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Optimization of the Design of Shell and Double Concentric Tube Heat Exchanger using the TLBO Algorithm

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Abstract

Heat Exchangers are devices that allow energy in form of heat to be transferred between two or more fluids. HEs are utilized in a wide range of commercial processes, including chemical, steel, and power generation. Here, we focus on the optimization of shell and dual concentric tube HEs. This type of HE has been in use for many years because of their reliable service to enterprise, the availability of a set of symbols, and perfection in design and modeling, and they are made from a wide range of materials. In this paper a novel computative technique called TLBO Algorithm is used, and the aim is to lower the overall cost by designing the HE with a shell and two concentric tubes. For every iteration, the algorithm detects and replaces duplicate solutions in order to produce an effective functional evaluation, which may then be used to choose the best solution. With a usual start, it proves to be quite efficient, and therefore this algorithm will assist us in achieving our aim function, which is in contrast to conventional HEs. With GA based HEs with a shell and twin concentric tube HEs, the overall cost has decreased by roughly 43% and 34%, respectively.

Keywords: *Economic Optimization, Heat Exchangers, TLBO Algorithm*

1.0 Introduction

A comprehensive and rigorous analysis will be necessary to lower total expenses. A HE with a shell and tube structure was just what we needed. As the name implies, a shell surrounds a cluster of tubes in this form of HE. To create a temperature differential between dual fluid, one fluid passes thru the tubes while the other passes over the shell side. The HE is the most prevalent of the several kinds due to its well-established design and manufacturing procedures. HE's with shell and twin concentric tubes.

Many scholars have worked on shell and tube HEs to address the design constraints imposed by geometric and operational variables. An automated approach was employed by Logsdon *et al.*[1] for the optimal design of HEs. Shenoy *et al.*[2] created a design that was as tiny and inexpensive as feasible. The other studies describe design variables that are determined by the design specifications as well as assumptions made about a number of mechanical and thermal characteristics.

Andrade *et al.*[3] presented a complete equipment design in the synthesis and optimization of HE networks, whereas the other studies describe design variables that

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are determined by the design specifications as well as assumptions made about a number of mechanical and thermal characteristics by achieving a balance abrupt reductions in pressure and heat exchange recitals based on multiple iterative efforts to match the parameters, an appropriate design may be constructed[4].

Costa *et al*.[5] offered a design optimization of STHE's as a result of a study through the tube summary chart, with the purpose of reducing the equipment's thermal surface. For shell-and-tube HE's with only monophonic movement of fluids on both sides of the shell - and - tube, Kara *et al*.[6] a computer-aided layout model was employed to evaluate the ideal heat transfer surface area. Caputo *et al*.[5] proposed a heat transfer design that is cost-effective. The STHE for the concentric tubes was introduced by Baadache *et al*.[7,8] and the design of HE for the same was created using the Genetic Algorithm.

With shell and twin concentric tubes, the research addresses the usage of a GA to construct a HE with the goal of lowering the total cost. The Vector Evaluated Genetic Algorithm was created by Schaffer *et al*.[9], It's been utilized to tackle a variety of more than one challenges as well[10,11].

Mishra *et al*.[12] is used genetic algorithm-based optimization for crossflow HE, where the objective function is the amount of entropy factors must be kept as low as possible. Ortega *et al*.[13] made use of a genetic algorithm to decrease the shell and tube HEs' total operating cost at concession.

Eryener *et al*.[14] proposed a thermo-economic analysis approach for STHE layout focused on baffle spacing. Asadi *et al*.[15] utilised the cuckoo search method to build shell and tube HEs, whereas Guo *et al*.[16] used the genetic algorithm. The Bell Delaware methodology characterise the shell-side flow in this approach. However, it is vital to investigate the use of non-traditional optimization techniques. Sahin *et al*.[17] proposed a layout and practical optimization of STHE's utilising the Artificial Bee Colony (ABC) approach. A work on shell-and-tube HE design optimization was published by Queiroz *et al*[18]. Vahdat *et al*.[19] established an economic optimization of a shell and tube HE using constructal theory. To optimise the design of shell and tube HEs, Soleimani *et al*.[20] employ global sensitivity analysis and the harmony search method. Particle swarm optimization was utilised by Biscaia *et al*.[21] to improve the design of shell and tube HEs. Some researchers employ objective functions to lower overall cost in optimization problems, and heat transfer has been the focus of the majority of the research. Tharakeshwar *et al*.[22] employ a multi-objective optimization of bat method in this research for shell and tube HEs, and it considers multivariate functions such as competence and total cost optimization[23]. Veerabhadrappa *et al*. For HE of shell and dual concentric tubes, the Cuckoo Optimisation Algorithm (COA) approach has been devised. As part of the optimization effort, the HE with a shell and two concentric tubes will be sized.

The optimization procedure starts with the selection of essential geometrical parameters including tube numerals, interior and exterior tube diameters, baffle gradient, and so on. The method takes into account the quantitative and operational constraints that are frequently recommended by design standards. The examples presented show that the TLBO Algorithm is an effective tool for HE optimum design[24].

The TLBO Algorithm is used to fix HE optimization problems involving several design variables. However, it is vital to Look into using non-orthodox methods. The TLBO algorithm is one that can be followed. A case study was offered to show the capability of the proposed technology, providing considerable cost reductions as compared to the traditional shell-and-tube HE designs.

2.0 Description of Heat Exchanger

A STHE is a form of heat exchanging device that is made using a wide cylindrical casing, or shell, with tubing bundles compacted inside. A shell surrounds a group of tubes in this type of heat exchanger it; they provide a temperature driving force within them (heat exchanger with shell and two tubes) by passing fluid streams of various temperatures in parallel/opposite flow separated by a physical boundary in the form of a pipe. The optimization of a heat exchanger with various design factors is more realistic and desired. To meet the design restrictions, many geometric and operational factors abound in STHE. The heat exchanger is an a necessary component of any energy reign and as a result, environmental protection policy, industrial businesses are the design variables that are determined by design standards and assumptions about a number of mechanical and thermodynamic factors.

2.1 Construction and Working

The major components of shell and double concentric tube HE's are heat transfer tubes, tube panels, shell and shell side collectors and distributors, tube flow rates and nozzles, baffles, tie rods. The shell, triple distributors, triple collectors, and channel coverings with quadrivial tube sheets are shown Figure 1c. Two fluids of same temperature enter through the primary and triennial distributors respectively, and exit through the triennial and primary collectors.

The intermediate collector (or distributor) passes the fluids of two distinct temperature levels. The first fluid (of the same temperature or as the triennial fluid) enter through the 1st distributor, drive through the 1st tube sheet and leaves the 1st collector, crossing the annular passages formed by the inner tubes and the 2nd tube sheet, before exiting through the 3rd tube sheet and the last front collector distributor, traverse the heat exchanger at the external of the shell side (two envelopes), and exits the HE by the 3rd collector in the same way as the second fluid. These shell and dual concentric tube heat exchangers are designed differently depending on the fluids being used. Fins can be used to corrugate tubes because of the baffles, fluid travelling towards the shell may often circulate in many passes. It makes it easier to irrigate all of the tubes,

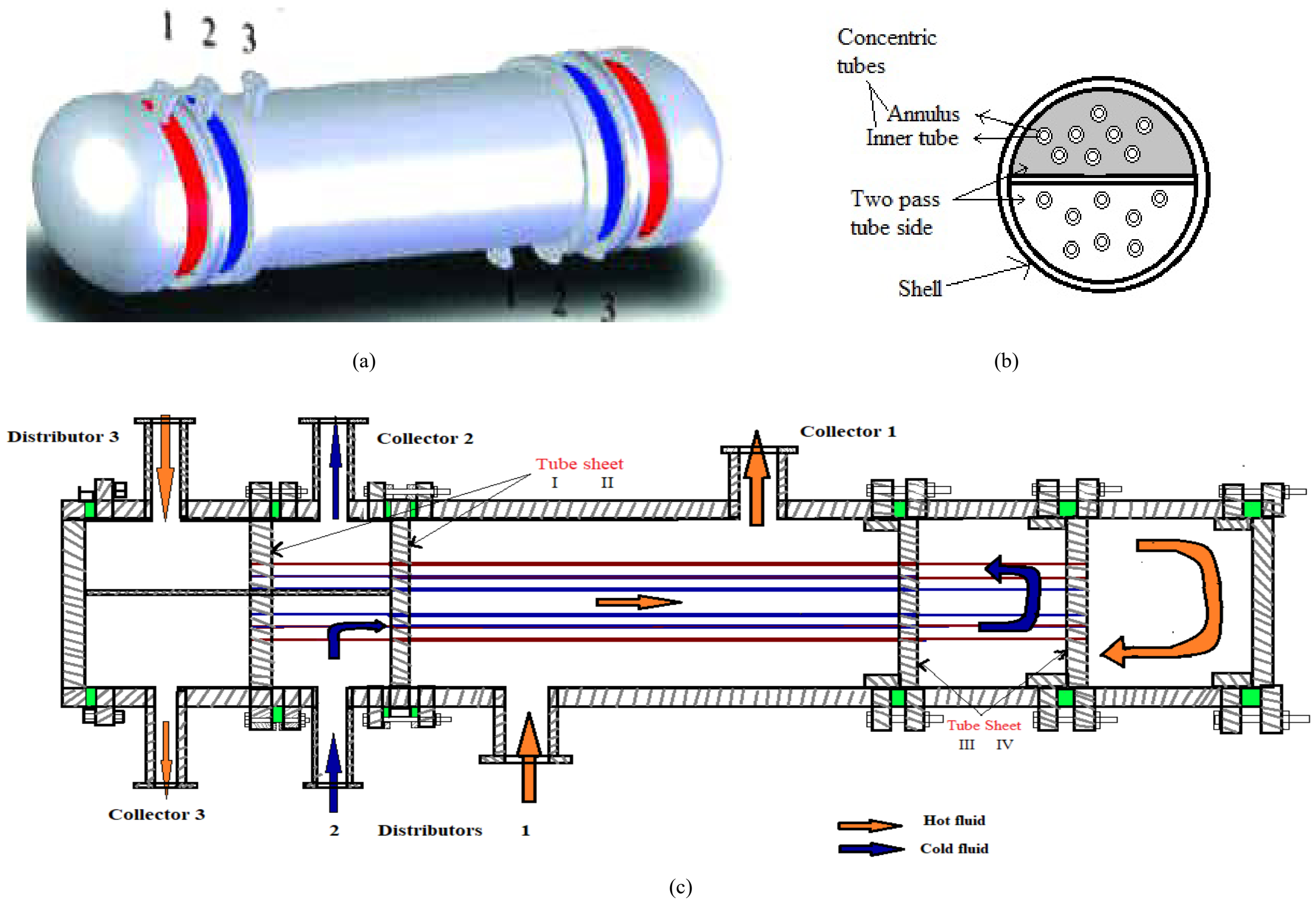

Figure 1. a. Diagrammatic portrayal of 3-D shell-and-dual concentric-tube heat exchanger[7]. b. Diagrammatic portrayal of shell-and-dual concentric-tube heat exchanger in top view. c. A diagrammatic view of shell-and-dual concentric-tube heat exchanger in crosses sectional view.

segmental baffles, disc and doughnut baffles, and orifice baffles are among the several forms of baffles. The tubes might be scattered or lined in their placement.

The fluid flowing in the concentric annulus tube's travels in the reverse way of the two other fluid's global circulation with similar characteristics.

3.0 Proposed Approach

The steps of the HE's optimum design technique are as follows:

- Studying the literature of the previous done work on HEs. Analysing and understanding their problem definitions and problem-solving approach
- The LMTD technique is used to calculate the heat transfer area by taking into account a set of design parameters.
- Investment, operational cost, and objective function evaluation.
- Using an optimization method to choose a novel set of values for the design variables.
- Iteration of the preceding phases until the objective function is reduced to its simplest form.

The whole procedure is catalogued in Figure 2: The mass flow of each fluids, entry and exit temperatures of each fluids, The models of tubular plate, the height, tube passes number (1, 2, 4), as well as tube fouling resilience are all fixed parameters supplied by the user: R_{foul} tube and R_{foul} shell side tubes and shell respectively, and thermodynamic characteristics of the fluids (two fluids), that is in Table 2.

The inside shell diameter D_s, the exterior tube external diameter D_o, the internal tube out the front of diameter do, and the interval between baffles E are factors that should be optimised with values ascribed iteratively by the optimisation method. The heat exchanger model calculation programme determines the standard of the heat transfer coefficients: surface area for heat exchange S, the total tube numbers N_t, the length L, and the velocity flow v_1, v_2, and v_3: Defining the features of the HE's structure and fulfilling the thermal standards on the tube side, circular passage side, and shell side.

The length "L" of the HE is a numerical many of the baffle distance "E," with leftover gap equal to the sum of baffle widths. The objective function is then estimated using the set on values from the velocities of flow and the productive features of the HE construction. The TLBO Algorithm adjusts the escalation trial assessment depending on The TLBO Algorithm alters the escalation variables' trial assessment depending on the goal function's value, which is then utilised to produce a new choice (goal) function that describes a new HE designs. The approach is continued until the minimum of the desired function is found. Algorithm Description Since its inception in 2011, TLBO has found a wide range of applications in several disciplines of engineering and research. It's a population-based strategy that can solve both restricted and incontinent optimization problems by constantly altering a population of individual solutions algorithm, a ground-breaking Meta - heuristic optimization technique based on educational facilities. Here a unique feature that may make the process of solving optimization issues easier. Controlling parameters are required by all evolutionary and metaheuristic algorithms, which vary depending on the task. These regulating variables are classified into 2 categories:

- Overarching factors such as population numbers, stages or iterations, and so on
- Defined specifications that are dependent on the algorithm. On the other hand, if the individual governing specifications are chosen incorrectly, the answer might become trapped in a maximum in the immediate vicinity, or the quality of the solution may suffer. In this case, the user should do assessment of sensitivity to determine the appropriate measurements of the individual parameters, despite the fact that this will take more time.

Unlike other optimization strategies, the TLBO process does not involve any of these specific values.

It only requires a few basic parameters, such as population numbers and cycles. It seems to be useful attribute which makes the algorithm's implementation easier. As a result, the TLBO algorithm is self-regulatory. The teacher–student connection in a classroom, the teacher's impact on pupils or students, and the interactions of learners and their effects on one another are all inspirations for the TLBOA. The algorithm's two core sectors, referred to as the area of teaching and learner phase, respectively, are instructor and pupils, respectively, in terms of generation size and number. It looks to be a fascinating feature that makes the algorithm easier to utilize.

4.0 TLBO Algorithm

The TLBO is separated into 2 phases: the 'Teacher phase' then the 'Learner phase.' The operation throughout each one of these stages is described below[26] and flowchart in Figure 2.

4.1 Teacher Phase

It's the initial stage in the algorithm, and here the kids pick up on the teacher's lessons. Based on his or her

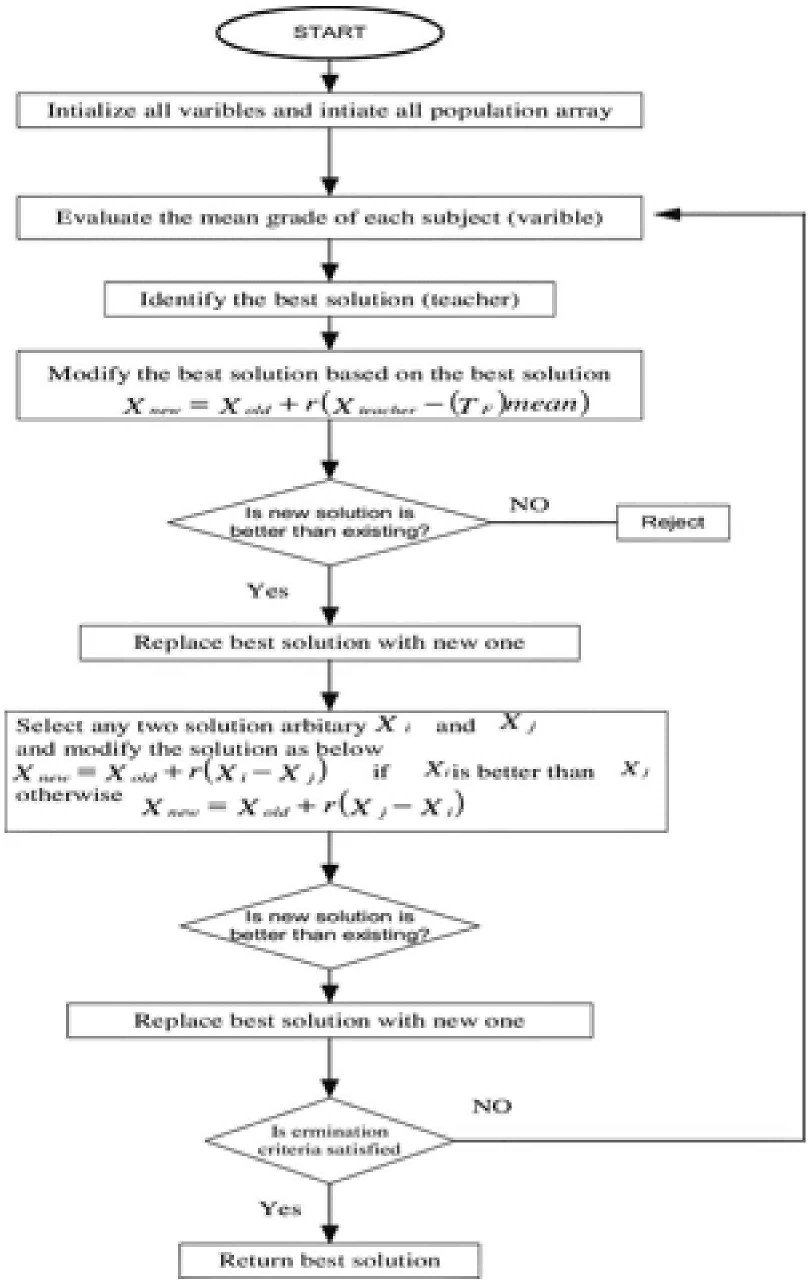

Figure 2. Proposed flowchart.

talents, a teacher strives to increase the average result of the class in the topic they are teaching throughout this phase. Assume that at any iteration, Here are 'm' topics, 'n' learners, and Mj,i is the total outcome of the learners in a given subject j so from this a best solution is expected. the best learner k_{best} outcome may be defined as the finest outcome $X_{total-kbest}$ that it achieved across all topics in the whole population of learners. The algorithm, however, considers the best learner identified as the instructor because a teacher is often thought of as a highly educated somebody who trains pupils in order for them to achieve better outcomes. The comparable result for each subject is represented by,

$$Difference\_Mean_{j,k,i} = r_i(_{Xj,kbest,i} - T_F M_{j,i}), \quad (a)$$

Where $X_{j,k\ best,i}$ In the topic j results, is the greatest student. The teaching variable, T_F, causes the movement in mean value, and ri is a unique number between 0 and 1 that influences the change in mean value. The value of T_F can be either 1 or 2. The value of T_F is arbitrarily chosen with the same chance as,

$$T_F = round\ [1+rand\ (0,1)\{2-1\}] \quad (b)$$

4.2 Learner Phase

It is the next component of the algorithm, in which learners gain description through interacting to one another. A pupil connects to other learners at random in order to improve his or her knowledge. If another student has greater knowledge than the learner, the learner learns new things. Select 2 learners P and Q at random from a population of 'n', such that $X'_{total-P,i} \neq X'_{total-Q,i.}$

$$X''_{j,P,i} = X'_{j,P,i} + r_i\left(X'_{j,Q,i} - X'_{j,P,i}\right),$$

$$\text{If } X'_{total\ -Q,I} < X'_{total\ -P,i} \quad (c)$$

$$X''_{j,P,i} = X'_{j,P_i} + r_i\left(X'_{j,P,i} - X'_{j,Q,i}\right),$$

$$\text{If } X'_{total}, P, R < X'_{total} - Q, i \quad (d)$$

Where, $X''_{j,P,I}$ If it delivers a superior function value, it is acceptable. The above expressions are for minimizing challenges. In the case of maximizing challenges, the equation (e-f) are used.

$$X''_{j,P,i} = X'_{j,P,i} + r_i\left(X'_{j,P,i} - X'_{j,Q,i}\right),$$

$$\text{If } X'_{total\ -Q,i} < X'_{total\ -P,,l} \quad (e)$$

$$X''^{,P,i} = X'_{j,P,_i} + r\left(X'_{j,Q,i} - X'_{j,P,i}\right),$$

$$\text{If } X'_{total} - P, i < X'_{toral\ -Q,I.} \quad (f)$$

5.0 Mathematical Modelling

The following equation describes the effect of heat transmission between hot and cold streams.

$$\Phi = \left(K_{1,2}S_{1,2}F + K_{2,3}S_{2,3}\right)\Delta T_{ML} \quad (1)$$

$$\Phi = m_f cp_1\left(T_{e1} - T_{s1}\right) = m_2 cp_2\left(T_{s2} - T_{e2}\right) \quad (2)$$

Where,

$$S_{1,2} = N_t \pi D_o L S_{1,2} = N_t \pi D_o L \quad (3)$$

$$S_{2,3} = N_t \pi d_o L \quad (4)$$

On designing the heat exchanger (heat transfer surface), The LMTD is more widely used than the other approaches (NUT, Ɛ-NUT, etc.). The other approaches are used to model a heat exchanger that already exists. ΔTML is given by

$$\Delta T_{ML} = \frac{\left(T_{e1} - T_{s2}\right) - \left(T_{s1} - T_{e2}\right)}{\ln\dfrac{\left(T_{e1} - T_{s2}\right)}{\left(T_{s1} - T_{e2}\right)}} \quad (5)$$

The number of tubes N_t is computed using the following formula:

$$Nt = C\left(\frac{D_s - 0.02}{D_o}\right)^n \quad (6)$$

The speed of the stream for the internal tube is given by:

$$v_3 = m_3 s\ /\ (\rho_3 N_t sp_3) \quad (7)$$

$$sp_3 = \pi di^{\,2}\ /\ 4 \quad (8)$$

$$m_3 = m_f\ /\ 2 \quad (9)$$

The Reynolds number is determined by:

$$Re_3 = v_3 d_i / v_3 Re_3 = v_3 d_i / v_3 \quad (10)$$

Equation h_3 is used to compute the convective heat transfer coefficient.

$$Nu_3 = \frac{h_3 d_i}{\lambda_3} = 0.023\mathrm{Re}_3^{0.8}\ \mathrm{Pr}_3^{0.4} \quad (11)$$

The heat transfer coefficient within the circular route is derived using the following formula: equation (12)

$$Nu_2 = \frac{h_2 d_h}{\lambda_2} = 0.023\mathrm{Re}_2^{0.8}\ \mathrm{Pr}_2^{1/3} \quad (12)$$

$$\mathrm{Re}_2 = v_2 d_h/\upsilon_2 \quad (13)$$
$$d_h = D_i - d_o \quad (14)$$
$$v_2 = m_2 s/(\rho_2 N_t sp_2) \quad (15)$$

$$sp_2 = \frac{\pi}{4}\left(D_i^2 - d_o^2\right) \quad (16)$$

In the shell, the convective heat transfer coefficient is given by:

$$h_1 = \frac{\lambda_1}{n} 0.36\mathrm{Re}_1^{0.55}\ \mathrm{Pr}_1^{1/3}\ (\mu_1/\mu_w)^{0.14} \quad (17)$$

Equivalent diameter for the heat transmission coefficient in the shell is estimated using an aligned configuration.

$$D_e = \frac{1.27}{D_0}\left(s_t^2 - 0.785D_o^2\right) \quad (18)$$

For staggered arrangement:

$$D_e = \frac{1.10}{D_0}\left(s_t^2 - 0.785D_o^2\right) \quad (19)$$

For Outside the concentric two tubes, the Re_1 number is given as:

$$Re_1 = v_1 D_e/v_1 \quad (20)$$

$$v_1 = m_1/(\rho_1 sp_1) \quad (21)$$

$$m_1 = m_f/2 \quad (22)$$

$$sp_1 = \frac{(s_t - D_o)ED_s}{s_t} \quad (23)$$

Here between annular channel and the middle tube, the entire factor of energy transfer is given by:

$$K_{2,3} = \frac{1}{\frac{1}{h_2} + R_{\mathrm{fool},2} + \frac{d_0}{d_i}\left(R_{\mathrm{fool},3} + \frac{1}{h_3}\right)} \quad (24)$$

Here between annular channel and the shell, the total heat transfer coefficient is given by:

$$K_{1,2} = \frac{1}{\frac{1}{h_1} + R_{\mathrm{fool},1} + \frac{D_0}{D_i}\left(R_{\mathrm{fool},2} + \frac{1}{h_2}\right)} \quad (25)$$

Objective function:
Current total cost

$$C_{\mathrm{tot}} = C_{\mathrm{inv}} + C_{0D} \quad (26)$$

Capital cost

$$C_{\mathrm{inv}} = a_1 + a_2(s_{1,2} + s_{2,3})^{a3} \quad (27)$$

$$s_{1,2} = N_t \pi D_o L \quad (28)$$

$$s_{2,3} = N_t \pi d_o L \quad (29)$$

Total discounted operating cost

$$C_{0D} = \sum_{k=1}^{ny} \frac{C_0}{(1+i)^k} \quad (30)$$

$$CO = PC_E H \quad (31)$$

$$P = \frac{1}{\eta}\left(\frac{m_1 \Delta P_1}{\rho_1} + \frac{m_2 \Delta P_2}{\rho_2} + \frac{m_3 \Delta P_3}{\rho_3}\right) \quad (32)$$

The pressure fall in the inner tubes, annular channel, and shell are estimated total frictional pressure fall and pressure decreases owing to the heat exchanger entrance and departure.

$$\Delta P_3 = \Delta P_{\mathrm{friction}} + \Delta P_{\mathrm{singular}} = \frac{\rho_s v_3^2}{2}\left(\frac{L}{d_i} f_i + p\right) s \quad (33)$$

where f_t is the Darcy [4] friction coefficient, given by

$$f_t = (1.82 \log_{10}\mathrm{Re}_3 - 1.64)^{2} \quad (34)$$

In the literature, many values of the constant p are mentioned. p = 2.5 is used by Sinnott[18]. The following phrases can be used for tubes with an annular passage section: Pressure drop due to friction

$\Delta P_{friction} = f_t \frac{L}{d_2} \frac{\rho_2 v_2^2}{2}$

Darcy's coefficient is calculated as follows:

$f_t = k_a f$

Here, k_a can be estimated by Idelcik. The Blasius relation[19] gives Darcy's coefficient, f:

$f = 0.3164\ Re_2^{-0.25}$ for $2300 \le Re_2 \le 10^5$

$k_a = 1.37$

and that of Herman[24],

$f = 0.0054 + 0.3964\ Re_2^{-0.30}$ for

$10^5 \le Re^2 \le 10^6$

$k_a = 1$

Singular pressure fall

$\Delta P_{singular} = (3/2)\ \rho_2\ v_2^{\ 2}$

The inflow and outflow cause a pressure drop and is given by

$\Delta P_{entrance} = (3/4)\ \rho_2 v_2^{\ 2}$

Pressure drops in the annular passage

$\Delta P_2 = \Delta P_{friction} + \Delta P_{singular} + \Delta P_{entrance}$

For the shell side

$$\Delta P_1 = \frac{\rho_1 v_1^2}{2} \frac{L}{E} f_s \frac{D_s}{D_e}$$

6.0 Results and Discussions

6.1 Validation

To authenticate the one of the findings confirmed the method utilised with Caputo *et al.*[6] Table 2. The same correlations were done by calculating the HT area, with the exception of the transmission of thermal coefficient of the inner tube side and the N_t, which were calculated using Selbas *et al*[12]. To achieve our objective, a TLBO Algorithm was used with the same number of initiations and the same systems of selection (selection roulette)[6].

The delineation framework of each of medium is given in the Table 1. They are utilized as input characteristics for the modeling programme to discover the optimal heat exchanger architecture that meets the objective function's minimization criterion.

The acquired findings are collated to those in the literature (Table 2) in order to determine the advantages of this novel form of HE. The following are the boundaries (minimum and maximum) of the optimization variables: The outer between 0.002 and 0.008 m; the shell interior dia D_s ranges across 0.1 and 1.5 m; the exterior tube outside dia D_o ranges across 0.01 and 0.05 m; the baffle spacing E have range between 0.05 and 0.5 m. Small tubes (evaporator and condenser) are often employed by the regelation business, and they can be supported within exterior separators in the tube. All operational cost numbers were estimated using the below assumptions: ny = 10 years, I = 10% annual rate of updating, CE = 0.12 €/kWh energy cost, and H = 7000 h/year annual operating time.

Case#1 Methanol-seawater. Caputo *et al.*[6] investigated this instance, and the heat exchanger's basic design of

Table 1. Specifications[8]

	Mass Flow (kg/s)	T input (°C)	T ouput (°C)	ρ (kg/m³)	C_p (kJ/kg.K)	μ (Pa.s)	Λ (W/m.K)	*Rfouling* (m².K/W)
Shell side: methanol	27.80	95.0	40.0	750	2.84	0.00034	0.19	0.00033
Tubeside: sea water	68.90	25.0	40.0	995	4.20	0.00080	0.59	0.00020

dual avenues on the tube side and one passage on the shell side was maintained. Due to the shorter tube lengths, the HT area was minimized, but the shell diameter was somewhat raised. The tube numbers with smaller diameters grew dramatically. This novel construction increased the pressure fall on tube and shell side without negatively impacting the operating costs. The capital cost also decreases of about 43% with respect to traditional heat exchanger and 34% with GA based STHE.

7.0 Discussion

An advanced type of HE that is found by a optimisation is more profitable and economic this optimisation helps to design a heat exchanger that satisfies our objective function. Here when compared to typical (traditional) heat exchangers the overall cost has dropped by around 43% and 34% with GA based shell and dual concentric tube HE considering both inline and staggered arrangements.

The design moderation and the use of TLBO Algorithm and its optimization was beneficial in economic point of view, as hot fluid comes in contact with both cold fluids, the HT area is increased by a factor of one and thanks to TLBO algorithm to achieve our objective. The conventional and indicated class of ratio given (0.5 L/D 15) is used to decrease pressure fall in the HE, but considering in this work, the goal function considers the overall cost in economic point of view considering HE, including pressure fall, and yields the following results when compared to the conventional range (0.5 L/D 15) (Table 2), that is L/D value of 0.989 with inline arrangement and 1.176 with staggered arrangement. However, the TLBO Algorithm competence was demonstrated when it was used to design the shell and tubes.

8.0 Conclusion

In this work the TLBO algorithm is effectively applied in the design of STHE and the performance of the HE length is greatly regulated by the tube radii that form the HE. As contrasted to a STHE with the same external tube diameter and shell diameter, optimising a shell-and-dual concentric tube HE longitudinally saves a significant amount of area and resource. The TLBO Algorithm is effectively employed to design the HE of shell and dual concentric tubes in this study. The outcomes are collated to those reported in the literature. Comparing with traditional STHE, the total operating costs have significantly decreased, that is in comparison to typical heat exchangers, and the overall cost has dropped by around 43% and 34% with GA based shell and double concentric tube HE with respect to both inline and staggered arrangements.

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Nomencleature

a_1	Constant (€)
a_2	Constant (€/m^2)
a_3	Constants
c	Constants
c_p	Specific heat capacity (J/kg K)

C_{inv}	Capital investment (€)
C_E	Cost of energy (€/kW h)
C_o	Annual cost of operation (€/year)
C_{OD}	Total operating cost at a concession (€)
C_{tot}	Total cost per year (€)
d	Internal tube diameter (m)
d_h	Hydraulic diameter (m)
D	External tube diameter (m)
D_e	Equivalent shell diameter (m)
D_s	Inner diameter of the shell (m)
E	Baffles spacing (m)
f_t	Tube side Darcy friction factor
f_s	Factor of friction on shell side
F	Logarithmic mean temperature difference emendation factor
h	Coefficient of convective heat transfer (W/m^2 K)
H	Working hour on an annual basis (h/year)
i	Discount rate on an annual basis (%)
K	Overall heat transfer coefficient (W/m^2 K)
L	Length of tubes (m)
m	Mass flow rate (kg/s)
n	Constants
n_y	Life expectancy of equipment
Nu	Nusselt Number
N_t	Number of tubes
P	Pumping energy (W)
PR	Prandtl number
Rfoul	Resistance to fouling (m^2 K/W)
Re	Reynolds number
Sp	Area to pass through (m^2)
s	The total numeral of passes

st	Tube pitch (m)
S	Surface area for heat exchange (m^2)
T	Temperature of the fluid (°C)
v	The velocity of the fluid (m/s)
V	Volume of the heat exchanger (m^3)
Greek symbols	
ΔP	Drop in pressure (Pa)
ΔTML	Mean logarithmic temperature Difference (°C)
η	Efficiency of pump in general
λ	Thermal conductivity (W/m K)
μt	Viscosity at tube wall temperature (Pas)
μw	Viscosity at core flow temperature (Pas)
v	Kinematic viscosity (kg/s m)
π	a mathematical constant
ρ	Fluid density (kg/m^3)
Φ	Heat duty (W)
Indices.	
1	Shell
2	Annulus
3	Central tube
f	Cold fluid
i	Inside
o	Outside
HE	Heat Exchanger
STHE	Shell and tube heat exchanger