside diameter (m)
ec	energy cost ($\$/kWh$)
f	friction coefficient
fd	unit cost of heat exchanger per area ($\$/m^2$)
g	gravitational force (N)
h	heat transfer coefficient (W/m^2K)
H	annual operating time (h/yr)
h_G	gravitational heat transfer coefficient (W/m^2K)
h_{sh}	shear heat transfer coefficient (W/m^2K)
i_{fg}	latent heat of vaporization (kJ/kg)
i'_{fg}	modified latent heat of vaporization (kJ/kg)
k	thermal conductivity (W/mK)

L	tube length (m)
m	flow constant
n	flow constant
Nu	Nusselt number
P	pumping power (W)
p_L	longitudinal pitch (m)
Pr	Prandlt number
p_t	transverse pitch (m)
Re	Reynolds number
$\tilde{R}e$	average Reynolds number
T	temperature $(^{\circ}C)$
C_1	capital recovery factor
$C2$	capital investment factor
C_i	capital cost (Rs/yr)
C_o	operating cost (Rs/yr)
C_{tot}	Total cost (Rs/yr)
U	overall heat transfer coefficient (W/m^2K)
v	velocity (m/s)
w	mean flow width (m)
x	dryness fraction

Greek letters

$\triangle E_p$	pumping power (W)
$\triangle p$	pressure drop (Pa)
$\triangle T$	temperature difference $(^{\circ}C)$
δ	film thickness (m)
θ_{tp}	tubes layout angle $(^{\circ})$
μ	dynamic viscosity $(Pa \cdot s)$
ρ	density (kg/m^3)
ϕ	circumferential angle $(^{\circ})$

Subscipts

g	gas
l	luquid
m	mean
N	number of tube
o	outer side
s	shellside
sat	saturated
t	tubeside
w	wall

Chapter 1

Introduction

1.1 General

Condenser is a most important component in convectional power plant. Condensers are normally used in process industries, refrigeration and air conditioning, and oil refinery also. Shell and tube condenses are the most commonly used heat exchangers in process industries because they are very simple in manufacturing and their adaptability to different operating conditions.

Steam surface condenser is a shell and tube heat exchanger installed on the exhausted steam from a steam turbine in thermal power stations. In Steam surface condenser, condensation process takes place where the steam changes phase from gas to liquid (water) through the rejecting heat to the tubes to cooling water at a pressure below atmospheric pressure. Where cooling water is in short supply, an air-cooled condenser is often used. Surface condensers are also used in applications and industries other than the condensing of steam turbine exhaust in power plants. Its performance has critical effects on overall operation system. The typical power plant condenser is show in Figure1.1

The design of Steam surface condenser, including thermodynamic and fluid dynamic design, cost estimation and optimization, represents a complex process containing an

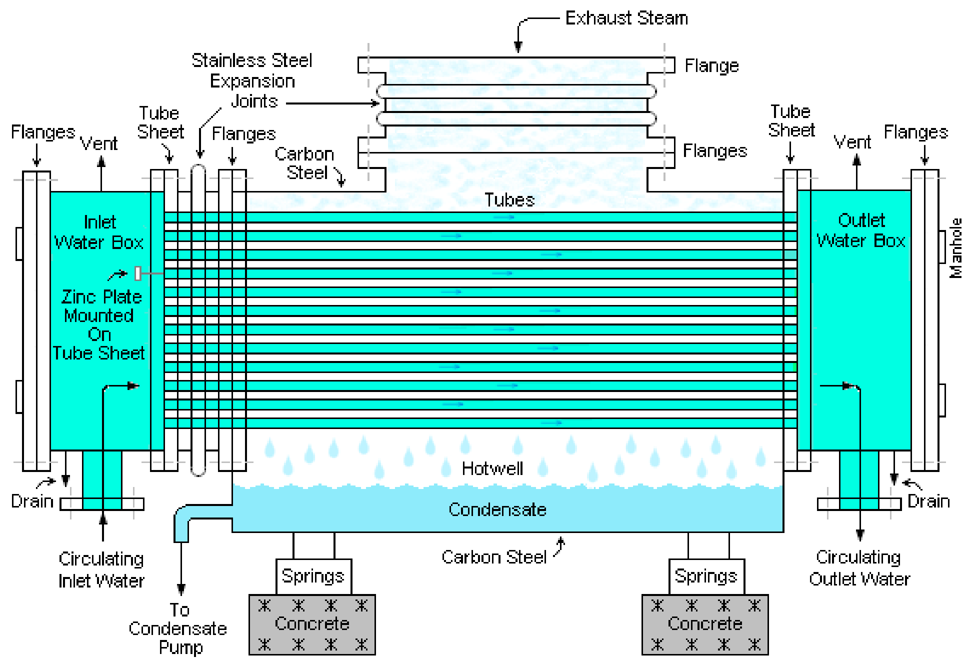


Figure 1.1: Steam surface condenser

integrated whole of design rules and empirical knowledge of various field. There are many previous studies on the optimization of heat exchanger. Several investigations had been done using different optimization techniques considering different objective functions to optimize heat exchanger design.

Classification of condenser

It is classified according to transfer process; Direct contact and Indirect contact which further sub-classified as below[1]:

1. Direct contact
   (a) Pool
   (b) Spray and tray

   (c) Packed column

2. Indirect contact

   (a) Shell-and-tube

      i. Power Industry

         A. surface condenser

         B. Feed water heater

      ii. Process Industry

         A. E type TEMA

         B. G type TEMA

         C. H type TEMA

         D. J type TEMA

         E. X type TEMA

         F. Total condensation (Reflux and knockback)

   (b) Extended surface

      i. Tube-fin air-cooled condenser

      ii. Plate fin cryogenic condenser

   (c) Plate-type

      i. Plate

      ii. Spiral

Condenser is also classified according to flow arrangement in two types; Jet condenser and Surface condenser which further sub-classified as[2]:

1. Jet condenser

   (a) Parallel flow type

   i. Low level type
   ii. High level type

   (b) Counter flow type

   (c) Ejector type

2. Surface condenser

   (a) Down-flow type

   (b) Central-flow type

   (c) Inverted-flow type

   (d) Regenerative type

   (e) Evaporative type

Surface condenser is further classified as Air-cooled surface condenser and water cooled surface condenser. An air-cooled condenser is significantly more expensive and cannot achieve as low a steam turbine exhaust pressure as water cooled surface condenser. Therefore in power industry water cooled surface condenser is normally used.

1.2 Need of steam surface condenser

The main purposes of the condenser are to condense the exhaust steam from the turbine for reuse in the cycle and to maximize turbine efficiency by maintaining proper vacuum. As the operating pressure of the condenser is lowered (vacuum is increased), the enthalpy drop of the expanding steam in the turbine will also increase. This will increase the amount of available work from the turbine (electrical output). By lowering the condenser operating pressure, the following will occur:

- Increased turbine output
- Increased plant efficiency
- Reduced steam flow (for a given plant output)

It is therefore very advantageous to operate the condenser at the lowest possible pressure (highest vacuum).

1.3 Importance of steam surface condenser

The steam turbine itself is a device to convert the heat in steam to mechanical power. The difference between the heat of steam per unit weight at the inlet to the turbine and the heat of steam per unit weight at the outlet to the turbine represents the heat which is converted to mechanical power. Therefore, the more the conversion of heat per pound or kilogram of steam to mechanical power in the turbine, the better is its efficiency. By condensing the exhaust steam of a turbine at a pressure below atmospheric pressure, the steam pressure drop between the inlet and exhaust of the turbine is increased, which increases the amount of heat available for conversion to mechanical power. Most of the heat liberated due to condensation of the exhaust steam is carried away by the cooling medium (water or air) used by the surface condenser.

1.4 Condensation of condenser

Condensation

When steam comes in contact with a surface at a lower temperature, condensation occurs. The most common type of condensation affects in heat exchangers is surface condensation where a cooled wall, at a temperature less than the local saturation temperature of the vapor, is comes in contact with a vapor. In this situation, the vapor molecules that strike the cold surface may strike to it and condense into liquid. The resulting liquid (i.e., condensate) will accumulate in one of two ways.

If the liquid wets the cold surface, the condensate will form a continuous film and this mode of condensation is known as film wise condensation. If the liquid does not wet the cold surface, it will form into numerous microscopic droplets. This mode of condensation is known to as a drop wise condensation. It results in much larger heat transfer coefficients than during film wise condensation. Because long-term drop wise condensation condition are very difficult to sustain. Today all surface condensers are designed to operate in the film wise mode.

1.4.1 Film condensation on a single horizontal tube

1.4.1.1 Laminar film condensation

A generic problem for both power and plant process tubular condensers is that of predicting the condensation heat transfer coefficient on the shell side. In the vacuum condenser for power plant, the main problem affecting shellside heat transfer performance is that of removal of the unknown quantity of air that leak into the steam at the turbine gland.

Nusselt processed the case of laminar film condensation of a quiescent vapor on an isothermal horizontal tube as described in Figure 1.2.

Here, the motion of the condensate is determined by a balance of gravitational and viscous forces. In the Nusselt analysis, convection terms in the energy equation are neglected: thus the local heat transfer coefficient around the tube can be written as:

$$h(\phi) = \frac{k_l}{\delta(\phi)} \tag{1.1}$$

Here δ is the film thickness that is a function of the circumferential angle ϕ. At the top of the tube, where the film thickness is minimum and the heat transfer coefficient

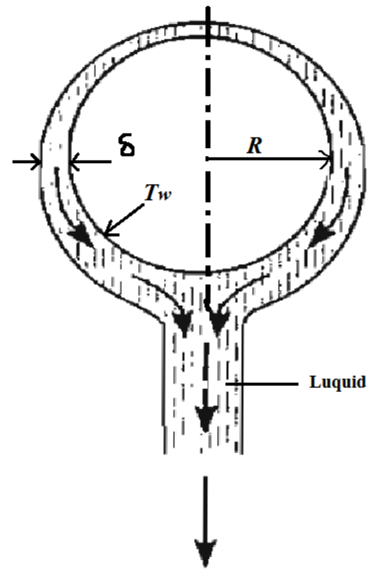


Figure 1.2: Flow on a horizontal tube

is maximum but it is zero when the film thicken increases to infinity. The Nusselt theory gives the following average heat transfer coefficient[3]:

$$\frac{h_m d_o}{k_l} = 0.728 \left[\frac{\rho_l(\rho_l - \rho_g) g i_{fg} d_o^3}{\mu_l (T_{sat} - T_w) k_l} \right]^{0.25} \tag{1.2}$$

Rohsenow[4] recommended new term to above equation using modified latent heat due to thermal advection effects which is

$$i'_{fg} = i_{fg} + 0.68 c_{p,l} \left(T_{sat} - T_w \right) \tag{1.3}$$

Thus equation (1.2) becomes

$$\frac{h_m d_o}{k_l} = 0.728 \left[\frac{\rho_l(\rho_l - \rho_g) g i'_{fg} d_o}{\mu_l (T_{sat} - T_w) k_l} \right]^{0.25} \tag{1.4}$$

1.4.1.2 Forced convection

When the vapor around a horizontal tube is moving at high velocity, the analysis for film condensation is affected in two important ways

1. The surface shear stress between the vapor and the condensate and
2. The effect of vapor separation.

The analytic investigations of this problem were extensions of Nusselt's analysis to include the interfacial shear boundary condition at the edge of the condensate film. Shekriladze and Gomelauri[5] assumed that the primary contribution to the surface to the surface shear was due to the challenge in momentum across the interface. Their simplified solution for an isothermal cylinder without separation and with no body forces is

$$Nu_m = \frac{h_m d}{k_l} = 0.9\tilde{Re}^{1/2} \tag{1.5}$$

Where $\tilde{Re}$ is defined as a two –phase Reynolds number involving the vapor velocity and condensate properties $\rho vd/\mu$, When both gravity and velocity are included, they recommend the relationship:

$$\frac{Nu_m}{\tilde{Re}^{0.25}} = 0.64\left(1 + (1 + 1.69F)^{1/2}\right)^{1/2} \tag{1.6}$$

where

$$F = \frac{gd\mu_l i_{fg}}{v_g^2 k_l \triangle T} \tag{1.7}$$

Equation 1.3 neglects vapor separation, which occurs somewhere between 82° and 180° from the stagnation point of the cylinder. After the separation point, the condensate film rapidly thickens and, as a result, heat transfer is reduced. A conservative approach suggested by Shekriladze and Gomelauri is to assume that there is no heat transferred beyond the separation point. If the minimum separation angle of 82° is

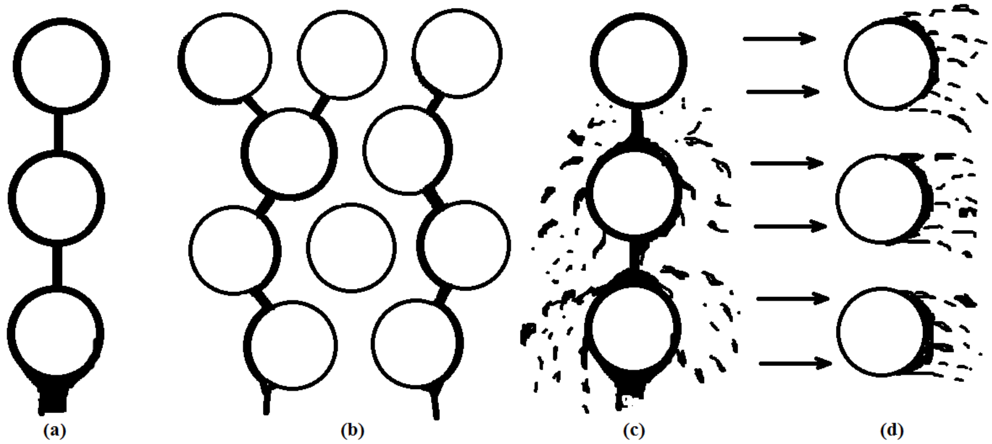


Figure 1.3: Schmetic representation of condensate flow

then chosen, a most conservative equation results and the heat transfer decreases by approximately 35%. Therefore, Equation 1.5 reduces to:

$$Nu_m = 0.59\tilde{Re}^{1/2} \tag{1.8}$$

An interpolation formula based on this conservative approach, which satisfied the extremes of gravity-controlled and shear-controlled condensation, was proposed by Butterworth[5]:

$$\frac{Nu_m}{\tilde{Re}^{0.25}} = 0.416\left(1 + (1 + 1.947F)^{1/4}\right)^{1/4} \tag{1.9}$$

Vapor boundary layer effects, especially separation, and the pressure gradient effect around the lower part of the tube provide significant difficulties in arriving at an accurate analytic solution.

1.4.2 Film condensation over tube bundles

During film condensation over tube bundles, the conditions are much different than for a single tube. The neighboring multiple tubes in the presence of vapor flow creates complexities as explained schematically in figure 1.3[6].

In the idealized case figure a condensate from a given tube is assumed to drain by gravity to the lower tubes in a continuous, laminar sheet. In reality, depending on the spacing-to-diameter ratio of the tubes and depending on whether they are arrange in a staggered in-line configuration, the condensate form one tube may not fall on the tube directly below it but instead may flow sideways Figure 1.3 b. Also, it is well known experimentally that condensate does not drain from a horizontal tube in a continuous sheet but in discrete droplets along the tube axis. When these droplets strike the lower tube, considerable splashing can occur figure 1.3 c, causing ripple and turbulence in the condensate film. Perhaps most important of all, large vapor velocities can create significant shear forces on the condensate, stripping it away, independent of gravity Figure 1.3 d.

1.4.2.1 Effect of condensate inundation

In the absence of vapor velocity, as condensate flows by gravity onto lower tubes in a bundle, the flow of condensate from one tube to another thickness the condensate layer around the lower tubes and heat transfer coefficient should decrease.

Nusselt[3] analyzed film condensation on a vertical in-line column of horizontal tubes. He assumed that all the condensate from a given tube drains as a continuous laminar sheet directly onto the top of the tube below it. With this assumption, together with the $(Tsat - Tw)$ remaining the same for all tubes, he showed that the average coefficient for a vertical column of N tubes, compared with the coefficient for the first tube (i.e., the top tube in the row), is

$$\frac{h_{m,N}}{h_1} = N^{-1/4} \tag{1.10}$$

In Equation 1.10, h_1 is calculated using Equation 1.2. In terms of the local coefficient for the Nth tube, the Nusselt theory gives:

$$\frac{h_N}{h_1} = N^{3/4} - (N-1)^{3/4} \tag{1.11}$$

Kern[6] proposed a conservative relationship:

$$\frac{h_{m,N}}{h_1} = N^{-1/6} \tag{1.12}$$

Or in terms of the local value:

$$\frac{h_N}{h_1} = N^{5/6} - (N-1)^{5/6} \tag{1.13}$$

Eissenberg[6] experimentally investigated the effects of condensate inundation by using a staggered tube bundle by considering side-drainage model as in figure 1.3 b. An expression is given by him:

$$\frac{h_{m,N}}{h_1} = 0.60 + 0.42N^{-1/4} \tag{1.14}$$

Numerous experimental measurements have been made in studying the effect of condensate inundation. The data, however, are very scattered. As a result, it is not too surprising that is no successful theoretical model today that can predict accurately the effect of condensate inundation on the condensation performance for a variety of operating conditions. For design purposes, the Kern expressions either Equation 1.12 or 1.13 are conservative, and have been recommended by Butterworth.

1.4.2.2 Effect of vapor shear

In the tube bundles, the influence of vapor shear has been measured by Kutateladze et al.[5], found that there was little difference between the downward-flow and horizontal-flow data obtained, but the upward-flow data were as much as 50% lower in the range $0.1<F<0.5$. What was arrived at is the following empirical expression that correlated the downward flow and horizontal flow data reasonably well:

$$\frac{Nu_m}{\tilde{Re}^{1/4}} = 0.96F^{1/5} \tag{1.15}$$

For $0.03<F<600$. Cavillini et al.[5] compared their data with the prediction of Shekriladze and Gomelauri, Equation 1.6 and found the prediction to be conservative.

In a tube bundle, it is not clear which local velocity should be used to calculate the vapor shear effects. Butterworth pointed out that the use of the maximum cross sectional area gives a conservative prediction. Shklover and Buevich[5] have been used the mean local velocity through the bundle. They calculated this velocity based on a mean flow width that is given by:

$$w = \frac{p_L p_t - \frac{\pi d^2}{4}}{p_L} \tag{1.16}$$

Where $p_L$and p_t are the tube pitches (i.e., centerline-to-centerline distance) in the longitudinal and transverse direction, respectively.

1.4.2.3 Combined effect of inundation and vapor shear

Initially, the effects of inundation and vapor shear were treated separately. The combined average heat transfer coefficient for condensation in a tube bundle was written as:

$$h_{m,N} = h_1 C_N C_{ug} \tag{1.17}$$

Where $h1$ represents the average coefficient for a single tube form Nusselt theory, Equation 1.2 and C_N and C_{ug} are correction factors to account for inundation and vapor shear, respectively.

However, in a tube bundle, a strong interaction exists between vapor shear and condensate inundation: and local heat transfer coefficients are very difficult to predict.. Butterworth proposed a relationship for the local heat transfer coefficient in the Nth tube row that separates out the effects of vapor shear and condensate inundation. A slightly modified form of his equation is

$$h_N = \left[\frac{1}{2}h_{sh}^2 + \left(\frac{1}{4}h_{sh}^4 + h_1^4\right)^{1/2}\right]^{1/2} \times \left[N^{5/6} - (N-1)^{5/6}\right] \tag{1.18}$$

Where h_{sh} is from Equation 1.8:

$$h_{sh} = 0.59 \frac{k_l}{d_1} \tilde{Re}^{1/2}$$

and h_1 is obtained using Equation 1.2.

Mcnaught[6] has suggested that shell-side condensation may be treated as two-phase forced convection. He therefore proposed the following relationship for the local coefficient for the Nth tube raw:

$$\left(h_{sh}^2 + h_G^2\right)^{1/2} \tag{1.19}$$

where h_G is given by Equation 1.13

$$h_G = h_1 \left[N^{5/6} - (N-1)^{5/6}\right]$$

and h_{sh} is given as:

$$h_{sh} = 1.26 \left[\frac{1}{X_{tt}}\right]^{0.78} h_l \tag{1.20}$$

In Equation (1.20), X_{tt} is the Lockhart-Martinelli parameter, define as:

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_g}\right)^{0.1} \tag{1.21}$$

The correlation includes the effect of condensate inundation. McNaught found that Equations (1.19) and (1.20) correlated 90% of the steam data within $\pm 25\%$.

1.5 Optimization

There are several optimization methods to solve design problem for heat exchangers. Some Investigators have used different techniques based on simulated annealing, global sensitivity analysis, harmony search algorithm, partical swarm optimization method, differential evolution strategies, genetic algorithm and traditional mathematical optimization algorithms. These optimization techniques were used for various objectives like minimum entropy generation, and minimum cost of shell and tube heat exchanger. Some of these studies focus mainly on a single geometrical parameter like optimum baffle spacing and some others try to optimize a variety of geometrical and operational parameters of the shell and tube heat exchanger. Determination of the most influential parameters from a set of the design parameters can greatly affect the performance of the optimization process. In surface condenser, optimization of design is possible using different design parameters like shell diameter, tube diameter, tube thickness, tube materials, number of tube passes, tube layout angle, and so on.

1.6 Organization of report

The Report has been organized in six chapters.

Introduction, theory related to steam surface condenser, its classification, need, importance, condensation and various optimization techniques is presented in Chapter 1. Chapter 2 presents review of status which includes discussion on seven research papers about the heat exchanger and its optimization. Motivation for the present work, Definition of the Problem, Objectives and Scope of the present work is also presented in the same chapter. A discussion on Genetic Algorithm as optimization tool for condenser design is presented in Chapter 3. Chapter 4 represents the Mathematical models used for design and optimization of steam surface condenser. The flow chart of the in house code, prepared for design and optimization, can be found here. Chapter 5 presents result and discussion. Comparison of results obtained from in house code for design and optimization of steam surface condenser with results obtained from HTRI and Siemens Ltd Baroda can be found here. Conclusion and future scope of project is presented in Chapter 6.

Chapter 2

Literature Review

Numbers of various research papers are studied of optimum temperature, optimum exergy destructions, optimum entropy and optimum design variable configurations for optimum heat transfer area, optimum pumping power and optimum cost too. These optimization need some nontraditional techniques viz. differential evolution (DE), partical swarm optimization (PSO), genetic algorithm (GA), global sensitivity analysis (GSA) or harmony search algorithm (HSA) for better solution. In first paper, DE method is explained for heat exchanger optimization. Second paper is about optimization of exergy destructions to industrial condenser. Third paper is about entropy minimization of heat exchanger using GA. Forth paper is regarding about the use of bell method for heat transfer coefficient of shellside with GA. Fifth Paper is about the use of global sensitivity analysis (GSA) and harmony search algorithm (HSA) for design optimization of shell and tube heat exchangers and sixth for particle swarm optimization (PSO) technique, for design optimization of shell-and-tube heat exchangers from economic view point.

2.1 Research papers

2.1.1 Differential evolution strategies for optimal design of shell-and-tube heat exchangers

Babu B V et al.[7] applied differential evolution (DE) optimization method and its various strategies for the optimal design of shell-and-tube heat exchangers. Main objective in the heat exchanger design is the estimation of the minimum heat transfer area required for a given heat duty, as it governs the overall cost of the heat exchanger. They have been successfully applied with different strategies for 1,61,280 design configurations using Bell's method to find the heat transfer area.

They used seven design variables in DE for heat exchanger design optimization. In which the design variable $x_1$takes 12 values for tube diameter in the range of 6.35 mm to 63.5 mm. x_2 represents the tube pitch either square or triangular taking 2 values represented by 1 and 2. x_3 takes the shell head types: floating head, fixed tube sheet, U tube, and pull through floating head represented by the numbers 1, 2, 3, and 4, respectively. x4 takes number of tube passes 1, 2, 4, 6, and 8 represented by numbers from 1 to 5. The variable x_5 takes 8 values of the various tube lengths in the range 152.4 mm to 609.6 mm. x_6 takes six values for the variable baffle spacing, in the range 0.2–0.45 times the shell diameter (0.2, 0.25, 0.3, 0.35, 0.4, and 0.45). x_7 takes seven values for the baffle cut in the range 15–45% (0.15, 0.2, 0.25, 0.3, 0.35, 0.4, and 0.45). They found the population-based algorithm DE provides significant improvement in the optimal designs, by achieving the global optimum, compared to the traditional designs.

2.1.2 Optimum temperatures in a shell and tube condenser with respect to exergy

Haseli Y et al.[8] focused on evaluation of the optimum cooling water temperature during condensation of saturated water vapor within a shell and tube condenser, through minimization of exergy destruction.

The relevant exergy destruction was mathematically derived and expressed as a function of operating temperatures and mass flow rates of both vapor and coolant. Then, they define the optimization problem subject to condensation of the entire vapor mass flow. It was solved based on the sequential quadratic programming (SQP) method. The optimization results obtained at two different condensation temperatures of 46°C and 54°C for an industrial condenser. They studied case study for a condenser having the steam mass flow rate is 1 kg/s and increased the condensation temperature from 46 C to 54 C, found the optimal upstream coolant temperature which increased from 16.78°C to 25.17°C. Also, assumed an ambient temperature of 15°C, and got the exergy destruction which decreased from 172.5 kW to 164.6 kW.

They proved that increase in the steam mass flow rate may result in a lower optimum cooling water temperature, which consequently leads to lower exergy efficiency. Also, the optimal exergy destruction increases at higher rates of the steam mass flow. Therefore the exergy destruction and exergy efficiency of condensation of vapor in a shell and tube condenser can be formulate and they can be expressed as functions of several operating parameters, such as the inlet and outlet cooling water and condensation temperatures.

2.1.3 Optimization design of shell and tube heat exchanger by entropy generation minimization and genetic algorithm

Guo Jiangfeng et al.[9] optimized a shell and tube heat exchanger design and demonstrate the dimensionless entropy generation rate by scaling the entropy generation on the ratio of the heat transfer rate to the inlet temperature of cold fluid which employed as the objective function. Also, some geometrical parameters of the shell and tube heat exchanger were taken as the design variables and the genetic algorithm was applied to solve the associated optimization problem. They have successfully applied genetic algorithm (GA) for optimization. It was shown that for the case that the heat duty is given, not only can the optimization design increase the heat exchanger effectiveness significantly, but also decrease the pumping power dramatically.

They have presented three optimization design examples. In the first example five design variables (tube outer diameter, number of tubes, ratio of the baffle spacing to shell diameter, Central angle of baffle cut, Outlet temperature of the cold fluid in the heat exchanger) were selected for a given heat load. It was shown that the optimization design process can remarkably increase the heat transfer effectiveness and decrease the pressure drop. In the second example, shown that the entropy generation number defined by Bejan suffers from the 'entropy generation paradox', while the modified entropy generation number can avoid such a paradox. Therefore the modified entropy generation number is preferable in the heat exchanger optimization design applications. They attempt to use the modified entropy generation number as the objective function in the heat exchanger optimization design. In the third example, the heat transfer area was fixed which obtained by the traditional heat exchanger design method, then the tube and baffle geometrical parameters are selected as the design variables and thus allowed to change in the optimization design process.

2.1.4 Use of genetic algorithms (GAs) for the optimal design of shell-and-tube heat exchangers

Ponce-Ortega J M et al.[10] used the Bell–Delaware method of the shell-side flow with no simplifications for shell and tube heat exchanger. They applied genetic algorithm (GA) for optimization procedure. The use of GA together with the Bell–Delaware method allowed by them for several design factors, and later subjected to a rating test, to be calculated as part of the optimum solution. Also, the objective function can accommodate any type of information available for the cost of heat exchanger having highly non-linear functions that arise from a detailed cost model for a heat exchanger can be handled using genetic algorithm. They studied that genetic algorithms provide better expectations to detect global optimum solutions than gradient methods, in addition to being more robust for the solution of complex problems.

Table 2.1: Result obtained by Ponce-Ortega J M et al. for the optimization of shell and tube heat exchanger

Concept	Mizutani et al.	Ponce-Ortega et al. using GA	
		Design A	Design B
$A(m^2)$	202	242.88	161.34
$h_t(W/m^2K)$	6,480.00	1,628.94	4,493.71
$h_s\,(W/m^2K)$	1,829.00	2,991.26	2,003.71
$V_t\,(m/s)$	-	0.83	1.00
$v_s\,(m/s)$	-	0.37	0.40
$U\,(W/m^2K)$	860.00	714.51	873.62
Number of tubes	832	653	739
Tubes arrangement	Square	Triangular	Square
Number of tube-passes	2	6	1
$D_{ti}\,(mm)$	12.60	22.918	10.92
$D_t\,(mm)$	15.90	25.40	12.70
Number of baffles	8	8	13
Heat kind	Fixed	Floating pull	Floating pull
Hot fluid allocation	Shell	Tube	Shell
$F_t^{\star}\triangle T_{LM}$	24.90	25.01	30.79
$D_s\,(mm)$	0.687	1.105	0.639
Total tube length (m)	4.88	4.88	5.602
Baffle spacing (m)	0.542	0.516	0.391
Baffle cut (%	25	25	31.85
$\triangle p_t\,(Pa)$	22,676.00	10,981.30	7,748.18
$\triangle p_s\,(Pa)$	7,494.00	4,714.28	6,828.43
Pumping cost $(\$/yr)$	2,424.00	960.36	1,033.98
Area cost $(\$/yr)$	2,826.00	3,142.59	2,468.77
Total annual cost $(\$/yr)$	5,250.00	4,102.95	3,502.75

The examples analyzed by them shown that genetic algorithms provide a valuable tool for the optimal design of heat exchangers.

They demonstrated three examples on design optimization of shell and tube heat exchanger using GA. Out of three, one example is shown here:

The GA was used for this problem without the constraints in tube length and baffle cut imposed by Mizutani et al.[11]. The results are reported as Design B in table 2.1.One can notice a significant reduction in the total area required by the exchanger. This is the result of the number of passes being reduced to one, and of smaller tube diameters being selected. This arrangement produces higher stream velocities with better heat transfer coefficients, which provide a smaller area. Another issue worth of mention is that the relationship L_{tt}/D_s is higher than for the other two designs. Design B has a total annual cost 49.88% lower than the one obtained by Mizutani et al., and 17.13% lower than Design A.

2.1.5 Design optimization of shell and tube heat exchangers using global sensitivity analysis and harmony search algorithm

Fesanghary M et al.[12] studied the use of global sensitivity analysis (GSA) and harmony search algorithm (HSA) for design optimization of shell and tube heat exchangers (STHX) from the economic point of view. They used GSA for reduce the size of the optimization problem using non-influential geometrical parameters which have the least effect on total cost of STHXs. Then, they applied the HSA which is a meta-heuristic based algorithm to optimize the influential geometrical parameters.

They have taken the capital cost of a STHX as an objective function to optimal design of a STHX which is capable of accomplishing the prescribed thermal duty with minimum combined investment and operating cost.

2.1.6 A new design approach for shell-and-tube heat exchangers using genetic algorithms from economic point of view

Resat Selbas(2005),[13] et al. studied the estimation of minimum heat transfer area required for a given heat duty of heat exchanger. They used genetic algorithms (GA) successfully for the optimal design of shell-and-tube heat exchanger for I type and U type. They have taken 47040 design combination for heat exchanger using different shell diameter, different tube pass, different tube diameter, different layout, and no of baffle choice and use GA for find optimum design based on minimum heat transfer area to reduce the cost of heat exchanger for a given heat duty. The LMTD method was used to determine the heat transfer area for a given design configuration.

They considerd following objective function for optimization,

$$C_{tot} = C_i + C_o \tag{2.1}$$

where,

$$C_i = A \times fd \times C_1 \tag{2.2}$$

and

$$C_o = \frac{(E_t + E_s) \times H \times ec \times C_1 \times C_2}{3600 \times 1000} \tag{2.3}$$

They considered a case for design a heat exchanger of duty: 20 kg/s of water leaves from the base of a system at 75°C and is to be cooled to 50°C by exchange with water coming from city pipeline at 20°C and exit temperature of the water is assumed to be 35°C. For the economic calculations, fd, unit cost of heat exchanger per area was assumed to be 25 \$/m2, fe, electric cost, 0.1 \$/kWh for Turkey,H, annual operating period, 17,280,000 s/year, I, interest rate, 60%, ec, rate of increase of energy costs, 50%, tp, total operating period, 10 years. The pump efficiencies were assumed to be 70%.

The GA was applied for optimization using the following solution scheme as shown in Figure 2.1 and got appreciable and faster result for cost vs heat transfer area as shown in Figure 2.2.

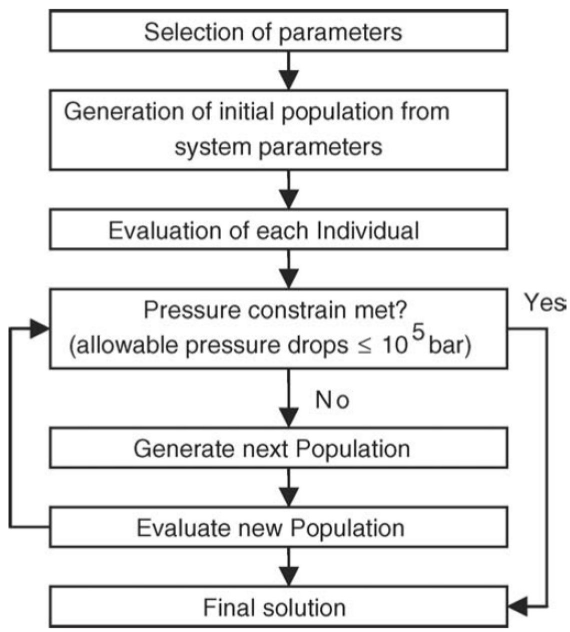


Figure 2.1: Solution method scheme

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

Patel V K et al.[14] explored the use of a non-traditional optimization technique; called particle swarm optimization (PSO), for design optimization of shell-and-tube heat exchangers from economic view point. Minimization of total annual cost was considered as an objective function of heat exchanger. They have selected three design variables such as shell internal diameter, outer tube diameter and baffle spacing for optimization. Also, two tube layouts viz. triangle and square are considered for optimization. Four different case studies are presented to demonstrate the ef-

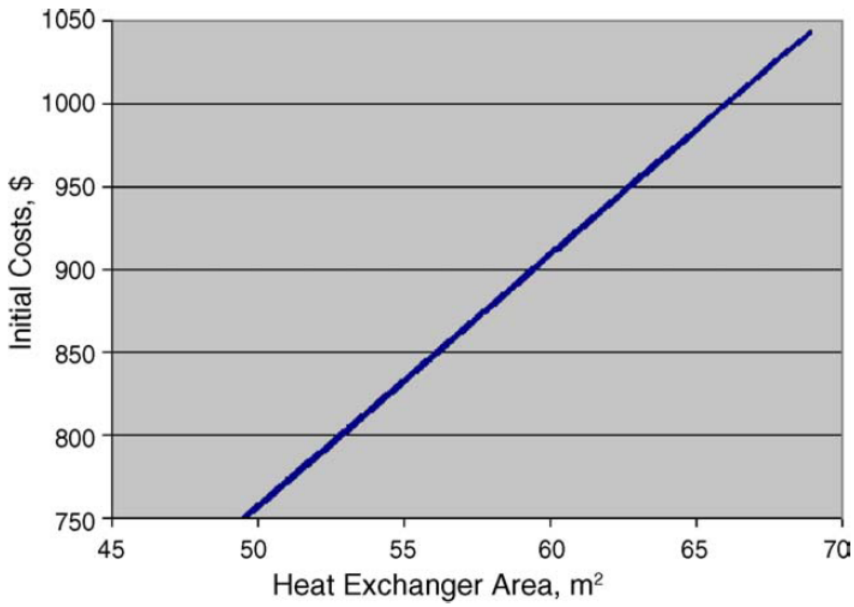


Figure 2.2: Variation of initial costs with area

fectiveness and accuracy of the proposed algorithm. They got appreciable results using partical swarm optimization technique for case studies and compared same with genetic algorithm using following objective function.

$$C_{tot} = C_i + C_{od} \tag{2.4}$$

Total cost C_{tot} is taken as the objective function, which includes capital investment (C_i) energy cost (C_e), annual operating cost (C_o) and total discounted operating cost (C_{od}). where,

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

where, $a_1 = 8000$, $a_2 = 259.2$, $a_3 = 0.93$ for exchanger made with stainless steel for both shell and tubes. A is heat transfer area.

The total discounted operating cost related to pumping power to overcome friction losses is computed from the following equation,

$$C_o = PC_eH$$

$$C_{od} = \sum_{x=1}^{ny} \frac{C_o}{(1+IR)^x}$$

where, ny is the life of condenser in year.

2.2 Motivation

Power sector is a predominant factor for the growth of country. At present so many power plants are built due to increasing the demand of energy. Steam surface condenser is a unit used in power plant to increase the efficiency of turbine and hence increases the net output of the plant. In the design of steam surface condenser, proper correlations must be used to calculate the overall heat transfer coefficient, area, pressure drop, and cost of condenser. For industrial application design should be optimized. When designing a condenser so many design combination comes to picture, but best design is what give a better design with low cost. At present various commercial programs are available for designing and rating of shell and tube heat exchanger such as HTRI, HTFS, THERM and CC-THERM. These required various optimization strategies to consist any optimization of design that need from industries point of view.

2.3 Problem definition

Problem definition of the project is:

Steam surface condenser design based on cost optimization using genetic algorithm

2.4 Objective

2.4.1 General Objective

To design the steam surface condenser using appropriate correlation for rating and then optimize the cost of design with nontraditional optimization tool.

2.4.2 Specific Objective

- To design the steam surface condenser and rate the same using Nusselt's correlation.
- The rating is compare with HTRI (Heat transfer Research Institute) software's result and HEI (Heat Exchanger Institute)'s results.
- To optimize the capital and total cost of surface condenser design.
- To implement genetic algorithm (GA) for optimization of the above cost and compared the results between them.
- Also, find out the GA's results for varying the search space of design variables and compared the same.

2.5 Scope

The scope of the project is the design of steam surface condenser which optimize through genetic algorithm (GA). GA is helpful to optimize the cost by selecting optimum design variables such as shell diameter, tube diameter, tube thickness and tube material with wide range to solve optimum heat transfer area and pumping power then to find out the optimum cost of condenser as based on capital cost and total cost.

Chapter 3

Genetic Algorithm (GA)

Introduction

There are to many techniques for global optimization (i.e. finding the global minimum or maximum of some complex functional). Genetic algorithm (GA) acts on the principles of natural genetics and natural selection. It is computerized search and optimization algorithm. Professor John Holland of the University of Michigan, Ann Arbor envisaged his seminal work (Holland, 1975). Thereafter, a number of his students and other researchers have contributed to developing this field. To date, most of the GA studies are available through a few books (Davis, 1991; Goldberg, 1989; Holland, 1975; Michalewicz, 1992) and through a number of international conferences proceeding (Belew and Booker, 1991; Forrest, 1993; Grefenstette, 1985, 1987; Rawlins, 1991; Forrest, 1993; Grefenstette, 1985, 1987; Rawlins, 1991; Schaffer, 1989, Whitley, 1993). An extensive list of GA-related papers is referenced elsewhere (Goldberg, et. al, 1992).

3.1 Working principles

To illustrate the working principles of GAs, an unconstrained problem is considered. Let us consider the following maximization problem:

$$Maximize\ f(x), \quad x_i^{(L)} \leq x_i \leq x_i^{(U)}, \quad i = 1, 2, ..., N.$$

where, $x_i^{(L)}$ and $x_i^{(U)}$ are the lower and upper bound the variable x_i can take. Although a maximization problem consider here, a minimization problem can also be handled using GAs. The working of GAs is completed by performing the following tasks:

3.1.1 Coding

For above problem, variable x_i's are first coded in string structures. Binary-coded strings having 1's and 0's are mostly used. The length of the string is usually determined according to the desired result accuracy. For example, if four bits are used to code each variable in a two-variable function optimization problem, the strings (0000 0000) and (1111 1111) would represent the points

$$\left(x_1^{(L)}, x_2^{(L)}\right)^T \qquad \left(x_1^{(L)}, x_2^{(L)}\right)^T,$$

respectively, because the substrings (0000) and (1111) have the minimum and the maximum decoded values. Any other eight-bit string can be found to represent a point in the search space according to affixed mapping rule. Normally, the following mapping rule is used:

$$x_i \;=\; x_i^{(L)} + \frac{x_i^{(U)} - x_i^{(L)}}{2^{l_i} - 1} \qquad decoded\,values\ (s_i) \tag{3.1}$$

In the above equation, the variable x_i is coded in a substring s_i of length l_i. The coded value of a binary substring s_i is calculated as $\sum_{i=0}^{l-1} 2^i s_i$, where $s_i \in (0, 1)$ and

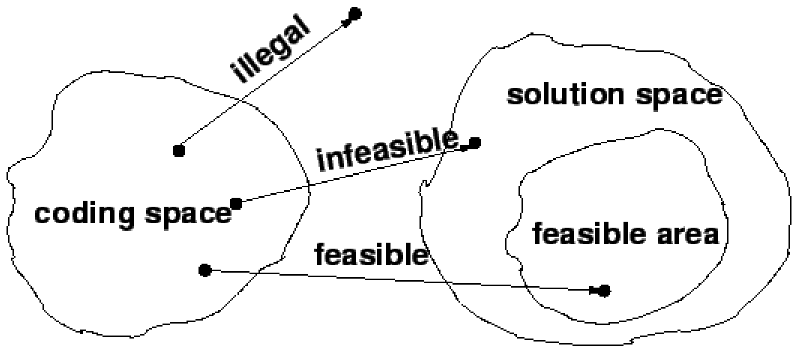


Figure 3.1: Coding in GA

the string s is represented as $(s_{l-1}s_{l-2}...s_2s_1s_0)$. For example, a four-bit string (0111) has a decoded value equal to $(1)\,2^0 + (1)\,2^1 + (1)\,2^2 + (0)\,2^3$ or 7.

Here it is mention that with four bits to code each variable, there are only 2^4or 16 distinct substrings possible, because each bit position can take a value either 0 or 1. The accuracy obtained with a four bit coding is only approximately 1/16th of the search space. But as the string length is increased by one, the obtainable accuracy increases exponentially to 1/32th of the search space. It is not necessary to code all variables in equal substring length. The length of substring represent a variable depends on the desired accuracy in that variable. With this concept, generally we say that with an l_i bit coding for a variable, the obtainable accuracy in that variable is approximately $\left(x_i^{(U)} - x_i^{(L)}\right)/2^{l_i}$. once the coding of the variables has been done, the corresponding point $x = (x_1, x_2, ...x_N)^T$ can be found using Equation 3.1. Thereafter, the function value at the point x can also be calculated by substituting x in the given objective function $f\,(x)$.

3.1.2 Fitness function

GAs mimic the "survival of the fittest" principle of the nature to make a search process. Therefore, GAs are naturally suitable for maximization problems. So minimization problems are suitably transformed into maximization problems by using transformation. For maximization problems, the fitness function ($\digamma(x)$) can be considered to be the same as the objective function or $\digamma(x) = f(x)$. For minimization problem, the fitness function is an equivalent maximization problem chosen that the optimum point remains unchanged by transformation. For example of such transformation is:

$$\digamma(x) = 1/(1 + f(x)) \tag{3.2}$$

This transformation does not alter the point of minimum, but converts a minimization problem to an equivalent maximization problem.

3.1.3 GA operators

The operation of GAs starts with a population of random strings representing design variables. Thereafter, each string is evaluated to find the fitness value. The population is then operated by three main operators which are:

1. Reproduction
2. Crossover
3. Mutation

These operators create a new population of points. The new population is further evaluated and tested for termination criteria. If the termination criteria are not met, the population is continued until the termination criterion is met. One cycle of these operations and the subsequent evaluation procedure is known as a *generation.*

3.1.3.1 Reproduction

Reproduction (or selection) is an operator that makes more copies of better strings in a new population. Reproduction is the first operator applied on a population.

It selects good strings in a population and forms mating pool. That why the reproduction is sometimes known as the *selection operator*. To sustain the generation of new population, the reproduction of the individuals in the current population is necessary. For better individuals, these should be from the fittest individuals of the previous population. There exist number of reproduction operators in GA, but the essential in all of them is that the above average strings are picked from the current population and their multiple copies are inserted in the mating pool in a probabilistic manner.

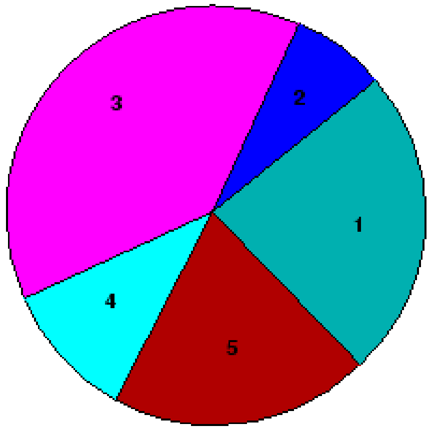
Roulette-wheel selection

The commonly used reproduction operator is the proportionate reproduction operator where a string is selected for the mating pool with a probability proportional to its fitness. Thus, the i-th string in the population is selected with a probability proportional to $\mathcal{F}_i$. Since the population size is usually kept fixed in a simple GA, the sum of the probability of each string being selected for the mating pool must be one. Therefore, the probability for selecting the i-th string is:

$$p_i = \frac{\mathcal{F}_i}{\sum_{j=1}^{n} \mathcal{F}_j} \tag{3.3}$$

where n is the population size. This selection scheme is also imagine as a roulette-wheel with its circumference marked for each string proportionate to the string's fitness. The roulette wheel is spun n times, each time selecting an instance of the string chosen by the roulette-wheel pointer. Since the circumference of the wheel is marked according to a string's fitness, this roulette-wheel mechanism is expected to make $\mathcal{F}_i/\overline{\mathcal{F}}$ copies of the i-th string in the mating pool. The average fitness of the population is calculated as

$$\overline{\mathcal{F}} = \sum_{i=0}^{l-1} \mathcal{F}_i/n. \tag{3.4}$$



Population	Fitness
1	25.0
2	5.0
3	40.0
4	10.0
5	20.0

Figure 3.2: Roulette-wheel

Figure3.2[15] shows a roulette-wheel for each individuals having different fitness values. Since the third individual has fitness value than any other, it is expected that the roulette-wheel selection will choose the third individual more than any other individual. This roulette-wheel selection scheme can be simulated easily. Using the fitness value f_i of all strings, the probability of selecting a string p_i can be calculated. Thereafter, the cumulative probability $p_n$of each string being copied can be calculated by adding the individual probabilities from the top of the list. Thus, the bottom-most string in the population should have a cumulative probability values from $p_{i-1}$to p_i. The first string represents the cumulative values from zero to p_1. Thus, the cumulative probability of any string lies between 0 and 1. In order to choose n strings, n random numbers are between 0 and 1 are created at random. Thus, a string that represents the chosen random number in the cumulative probability range (calcu;ated from fitness values) for the string is chosen to the mating pool. This way the string with a higher fitness values and therefore has a higher of being copied into the mating pool. On the other hand, a string with a smaller fitness value represents a smaller range in cumulative probability values and has a smaller probability of being copied into the mating pool.

3.1.3.2 Crossover

A crossover operator is used to recombine two strings to get a better string. In crossover operation, recombination process creates different individuals in the successive generations by combining material from two individuals of the previous generation. In reproduction, good strings in a population are probabilistically assigned a large number of copies and a mating pool is formed. It is important to note that no new strings are formed in the reproduction phase. In the crossover operator, new strings are created by exchanging information among strings of the mating pool.

The two strings participating in the crossover operation are known as parent strings and the resulting strings are known as children strings. It is intuitive from this construction that good sub-strings from parent strings can be combined to form a better child string, if an approximate site is chosen. With a random site, the children strings produced may or may not have a combination of good sub-strings from parent strings, depending on whether or not the crossing site falls in the approximate place. But this is not a matter of serious concern, because if good strings are created by crossover, there will be more copies of them in the next mating pool generated by crossover. It is clear from this discussion that the effect of crossover may be detrimental or beneficial. Thus, in order to preserve some of the good strings that are already present in the mating pool, all strings in the mating pool are not used in crossover. When a crossover probability, defined here as p_c is used, only $100p_c$ percent strings in the population are used in the crossover operation and $100\,(1-p_c)$ percent of the population remains as they are in the current population. A crossover operation is mainly responsible for the search of new strings even though mutation operator is also used for this purpose sparingly.

Two crossover operations exist in the literature. One site and two site crossover are the most common adopted. In most crossover operators, two strings are picked from the mating pool at random and some portions of the strings are exchanged

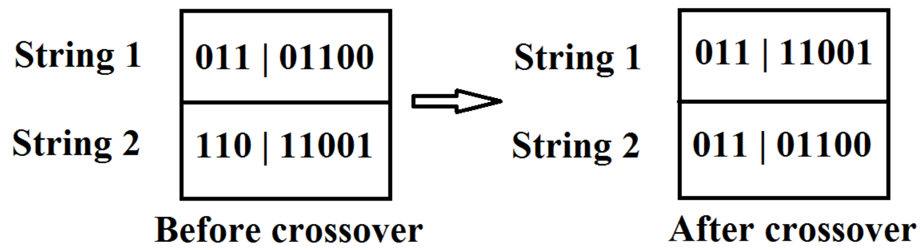


Figure 3.3: One site crossover

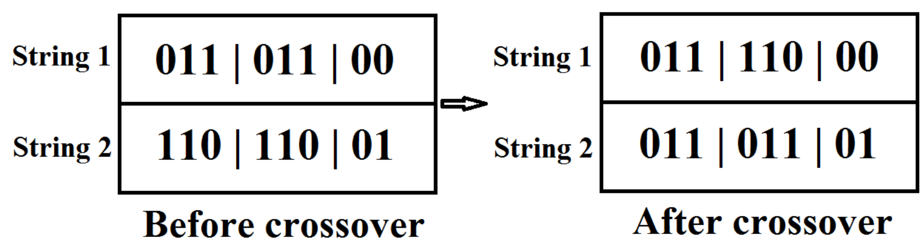


Figure 3.4: Two site crossover

between the strings. Crossover operation is done at string level by randomly selecting two strings for crossover operations. A one site crossover operator is performed by randomly choosing a crossing site along and the string by exchanging all bits in the right side of the crossing site as shown in figure 3.3.

In one site crossover, a crossover site is selected randomly (shown as vertical lines). The portion right of the selected site of these two strings is exchanged to form a new pair of strings. The new strings are thus combinations of the old strings. Two site crossover is a variation of the crossover, except that two crossover sites are chosen and the bits between the sites are exchanged as shown in figure 3.4.

One site crossover is more suitable when string length is small while two site crossover is suitable for large strings. The underlying objective of crossover is to exchange information between strings to get a string that is possibly better than the parents.

3.1.3.3 Mutation

Mutation adds new information in a random way to the genetic search process and ultimately helps to avoid getting trapped at local optima. It is an operator that introduces diversity in the population whenever the population tends to become homogeneous due to repeated use of reproduction and crossover operators. Mutation may cause the chromosomes of individuals to be different from those of their parent individuals.

Mutation in a way is the process of randomly disturbing genetic information. They operate at the bit level; when the bits are being copied from the current string to the other string, there is probability that each bit may become mutated. This probability is usually a quite small value, called as mutation probability p_m. A coin toss mechanism is employed: if random number between zero and one is less than the mutation probability, then the bit is inverted, so that zero becomes one and one becomes zero. This helps in introducing a bit of diversity to the population by scattering the occasional points. This random scattering would result in better optima, or even modify a part of genetic code that will be beneficial in later operations.

The need for mutation is to create a pint in the neighborhood of the current point, thereby achieving a local search around the current solution. The mutation is also used to maintain diversity in the population. For example, the following population having four-eight bit string may be considered:

0110	1011
0011	1101
0001	0110
0111	1100

It can be noticed that all four strings have a 0 in the left most bit position. If the true optimum solution requires 1 in that position, then neither reproduction nor crossover operator described above will be able to create 1 in that position. The inclusion of mutation introduces probability p_m of turning 0 into 1.

These three operators are simple and straightforward. The reproduction operator selects good strings and the crossover operator recombines good sub-strings from good strings together, hopefully, to create a better sub-string. The mutation operator alters a string locally expecting a better string. Even though none of these claims are guaranteed and/or tested while creating a string, it is expected that if bad strings are created they will be eliminated by the reproduction operator in that next generation and if good strings are created, they will be increasingly emphasized.

Application of these operators on the current population creates a new population. This new population is used to generate subsequent populations and so on, yielding solutions that are closer to the optimum solution. The values of the objective function of the individuals of the new population are again determined by decoding the strings. These values express the fitness of the solutions of the new generations. This completes one cycle of genetic algorithm called a generation. In each generation if the solution is improved, it is stored as the best solution. This is repeated till convergence. The flow chart of genetic algorithm is shown as in AppendixA.1.

3.2 GAs and traditional methods

As seen from the above description of the working principles of GAs, they are radically different from most of the traditional optimization methods. However, fundamental difference is described subsequently. GAs work with a string coding of variables instead of the variables. The advantage of working with a coding of variables is that the coding discrete the search space, even though the function may be continuous. On the other hand, since GAs require only function values at various discrete or discontinuous function can be handled with no extra burden. This allows GAs to be applied to a wide variety of problems. Another advantage is that the GA operators exploit the similarities in string structures to make an effective search. The most striking difference between GAs and many traditional optimization methods is that GAs work with a population of points instead of a single point. Because there is more than one string being processed simultaneously, it is very likely that the expected GA solution may be a global solution. Even though some traditional algorithms are population based, like Box's evolutionary optimization and complex search methods, those methods do not use previously obtained information efficiently. In GAs, previously found good information is emphasized using reproduction operator and propagated adaptively through crossover and mutation operators. Another advantage with a population based search algorithm is that multiple optimal solutions can be captured in the population easily, thereby reducing the effort to use the same algorithm many times.

Genetic algorithms differ from convectional optimization and search procedures in several fundamental ways. It can be summarized as follows:

1. GAs work with a coding of solution set, not the solutions themselves.
2. GAs search from a population of solutions, not a single solution.
3. GAs use payoff information (fitness function), not derivatives or other auxiliary knowledge.
4. GAs use probabilistic transition rules, not deterministic rules.

3.3 Advantages and disadvantages of GA

Advantages:

- No need for knowledge or gradient information about object or energy surface.
- Resistant to becoming trapped in local minima.
- Work well on large-scale optimization problems.

- Discontinuities present on surface have little effect on optimization.
- It can be used on a wide variety of problems.

Disadvantages:

- Trouble finding exact global minimum.
- Require a large number of energy or object function evaluations.
- Setting up configurations is not straightforward.

3.4 Application of GA

Nearly everyone can gain benefits from GAs, once he can encode solutions of a given problem to chromosomes in GA and compare the relative performance (fitness) of solutions. An effective GA representation and meaningful fitness evaluation are the keys of the success in GA applications. . The appeal of GAs comes from thesis simplicity and elegance as robust search algorithms as well as from their power to discover good solutions rapidly for difficult high-dimensional problems. GAs are useful and efficient when

1. The search space is large, complex or poorly understood
2. Domain knowledge is scarce or expert knowledge is difficult to encode to narrow the search space
3. No mathematical analysis is available
4. Traditional search methods fail

The advantage of the GA approach is the ease with which it can handle kinds of constraints and objectives; all such things can be handled as weighted components of the fitness function, making it easy to adapt the GA scheduler to the particular requirements of a very wide range of possible overall objectives.

GAs have been used for problem solving and for modeling. GAs are applied to many scientific, engineering problems, in business and entertainment, including:

- Optimization: GAs have been used in a wide variety of optimization tasks, including numerical optimization, and combinatorial optimization problems such as traveling salesman problem (TSP), circuit design [Louis 1993], job shop scheduling [Goldstein 1991] and video & sound quality optimization.

- Automatic Programming: GAs have been used to evolve computer programs for specific tasks, and to design other computational structures, for example, cellular automata and sorting networks.

- Machine and robot learning: GAs have been used for many machine-learning including classifier systems or symbolic production systems, and to design and control robots.

- Economic models: GAs have been used to model processes of innovation, the development of bidding strategies, and the emergence of economic markets.

- Models of social system: GAs have been to study evolutionary aspects of social systems, such as the evolution of cooperation [Chughtai 1995], the evolution of communication, and trail-following behavior in ants.

Chapter 4

Mathematical Models

Mathematical models are used in design of the steam surface condenser. The flow chart of this design is shown in AppendixA.2.

4.1 Heat transfer

Generally in steam surface condenser, steam expands from turbine exhaust on shell side and it condenses by cooling water in tube side. Therefore, Thermal analysis is done on both tube side and shell side for heat transfer and its area.

According to flow regime with low Reynolds number, the tube side heat transfer coefficient is computed from the following correlation,

$$h_t = \frac{k_t}{d}\left[3.657 + \frac{0.0677\left(Re_t Pr_t\left(d_i/L\right)\right)^{1.33}}{1 + 0.1 Pr_t\left(Re_t\left(d_i/L\right)\right)^{0.3}}\right] \tag{4.1}$$

(if $Re_t < 2300$ [16])

According to flow regime with transition flow, the tube side heat transfer coefficient is computed from Gnielinski correlation,

$$\frac{h_t d}{k_t} = \frac{(f/2)\left(Re_t - 1000\right)Pr_t}{1 + 12.7\left(f/2\right)^{1/2}\left(Pr_t^{2/3} - 1\right)} \tag{4.2}$$

(if $2300 < Re_t < 10,000$ [5])

where,

$$f = (1.58lnRe_t - 3.28)^{-2}$$

According to flow regime with higher Reynolds number, the tube side heat transfer coefficient is computed from Petukhov and Kirillov correlation,

$$\frac{h_t d}{k_t} = \frac{(f/2)\, Re_t Pr_t}{1.07 + 12.7\,(f/2)^{1/2}\left(Pr_t^{2/3} - 1\right)} \tag{4.3}$$

(for $Re_t > 10,000$ [5])

where,

$$f = (1.58lnRe_t - 3.28)^{-2}$$

Re_t is the tube side Reynolds number and given by,

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t} \tag{4.4}$$

Pr_t is the tube side Prandlt number and given by,

$$Prt = \frac{\mu_t cp_l}{k_t} \tag{4.5}$$

Flow velocity for tube side is found by,

$$v_t = \frac{m_t}{(\pi/4)\, d_t^2 \rho_t}\left(\frac{n}{N_t}\right) \tag{4.6}$$

where, N_t is the number of tubes and n is the number of tube passes. Which can be found approximately from the following correlation[5],

$$Nt = 0.875\left(\frac{CTP}{CL}\right)\frac{D_s^2}{(PR)^2\, d_o^2} \tag{4.7}$$

where, D_s is shell diameter, which would contain right number of tubes, N_t of tube diameter d_o. CTP is the tube count calculation constant whose values are suggested,

one-tube pass	CTP=0.93
two-tube pass	CTP=0.9
three-tube pass	CTP=0.85

and, CL is the tube layout constant whose values are suggested,

CL=1.0 for 90° and 45°
CL=0.87 for 30° and 60°

Using Nusselt correlation[3], heat transfer coefficient for a single horizontal tube with assume laminar film condensation is computed from the following correlation,

$$h_s = 0.728 \frac{k_l}{d_o} \left[\frac{\rho_l(\rho_l - \rho_g) g i_{fg} d_o^3}{\mu_l \triangle T_w k_l} \right]^{0.25} \tag{4.8}$$

The temperature difference $\triangle T_w$ is given by,

$$\triangle T_w = T_{sat} - T_w = \triangle T - R_t q"$$

where, $\triangle T$ is the local temperature difference between the streams; and $R_t$is the sum of all other resistance (based on the tube outer diameter); and $q"$ is the local heat flux, which are given by,

$$R_t = \frac{d_o}{d_i} \frac{1}{h_t} + \frac{d_o}{d_i} R_{fi} + \frac{t_w}{k_w} \frac{d_o}{D_m} + R_{fo} \tag{4.9}$$

and

$$q" = U_o \triangle T$$

where, D_m is approximated as:

$$D_m = \frac{d_o - d_i}{ln\,(d_o/d_i)} = \frac{1}{2}\,(d_o + d_i)$$

and t_w is the wall thickness; and U_o is the overall heat transfer coefficient based on the tube outer diameter.

Considering splashing effect on horizontal tube bundle, the average heat transfer coefficient of tube bundle is computed from the following Kern correlation[6],

$$h_{s,N} = h_s N^{-1/6} \tag{4.10}$$

The overall heat transfer coefficient based on the tube outer diameter (U_o) depends on both the shell side and tube side heat transfer coefficients and fouling resistances is given [5],

$$U_o = \frac{1}{R_t + \frac{1}{h_{s,N}}} \tag{4.11}$$

Considering for a condensation, the logarithmic mean temperature difference ($LMTD$) is computed from the following formula[17],

$$LMTD = \frac{T_{c,out} - T_{c,in}}{ln\left[\left(T_{sat} - T_{c,in}\right) / \left(T_{sat} - T_{c,out}\right)\right]} \tag{4.12}$$

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

$$A = \frac{Q}{UFLMTD} \tag{4.13}$$

where, the LMTD correction factor F is found as one for condensation and the heat capacity ratio R is zero [5].

Correlation of the heat transfer rate is given either by condensation or by sensible heat transfer which is computed from following formula,

$$Q = m_s h_{fg} = m_c C_{pc} \left(T_{c,out} - T_{c,in}\right) \tag{4.14}$$

where, $m_s$is the mass flow rate of steam; and $h_{fg}$is the enthalpy of vaporization which is the difference between enthalpy of saturated steam (h_v) and enthalpy of water (h_w). $m_c$is the mass flow rate of cooling water. T_{co} and $T_{ci}$are the temperature of cooling water at inlet and outlet respectively.

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

$$L = \frac{A}{\pi d_o N_t} \tag{4.15}$$

4.2 Pressure drop

In the heat exchanger there is a close physical and economical affinity between pressure drop and heat transfer. This pressure drop is the static fluid pressure which may be expanded to drive the fluid through heat exchanger. Increasing the flow velocity will cause a rise of heat transfer coefficient which results in compact heat exchanger design and lower investment cost. However increase of flow velocity will cause more pressure drop in exchanger which results in additional running cost. For this reason when designing a heat exchanger pressure drop must be considered with heat transfer.

Tube side pressure drop is computed from the following correlation[14],

$$\triangle P_t = \frac{\rho_t v_t^2}{2} \left(\frac{L}{d_i} f_t + p \right) n \tag{4.16}$$

where, p is a constant and its different values are considered by different authors, kern[14] assumed $p = 4$, while Sinnot[14] et al. assumed $p = 2.5$.

For steam surface condenser, only tube side is a running cost of condenser, while shell side steam expands in a condenser from turbine exhaust directly. That's why total pumping power is computed from tube side pressure drop only. This power is computed by following formula,

$$E_{\triangle P} = \frac{1}{\eta} \left(\frac{m_c}{\rho_c} \triangle P_t \right) \tag{4.17}$$

where, η is the efficiency of pump.

Table 4.1: Cost coefficient of condenser

Range	a	b
$A < 9\,m^2$	3135	0.463
$9\,m^2 < A < 90\,m^2$	1957	0.679
$A > 9\,m^2$	1042	0.810

4.3 Objective function

Total cost C_{tot} is taken as the objective function, which includes capital cost $C_c$and operating cost C_o.

$$C_{tot} = C_i + C_o \tag{4.18}$$

The capital cost of the exchanger C_i is computed as a function of heat exchanger surface area A using following formula[12],

$$C_c = aA^b$$

where, a and b are the cost coefficient constants. For Admiralty the value of a and b are shown in Table4.3.

Thus,

$$C_i = C_c \frac{i_R\,(1+i_R)^{TL}}{(1+i_R)^{TL}-1} \tag{4.19}$$

The operating cost of exchanger is computed using following formula,

$$C_{op} = TP \cdot ec \cdot E_{\triangle P}$$

Thus,

$$C_o = C_{op} \frac{i_R\,(1+i_R)^{TL}}{(1+i_R)^{TL}-1} \tag{4.20}$$

TP is the period of the time of operation per year; i_R is the interest rate; ec is the unit cost of the energy; and TL is the technical life of condenser.

The cost of exchanger using materials other than Admiralty are computed using proper material-cost ratio of the materials (cost of material/Cost of Admirelty) for exchanger surface area. Such as material for SS304 the $cost\,of\ the\,exchanger = material - cost\,ratio\,of\ SS304 \times C_i$. The current cost of the materials can be find from London Metal Exchange[18].

Chapter 5

Results and discussion

Results and discussion has been carried out in two sections. One is rating the steam surface condenser and other is on design and optimization of same.

5.1 Rating of steam surface condenser:

There are two separated code developed, one for rating & designing and second for optimization. In thermal design, the heat transfer area is evaluated from the overall heat transfer coefficient. In the present work, the overall heat transfer is computed using Nusselt correlation which is based on empirical correlation. The results obtained from designing and rating code are compared with data for overall heat transfer coefficient obtained from Siemens Ltd. Note that, the data of overall heat transfer coefficient obtained from Siemens Ltd. is actually from Heat Exchanger Institute (HEI)'s experimental data. The results are also compared with HTRI (Heat Transfer Research Institute)'s result.

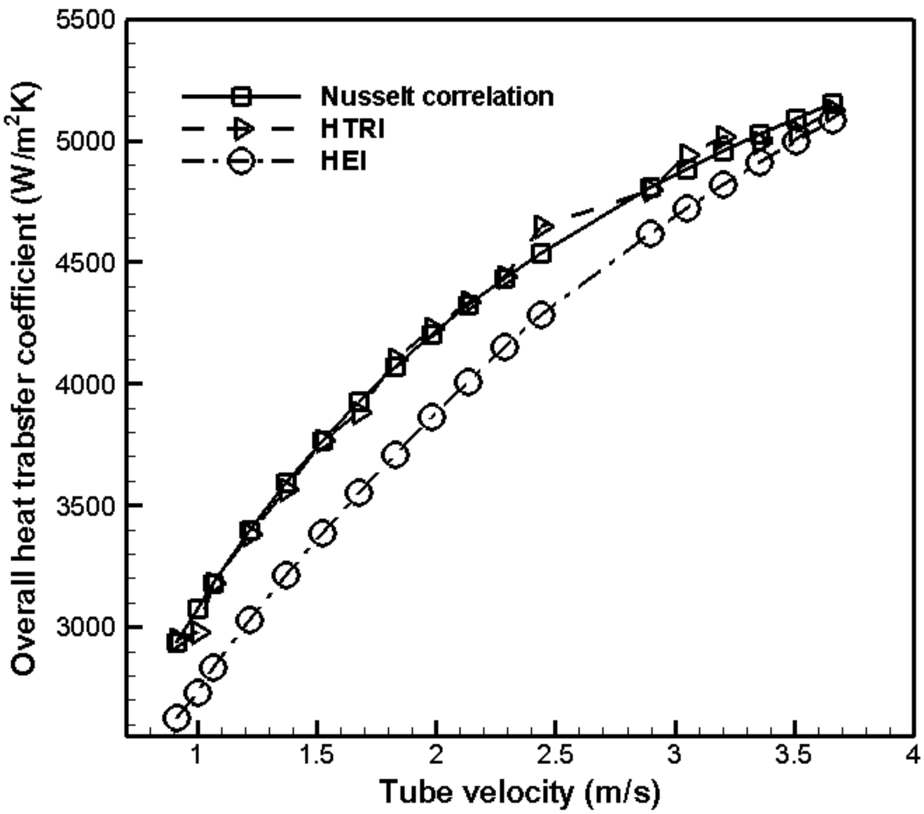


Figure 5.1: Overall Heat transfer coefficient vs velocity at tube at tube outer diameter of 19.05 mm using Admiralty tube material where inlet cooling water at 70°F

The specification use for comparing and validating the design and rating code is as follows.

In a steam condenser the cooling water at 21.11°C (70°F) is enter in Admiralty metal tube. The steam enters on shell side with mass flow rate of 15.9612 kg/s at 0.1 bar pressure and dryness fraction 0.98. For a given shell inner diameter, the tube outer diameters are varied. and computed the tube side velocity. The results are compared taking the shell outer diameter as 1905 mm, with Pitch ratio is 1.25, 90° tube layout, and 0.899 mm tube thickness for one tube pass. Four tube outer diameters (19.05, 25.4, 44.45, and 50.08 mm) are considered for comparisons.

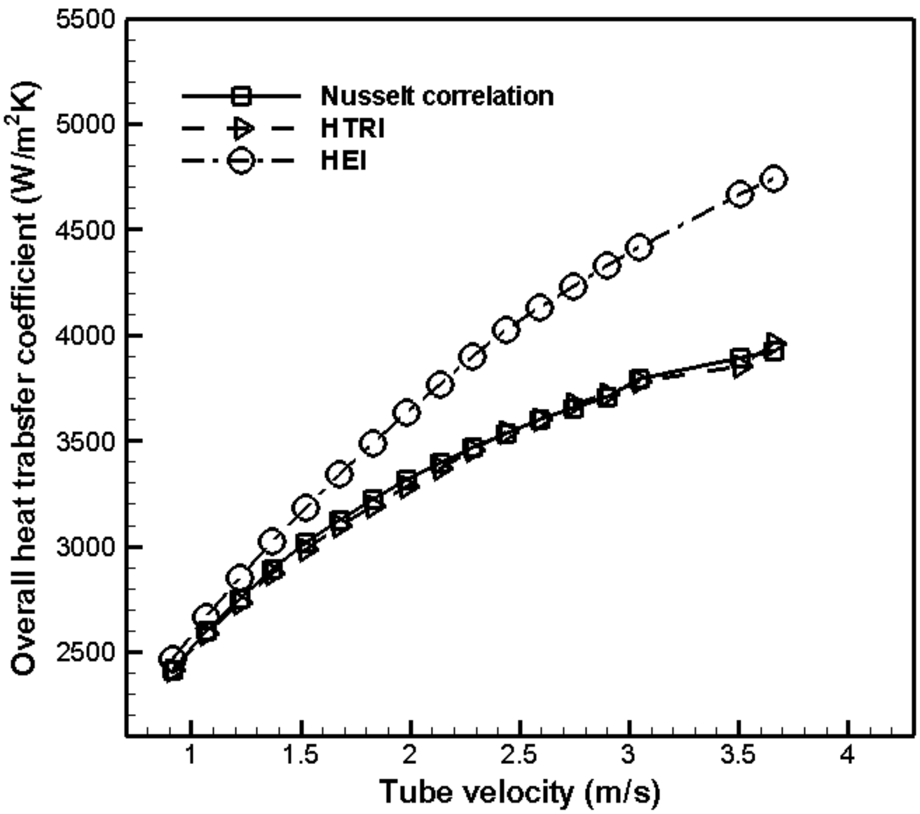


Figure 5.2: Overall Heat transfer coefficient vs velocity at tube outer diameter of 25.4 mm using Admiralty tube material where inlet cooling water at 70°F

The comparisons of results with HEI and HTRI are shown in Figure 5.1 to 5.4. The HTRI result sheets for this analysis to given specifications are shown in Appendix B.1 to B.4.

The results of Program for rating using Nusselt method is well match with HTRI and the maximum deviation found within 3.31%. The results of program are match with HEI's experimental result with maximum deviation of 11.256%, with permitted velocity between 0.9144 m/s to 2.3 m/s. The Nusselt results have values lower than the HEI's experimental values, so the heat transfer area found by Nusselt method is always comes higher than the area found by HEI results.

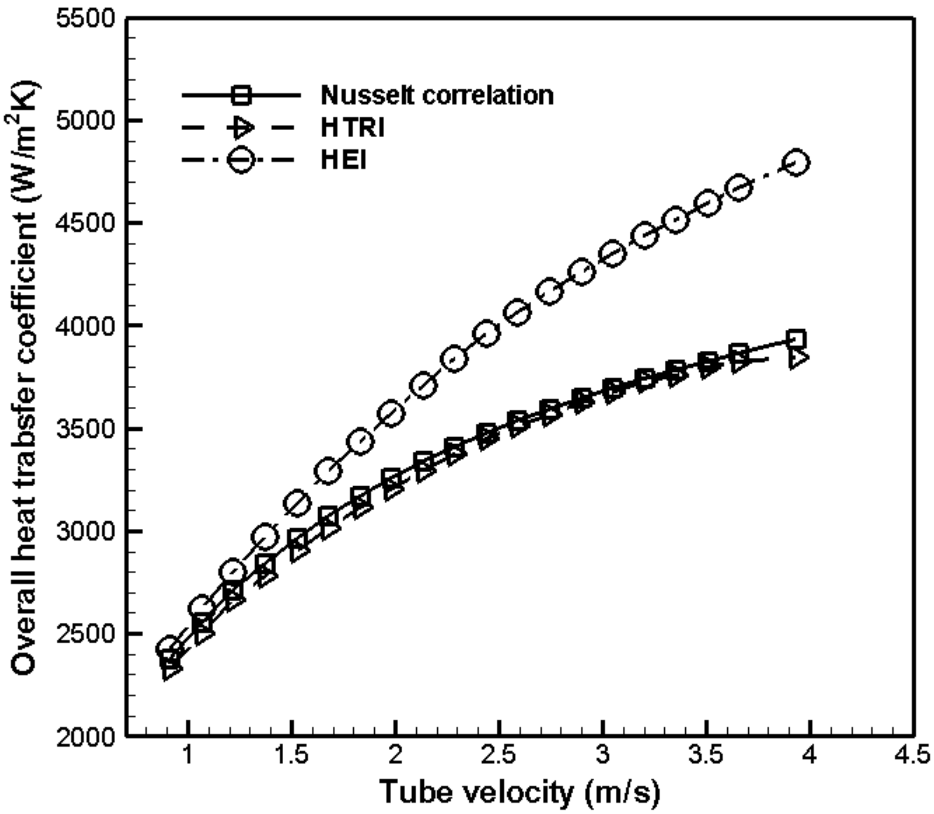


Figure 5.3: Overall Heat transfer coefficient vs velocity at tube outer diameter of 44.45 mm using Admiralty tube material where inlet cooling water at 70°F

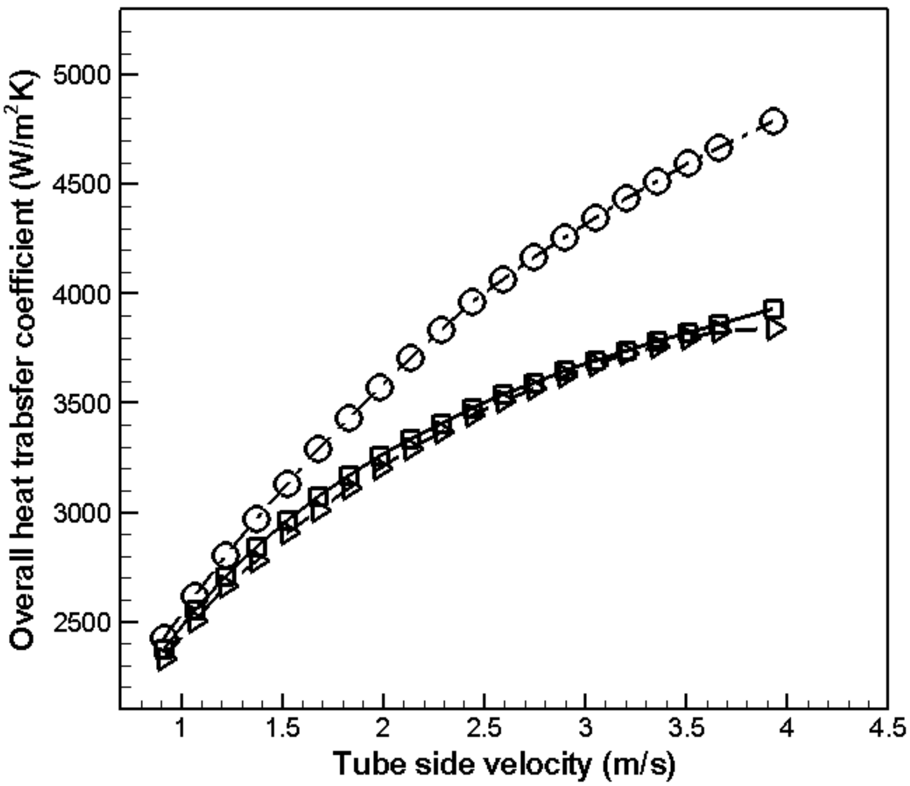


Figure 5.4: Overall Heat transfer coefficient vs velocity at tube outer diameter of 50.08 mm using Admiralty tube material where inlet cooling water at 70°F

5.2 Design and optimization of steam surface condenser

The effectiveness of the present approach using genetic algorithm (GA) is assessed by analyzing three case studies as following,

Case study 1:	18 MW Coal Based Captive Cogeneration Power Plant, Doc No: 1CYJ185288, by Siemens Ltd, Baroda
Case study 2:	9.9 MW Bua Sommai, Suwannaphum, Doc No: 1CYJ185182 by Siemens Ltd, Baroda
Case study 3:	24 MW KPR Sugar Plant, Doc No: 1CYJ185162 by Siemens Ltd, Baroda.

The specifications of above three cases are given in below:

Case study 1:

The steam with mass flow rate 16.70 kg/s having dryness fraction 0.88 from the turbine is condensed in condenser at pressure 0.103 bar. The cooling water is available at temperature of 32°C which enters in condenser inlet and comes out at temperature of 40.01°C.

Case study 2:

The steam with 9.12 kg/s having dryness fraction 0.88 from the turbine is condensed in condenser at pressure 0.103 bar. The cooling water is available at temperature of 32°C which enters in condenser inlet and comes out at temperature of 41.01°C.

Case study 3:

The steam with 20.641 kg having dryness fraction 0.88 from the turbine is condensed in condenser at pressure 0.111 bar. The cooling water is available at temperature of 32°C which enters in condenser inlet and comes out at temperature of 41.56°C.

Table 5.1: Design variables range taken from Siemens Ltd

No	Variables	Range
1	Shell inner diameter (mm)	4900.0-900.0
2	Tube outer diameter (mm)	24.0-19.0
3	Tube thickness (mm)	1.245-0.559
4	Tube materials	Admiralty brass, Titanium, Cu-Nickel, SS304

Table 5.2: Design variables range taken from TEMA

No	Variables	Range
1	Shell inner diameter (mm)	4900.0-900.0
2	Tube outer diameter (mm)	63.5-6.35
3	Tube thickness (mm)	3.404-0.559
2	Tube materials	Admiralty brass, Titanium, Cu-Nickel, SS304

The above case studies data are optimized design data obtained from Siemens Ltd, Baroda. The results of in-house GA code are compared with this data. The design is first optimized by considering capital cost of the condenser as an objective function and then optimized by considering total cost of the condenser as an objective function. The design is also optimized and compared between two design variables range, selected for comparison is shown in Table 5.1 and 5.2. Note that the first design variables range is taken from Siemens data while second is taken from Tubular Exchanger Manufacturers Association (TEMA) standard.

Table 5.3: GA Results of Case 1 comaped with Siemens Ltd. design using Siemens design variables range

Parameters obtained	Design variables range as per Siemens Ltd.		
	Siemens Ltd, Doc No: 1CYJ185288	GA solution	
		Capital cost as an objective	Total cost as an objective
$D_s\,(mm)$	2500	2280.25	2886.31
$d_o\,(mm)$	19.05	20.80	21.46
Tube thickness (mm)	0.7	0.72	0.63
Tube material	SS304	SS304	SS304
Number of Tubes	4246	3178	4783
$V_t\,(m/s)$	2.25	2.24	1.37
$\triangle p_t\,(kPa)$	-	19.90	6.86
$E_{\triangle P}\,(kW)$		24.54	8.46
$L\,(mm)$	5.05	6.03	4.50
$U\,(W/m^2K)$	-	2854.66	2465.61
$A(m^2)$	1263	1252.82	1450.51
$C_i\,(Rs/yr)$	1,91,507	1,90,256	2,14,229
$C_o\,(Rs/yr)$	-	-	8,57,962
$C_{total}(Rs/yr)$	1,91,507	1,90,256	10,72,191

Following points are taken care while optimizing condenser with design data taken from Siemens Ltd.

- There are 35 choices of shell inner diameter data from the set (i.e. 900, 950, 1000, 1050, 1100, 1150, 1200, 1250, 1300, 1350, 1400, 1450, 1500, 1550, 1600, 1700, 1800, 1900, 2000, 2100, 2200, 2300, 2400, 2500, 2700, 2900, 3100, 3300, 3500, 3700, 3900, 4100, 4300, 4500, and 4700 mm).
- There are 3 choices of tube outer diameter data from the set (i.e. 19.05, 21 and 23 mm)
- There are 4 choices of tube outer thickness data from the set (i.e. 0.559, 0.711, 0.889, 1.245 mm)

Table 5.4: GA Results of Different variables range for Case 1

Parameters obtained using GA	Capital cost as an objective		Total cost as an objective	
	Variables used as per Siemens range	Variables used as per TEMA range	Variables used as per Siemens range	Variables used as per TEMA range
$D_s\,(mm)$	2280.25	2647.80	2886.31	3077.91
$d_o\,(mm)$	20.80	20.32	21.46	35.23
Tube thickness (mm)	0.72	1.23	0.63	2.00
Tube material	SS304	SS304	SS304	SS304
Number of Tubes	3178	4493	4783	2019
$V_t\,(m/s)$	2.24	1.66	1.37	1.20
$\triangle p_t\,(kPa)$	19.90	11.20	6.86	5.69
$E_{\triangle P}\,(kW)$	24.54	12.27	8.46	6.24
$L\,(mm)$	6.03	5.63	4.50	9.37
$U\,(W/m^2K)$	2854.66	2381.54	2465.61	1837.48
$A(m^2)$	1252.82	1615.23	1450.51	2093.48
$C_i\,(Rs/yr)$	1,90,256	2,33,732	2,14,229	2,88,372
$C_o\,(Rs/yr)$	-	-	8,57,962	6,32,469
$C_{total}(Rs/yr)$	1,90,256	2,33,732	10,72,191	9,20,840

- There are 4 choices of tube materials can be use (i.e. Admiralty, SS304, Titanium, and Cu-Nickel).

Based on this choice of design variables, the total number of design combinations available are $35 \times 3 \times 4 \times 4 = 1680$.

Same way, the following points are taken care while optimizing condenser with design data taken from TEMA standard.

- There are 35 choices of shell inner diameter data from the set as per Siemens design variable range (same as Siemens range).

- There are 11 choices of tube outer diameter data from the set (i.e. 6.350, 9.525, 12.70, 15.875, 19.05, 22.225, 25.400, 31.750, 38.100, 50.8, and 63.5 mm).

- There are 9 choices of tube outer thickness data from the set (i.e. 0.559, 0.711, 0.889, 1.245, 1.651, 2.108, 2.769, 3.048, and 3.404 mm).
- There are 4 choices of tube materials can be use as per Siemens design variable range (i.e. same as Siemens range).

Based on this choice of design variables, the total number of design combinations available are $35 \times 11 \times 9 \times 4 = 13,860$, this means that if an exhaustive search is to be performed it will take at the maximum 13,860 function evaluations before arriving at global minimum cost of surface condenser. Therefore, strategy which takes few function evolutions using GA is the best one.

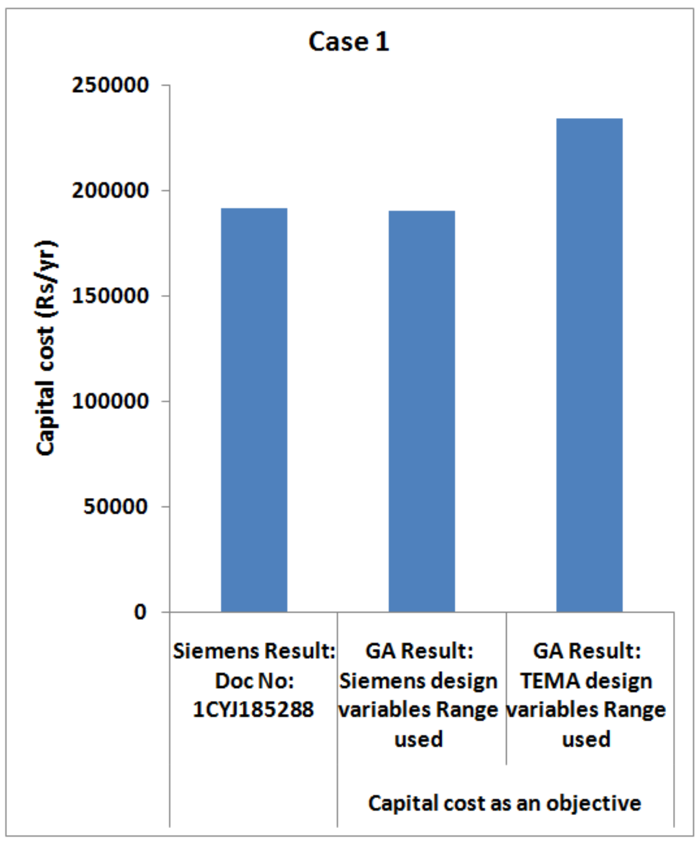


Figure 5.5: The Comparison of capital cost for Case 1

Considering the cost of condenser as the objective function, GA technique is applied to find the optimum heat transfer area and/or within optimum pumping power for a given heat duty of the surface condenser to select these design variables as an optimum solution.

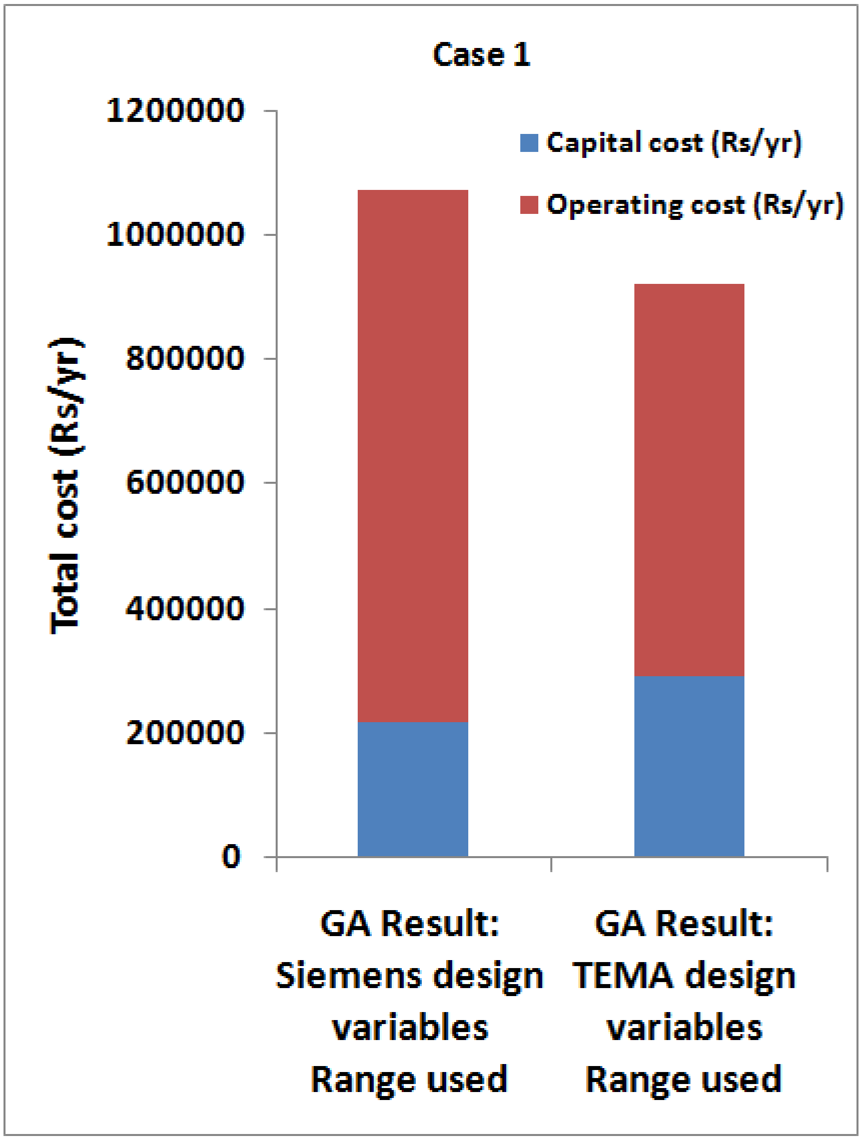


Figure 5.6: The Comparison of Total cost for Case 1

GA procedure selects design parameters randomly. After that, it generates initial population and evaluates each of them. Velocity and Pressure drop are estimated and calculated the tube length and if it is within the acceptable constrains, the final design occurs. This procedure goes on until optimum solution is found within all constrain.

Table 5.5: GA Results of Case 2 comaped with Siemens Ltd. design using Siemens design variables range

Parameters obtained	Design variables range as per Siemens Ltd.		
	Siemens Ltd, Doc no: 1CYJ185182	GA solution	
		Capital cost as an objective	Total cost as an objective
$D_s\,(mm)$	2100	1900.98	2131.67
$d_o\,(mm)$	19	20.47	20.54
Tube thickness (mm)	0.7	0.60	0.57
Tube material	SS304	SS304	SS304
Number of Tubes	2458	2282	2850
$V_t\,(m/s)$	1.95	1.53	1.21
$\triangle p_t\,(kPa)$	-	9.18	5.55
$E_{\triangle P}\,(kW)$	-	5.50	3.32
$L\,(mm)$	5.5	5.28	3.32
$U\,(W/m^2K)$	-	2709.12	2492.70
$A(m^2)$	801	775.43	842.75
$C_i\,(Rs/yr)$	1,32,431	1,28,996	1,37,996
$C_o\,(Rs/yr)$	-	-	3,36,548
$C_{total}(Rs/yr)$	1,32,431	1,28,996	4,74,544

For optimization of the above cases, the following values are considered in designed C program along with GA.

For Thermal design,

Cleanliness factor	0.85
Pump efficiency	0.85
Number of tube pass	2
Tube layout angle	30°
Pitch ratio	1.85-2.2

Table 5.6: GA Results for different variables range for Case 2

Parameters obtained using GA	Capital cost as an objective		Total cost as an objective	
	Variables used as per Siemens range	Variables used as per TEMA range	Variables used as per Siemens range	Variables used as per TEMA range
$D_s\,(mm)$	1900.98	2026.10	2131.67	2147.31
$d_o\,(mm)$	20.47	8.36	20.54	24.12
Tube thickness (mm)	0.60	0.65	0.57	0.71
Tube material	SS304	SS304	SS304	SS304
Number of Tubes	2282	15531	2850	2098
$V_t\,(m/s)$	1.53	1.68	1.21	1.20
$\triangle p_t\,(kPa)$	9.18	11.12	5.55	5.46
$E_{\triangle P}\,(kW)$	5.50	6.66	3.32	3.27
$L\,(mm)$	5.28	1.68	3.32	5.55
$U\,(W/m^2K)$	2709.12	3070.18	2492.70	2383.23
$A(m^2)$	775.43	684.24	842.75	881.46
$C_i\,(Rs/yr)$	1,28,996	1,16,564	1,37,996	1,43,108
$C_o\,(Rs/yr)$	-	-	3,36,548	3,31,172
$C_{total}(Rs/yr)$	1,28,996	1,16,564	4,74,544	4,74,280

For Economic calculation,

Operating Time $(hour/yr)$	8000
Electric cost (Rs/kWh)	5
Technical life of condenser (yr)	10
Interest rate (%)	20
Material cost ratio for Admirelty brass	1
Material cost ratio for SS304	0.05359
Material cost ratio for Titanium	3.1140
Material cost ratio for Cu-Nickel	1.40394

For GA operators,

Population size/iteration	40
substring length of each variable	10
Crossover probability	0.9
Mutation Probability	0.04
Crossover type	One site

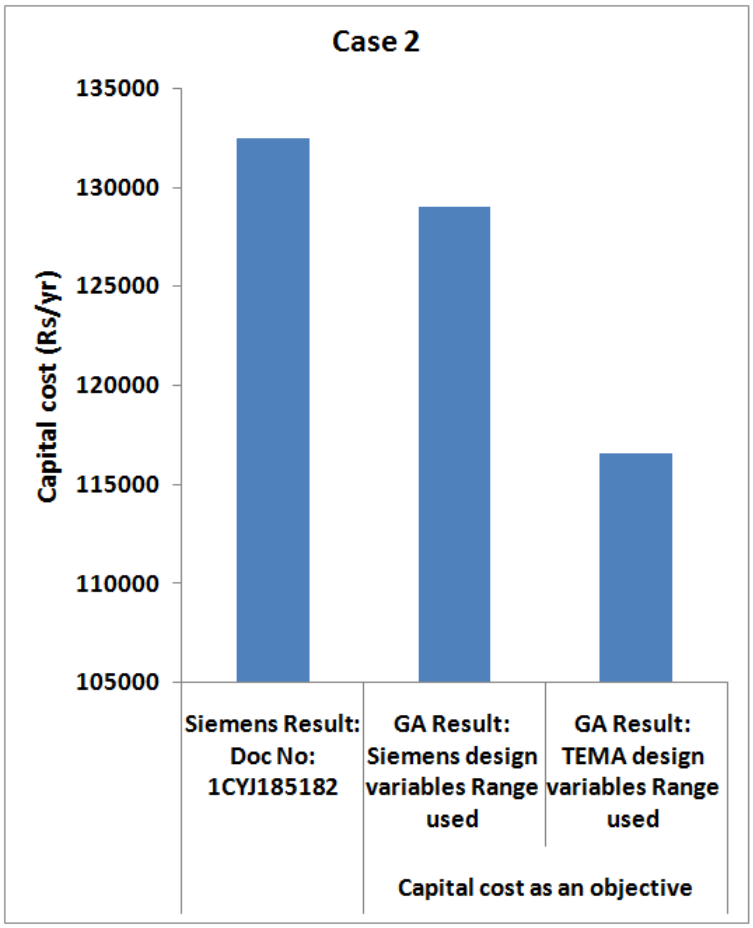


Figure 5.7: The Comparison of Capital cost for Case 2

The Constrains are:

Pressure drop (kPA)	50
Tube side velocity (m/s)	$1.2 < v < 2.3$
Tube length (m)	$L < 3.5 \times Ds$, $L > 1.5m$

These parameters are considered in all above cases to find optimum solution.

Results and discussion of case studies:

In case 1, the results of GA using design variables in Siemens range as in table 5.1 is well compared to the results of Siemens Ltd. If only capital cost of the condenser is considered, the heat transfer area as per Siemens design is 0.80% higher than the present work and hence the capital cost found in present work is 0.65% less than of Siemens data.

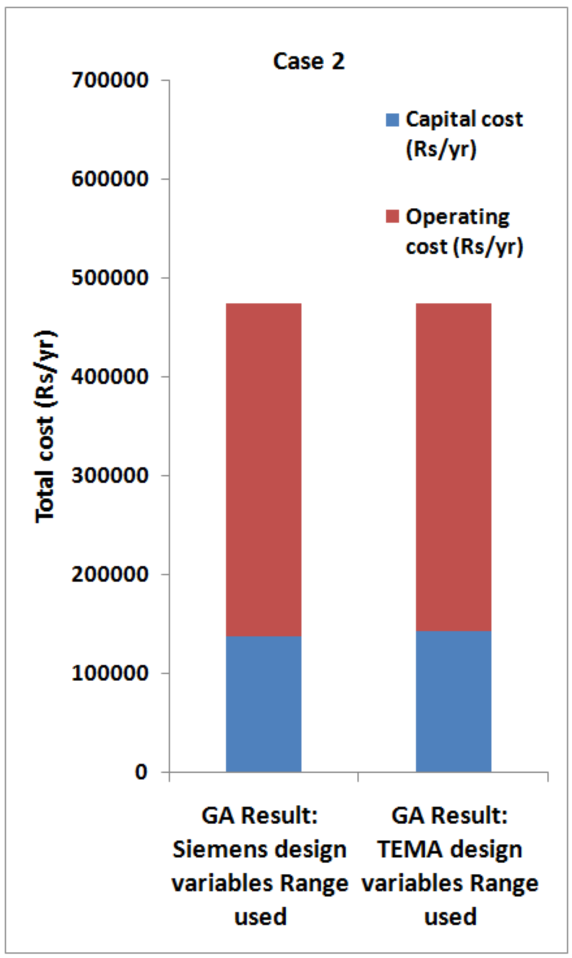


Figure 5.8: The Comparison of Total cost for Case 2

Table 5.7: GA Results of Case 3 comaped with Siemens Ltd. design using Siemens design variable range

Parameters obtained	Design variables range as per Siemens Ltd.		
	Siemens Ltd, Doc No: 1CYJ185162	GA solution	
		Capital cost as an objective	Total cost as an objective
$D_s\,(mm)$	2300	2389.74	2878.49
$d_o\,(mm)$	23	20.47	23.83
Tube thickness (mm)	0.7	0.64	0.67
Tube material	SS304	SS304	SS304
Number of Tubes	3136	3606	3861
$V_t\,(m/s)$	2.22	2.08	1.41
$\triangle p_t\,(kPa)$	-	17.72	7.64
$E_{\triangle P}\,(kW)$		27.06	9.75
$L\,(mm)$	7.0	6.34	5.88
$U\,(W/m^2K)$	-	2842.08	2459.90
$A(m^2)$	1572	1470.43	1698.88
$C_i\,(Rs/yr)$	2,28,652	2,16,610	2,43,490
$C_o\,(Rs/yr)$	-	-	9,88,173
$C_{total}(Rs/yr)$	2,28,652	2,16,610	12,31,663

If the total cost (capital cost + operating cost) is considered as an objective function for Siemens design variables range, the heat transfer area is increased by 13.63% and pumping power is reduced by 65.53%. From results, If condenser runs for 10 years, the cost of pumping power would rich too much high than capital cost. This running cost also the affects optimum design variables.

The results of GA using TEMA design range as shown in Table 5.2 is also well compared to the Siemens design range. If the capital cost is considered as an objective function, the optimized heat transfer area and capital cost in case 1 with TEMA design range is 22.44% and 18.60% higher than the Siemens design variables range respectively.

Table 5.8: GA Results of Different variables range for Case 3

Parameters obtained using GA	Capital cost as an objective		Total cost as an objective	
	Variables used as per Siemens range	Variables used as per TEMA range	Variables used as per Siemens range	Variables used as per TEMA range
$D_s\,(mm)$	2389.74	2925.42	2878.49	3183.48
$d_o\,(mm)$	20.47	35.29	23.83	38.86
Tube thickness (mm)	0.64	0.60	0.67	1.98
Tube material	SS304	SS304	SS304	SS304
Number of Tubes	3606	1818	3861	1775
$V_t\,(m/s)$	2.08	1.31	1.41	1.28
$\triangle p_t\,(kPa)$	17.72	6.39	7.64	6.31
$E_{\triangle P}\,(kW)$	27.06	8.14	9.75	8.05
$L\,(mm)$	6.34	8.88	5.88	10.43
$U\,(W/m^2K)$	2842.08	2336.21	2459.90	1849.53
$A(m^2)$	1470.43	1788.83	1698.88	2259.53
$C_i\,(Rs/yr)$	2,16,610	2,53,880	2,43,490	3,06,763
$C_o\,(Rs/yr)$	-	-	9,88,173	8,16,156
$C_{total}(Rs/yr)$	2,16,610	2,53,880	12,31,663	11,22,920

Similarly, while total cost of condenser is selected as an objective function, it is found that with TEMA design variable range, the heat transfer area is 30.72% and capital cost is 25.71% higher than Siemens design variables range. At a same time, it is also found that operating cost of condenser and hence the total cost reduces by 26.28% and 14.12% respectively. From above study, it is also concluded that the optimized result becomes better by increasing the search space for design variable using total cost as an objective function. These results are available in Table 5.4.

The results of comparison graph of Case 1 for capital cost based on capital cost as an objective function are shown in Figure 5.5. The results of comparison graph of Case 1 for total cost based on total cost as an objective function are shown in Figure 5.6.

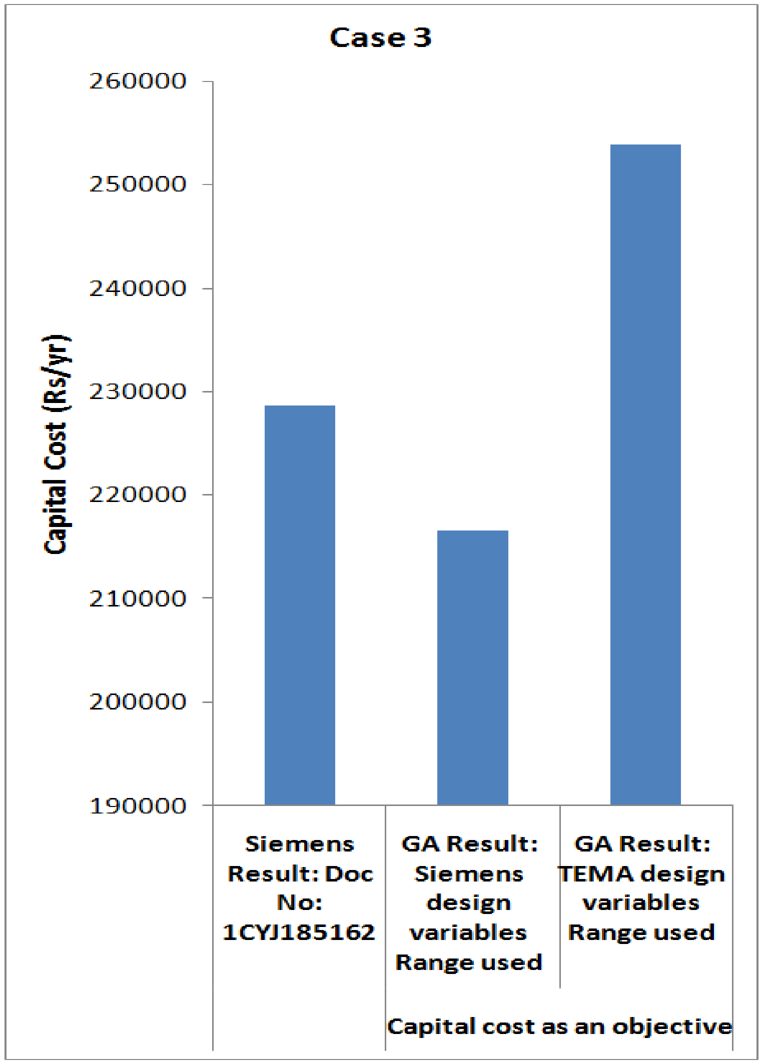


Figure 5.9: The Comparison of Capital cost for Case 3

The results of Case 2 are shown in Table 5.5 and 5.6. The results of comparison graphs of Case 2 for capital cost and total cost based on capital cost as an objective function are shown in Figure 5.7 and 5.8 respectively. The same way results of Case 3 are shown in Table 5.7 and 5.8 and the results of comparison graphs of Case 3 for capital cost and total cost based on capital cost as an objective function are shown in Figure 5.9 and 5.10 respectively.

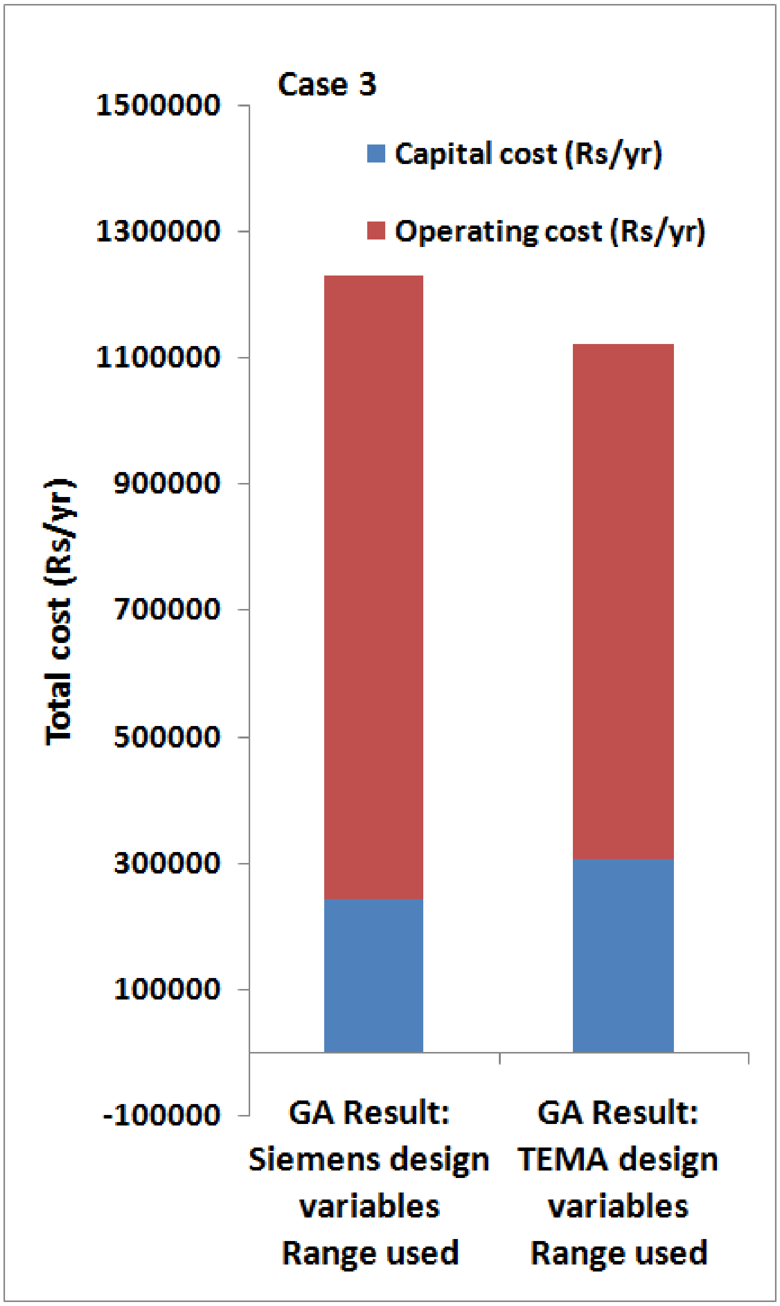


Figure 5.10: The Comparison of Total cost for Case 3

Chapter 6

Conclusion and Future Recommendation

Conclusion

Following conclusion has been made from the present work.

1. The design and rating of shell-and-tube surface condenser has been developed and the result obtained from the same has been compared with the result of HTRI software. It is found that the percentage difference in the results of overall heat transfer coefficient between present work and HTRI software is within 3.31%.

2. The results of present work are also compared with HEI's experimental results for overall heat transfer coefficient. It is found that the results for overall heat transfer coefficient are in well agreement with HEI's experimental results when tubeside velocity is less than to 2.3 m/s. The percentage deviation found is 11.256% however at high tubeside velocity the deviation increases.

3. Genetic algorithm is successfully implemented to cost optimize the surface condenser design based on heat transfer area and/or with pumping power using

various design variables such as shell inner diameter, tube outer diameter, tube thickness and tube material.

4. The optimization of steam surface condenser using GA has been carried between two objective functions, one for capital cost and second for total cost. Three different case studies for optimization have been carried using GA. It is found that in these cases, the operating cost was quite higher than the capital cost for steam surface condenser. Therefore, the optimization based on total cost is more reliable than capital cost.

5. It is also found that from the certain cases where total cost as an objective function, the total cost was quite less for TEMA design variables range for dimension which one has widen range than Siemens design variable range. Therefore, it is concluded that by increasing the dimension range of design variables using total cost as an objective function the efficiency of GA for optimization is improves.

Future recommendation

From the present work, following recommendation has been made.

1. In present code for design and rating only Nusselt correlation implemented to calculate the heat transfer coefficient on shell side . It recommended that more advance correlation can be implemented.

2. The present work only concentrate on GA as an optimization technique. However, this same can be compared with other optimization technique for better results.

3. In the present work only four design variables i.e. shell internal diameter, tube outer diameter, tube thickness, and tube material are considered. It

is recommended that apart from these four design variables the other design variables like i.e. number of tube pass, tube pitch and tube layout angle can also be considered.

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Appendix A

Flow Charts of Program Code

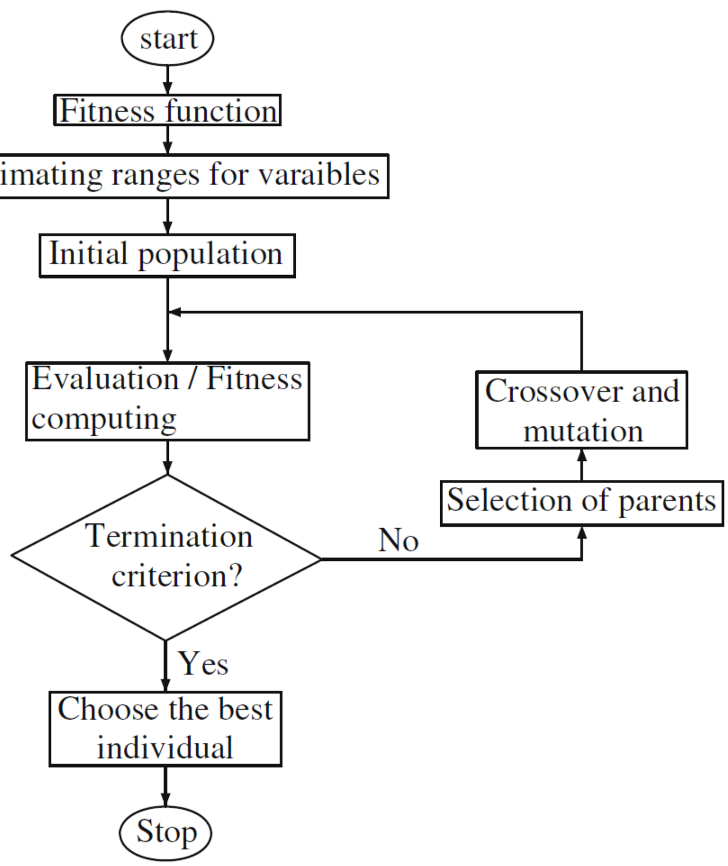

Figure A.1: Flow chart of Genetic Algorithm

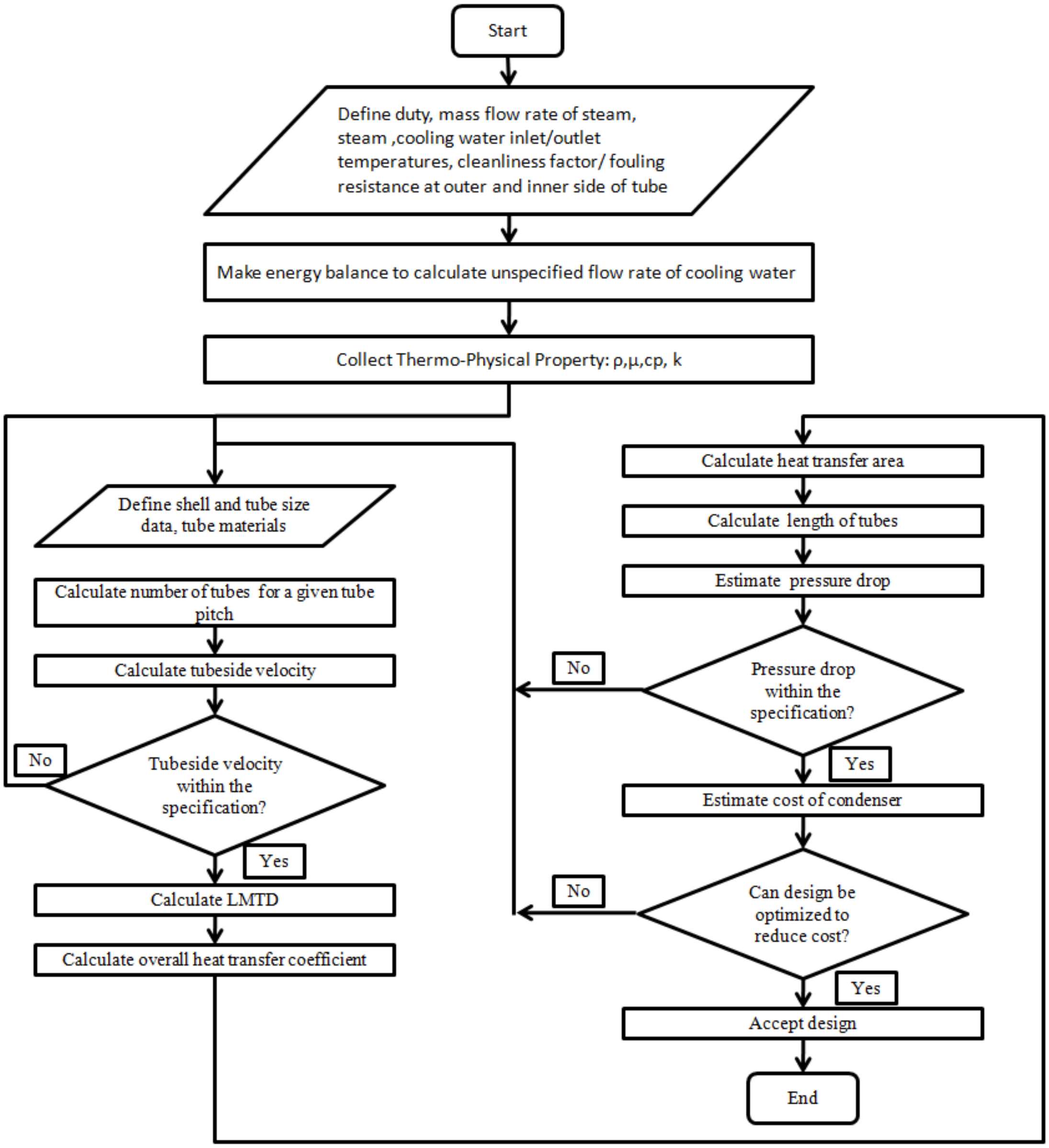

Figure A.2: Flow chart of surface condenser design

Appendix B

HTRI Software Results

HTRI **Output Summary** Page 1

Released to the following organization:
nirma
nit

Xist E Ver. 5.00 02-05-2011 17:23 SN: 1600211715 **SI Units**

Rating - Horizontal Single Pass TEMA BXM Shell With No Baffles

No Data Check Messages.
See Runtime Message Report for Warning Messages.

Process Conditions		Hot Shellside		Cold Tubeside	
Fluid name		Steam		water	
Flow rate	(kg/s)		15.9612		1163.82
Inlet/Outlet Y	(Wt. frac vap.)	0.980	0.000	0.000	0.000
Inlet/Outlet T	(Deg C)	45.43 *	40.61 *	21.11	28.97
Inlet P/Avg	(kPa)	10.000	8.871	0.000	0.000
dP/Allow.	(kPa)	2.258	0.000	14.426	0.000
Fouling	(m2-K/W)		0.000000		0.000000

Exchanger Performance					
Shell h	(W/m2-K)	9948.45	Actual U	(W/m2-K)	2956.93
Tube h	(W/m2-K)	4816.08	Required U	(W/m2-K)	655.80
Hot regime	(--)	Gravity	Duty	(MegaWatts)	38.2556
Cold regime	(--)	Sens. Liquid	Area	(m2)	3146.63
EMTD	(Deg C)	18.5	Overdesign	(%)	350.89

Shell Geometry			Baffle Geometry		
TEMA type	(--)	BXM	Baffle type	(--)	Support
Shell ID	(mm)	1905.00	Baffle cut	(Pct Dia.)	
Series	(--)	1	Baffle orientation	(--)	
Parallel	(--)	1	Central spacing	(mm)	976.268
Orientation	(deg)	0.00	Crosspasses	(--)	1

Tube Geometry			Nozzles		
Tube type	(--)	Plain	Shell inlet	(mm)	1714.50
Tube OD	(mm)	19.050	Shell outlet	(mm)	154.051
Length	(m)	10.973	Inlet height	(mm)	1433.51
Pitch ratio	(--)	1.2500	Outlet height	(mm)	11.113
Layout	(deg)	90	Tube inlet	(mm)	844.552
Tubecount	(--)	4896	Tube outlet	(mm)	844.552
Tube Pass	(--)	1			

Thermal Resistance, %		Velocities, m/s		Flow Fractions	
Shell	29.71	Shellside	28.61	A	0.000
Tube	67.81	Tubeside	1.02	B	0.954
Fouling	0.00	Crossflow	24.63	C	0.046
Metal	2.482	Window	0.00	E	0.000
				F	0.000

Figure B.1: HTRI result sheet at inner tube diameter of 19.05 mmusing Admiralty tube material where inlet cooling water at 70°F

HTRI

Output Summary Page 1

Released to the following organization:

nirma

nit

Xist E Ver. 5.00 02-05-2011 17:33 SN: 1600211715 **SI Units**

Rating - Horizontal Single Pass TEMA BXM Shell With No Baffles

No Data Check Messages.

See Runtime Message Report for Warning Messages.

Process Conditions		**Hot Shellside**		**Cold Tubeside**	
Fluid name		Steam		water	
Flow rate	(kg/s)		15.9612		1163.53
Inlet/Outlet Y	(Wt. frac vap.)	0.980	0.000	0.000	0.000
Inlet/Outlet T	(Deg C)	45.43 *	40.83 *	21.11	28.97
Inlet P/Avg	(kPa)	10.000	8.922	0.000	0.000
dP/Allow.	(kPa)	2.156	0.000	22.761	0.000
Fouling	(m2-K/W)		0.000000		0.000000

Exchanger Performance

Shell h	(W/m2-K)	8503.87	Actual U	(W/m2-K)	3495.03
Tube h	(W/m2-K)	6716.06	Required U	(W/m2-K)	1498.63
Hot regime	(--)	Gravity	Duty	(MegaWatts)	38.2461
Cold regime	(--)	Sens. Liquid	Area	(m2)	1346.23
EMTD	(Deg C)	19.0	Overdesign	(%)	133.21

Shell Geometry			**Baffle Geometry**		
TEMA type	(--)	BXM	Baffle type	(--)	Support
Shell ID	(mm)	1905.00	Baffle cut	(Pct Dia.)	
Series	(--)	1	Baffle orientation	(--)	
Parallel	(--)	1	Central spacing	(mm)	1193.22
Orientation	(deg)	0.00	Crosspasses	(--)	1

Tube Geometry			**Nozzles**		
Tube type	(--)	Plain	Shell inlet	(mm)	1714.50
Tube OD	(mm)	25.400	Shell outlet	(mm)	154.051
Length	(m)	10.973	Inlet height	(mm)	1443.04
Pitch ratio	(--)	1.2500	Outlet height	(mm)	11.113
Layout	(deg)	90	Tube inlet	(mm)	844.552
Tubecount	(--)	1571	Tube outlet	(mm)	844.552
Tube Pass	(--)	1			

Thermal Resistance, %		**Velocities, m/s**		**Flow Fractions**	
Shell	41.10	Shellside	28.40	A	0.000
Tube	56.01	Tubeside	1.70	B	0.955
Fouling	0.00	Crossflow	24.10	C	0.045
Metal	2.897	Window	0.00	E	0.000
				F	0.000

Figure B.2: HTRI result sheet at inner tube diameter of 25.4 mmusing Admiralty tube material where inlet cooling water at 70°F

HTRI **Output Summary** Page 1

Released to the following organization:
nirma
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Xist E Ver. 5.00 02-05-2011 17:35 SN: 1600211715 **SI Units**

Rating - Horizontal Single Pass TEMA BXM Shell With No Baffles

No Data Check Messages.
See Runtime Message Report for Warning Messages.

Process Conditions		**Hot Shellside**		**Cold Tubeside**	
Fluid name		Steam		water	
Flow rate	(kg/s)		15.9612		1163.13
Inlet/Outlet Y	(Wt. frac vap.)	0.980	0.000	0.000	0.000
Inlet/Outlet T	(Deg C)	45.43 *	41.13 *	21.11	28.97
Inlet P/Avg	(kPa)	10.000	8.987	0.000	0.000
dP/Allow.	(kPa)	2.026	0.000	46.512	0.000
Fouling	(m2-K/W)		0.000000		0.000000

Exchanger Performance					
Shell h	(W/m2-K)	6481.74	Actual U	(W/m2-K)	3834.10
Tube h	(W/m2-K)	10593.1	Required U	(W/m2-K)	5933.20
Hot regime	(--)	Gravity	Duty	(MegaWatts)	38.2331
Cold regime	(--)	Sens. Liquid	Area	(m2)	343.413
EMTD	(Deg C)	18.8	Overdesign	(%)	-35.38

Shell Geometry			**Baffle Geometry**		
TEMA type	(--)	BXM	Baffle type	(--)	Support
Shell ID	(mm)	1905.00	Baffle cut	(Pct Dia.)	
Series	(--)	1	Baffle orientation	(--)	
Parallel	(--)	1	Central spacing	(mm)	1789.83
Orientation	(deg)	0.00	Crosspasses	(--)	1

Tube Geometry			**Nozzles**		
Tube type	(--)	Plain	Shell inlet	(mm)	1714.50
Tube OD	(mm)	44.450	Shell outlet	(mm)	154.051
Length	(m)	10.973	Inlet height	(mm)	1447.80
Pitch ratio	(--)	1.2500	Outlet height	(mm)	11.113
Layout	(deg)	90	Tube inlet	(mm)	844.552
Tubecount	(--)	229	Tube outlet	(mm)	844.552
Tube Pass	(--)	1			

Thermal Resistance, %		**Velocities, m/s**		**Flow Fractions**	
Shell	59.16	Shellside	28.60	A	0.000
Tube	37.72	Tubeside	3.56	B	0.956
Fouling	0.00	Crossflow	23.98	C	0.044
Metal	3.127	Window	0.00	E	0.000
				F	0.000

Figure B.3: HTRI result sheet at inner tube diameter of 44.45 mmusing Admiralty tube material where inlet cooling water at 70°F

HTRI **Output Summary** Page 1

Released to the following organization:
nirma
nit

Xist E Ver. 5.00 02-05-2011 17:37 SN: 1600211715 **SI Units**

Rating - Horizontal Single Pass TEMA BXM Shell With No Baffles

No Data Check Messages.

See Runtime Message Report for Warning Messages.

Process Conditions		Hot Shellside		Cold Tubeside	
Fluid name		Steam		water	
Flow rate	(kg/s)		15.9612		1163.06
Inlet/Outlet Y	(Wt. frac vap.)	0.980	0.000	0.000	0.000
Inlet/Outlet T	(Deg C)	45.43 *	41.18 *	21.11	28.97
Inlet P/Avg	(kPa)	10.000	8.999	0.000	0.000
dP/Allow.	(kPa)	2.003	0.000	51.745	0.000
Fouling	(m2-K/W)		0.000000		0.000000

Exchanger Performance

Shell h	(W/m2-K)	6212.78	Actual U	(W/m2-K)	3834.22
Tube h	(W/m2-K)	11309.7	Required U	(W/m2-K)	7599.14
Hot regime	(--)	Gravity	Duty	(MegaWatts)	38.2308
Cold regime	(--)	Sens. Liquid	Area	(m2)	268.640
EMTD	(Deg C)	18.7	Overdesign	(%)	-49.54

Shell Geometry			Baffle Geometry		
TEMA type	(--)	BXM	Baffle type	(--)	Support
Shell ID	(mm)	1905.00	Baffle cut	(Pct Dia.)	
Series	(--)	1	Baffle orientation	(--)	
Parallel	(--)	1	Central spacing	(mm)	2147.79
Orientation	(deg)	0.00	Crosspasses	(--)	1

Tube Geometry			Nozzles		
Tube type	(--)	Plain	Shell inlet	(mm)	1714.50
Tube OD	(mm)	50.080	Shell outlet	(mm)	154.051
Length	(m)	10.973	Inlet height	(mm)	1455.51
Pitch ratio	(--)	1.2500	Outlet height	(mm)	11.113
Layout	(deg)	90	Tube inlet	(mm)	844.552
Tubecount	(--)	159	Tube outlet	(mm)	844.552
Tube Pass	(--)	1			

Thermal Resistance, %		Velocities, m/s		Flow Fractions	
Shell	61.71	Shellside	28.64	A	0.000
Tube	35.17	Tubeside	4.01	B	0.957
Fouling	0.00	Crossflow	23.96	C	0.043
Metal	3.121	Window	0.00	E	0.000
				F	0.000

Figure B.4: HTRI result sheet at inner tube diameter of 50.08 mmusing Admiralty tube material where inlet cooling water at 70°F