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

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Abstract

Heat exchangers are used to carry heat energy between two or more fluids. Different forms of heat exchangers are used in various applications. Here we are considered for optimization of shell and double concentric tube heat exchanger. This kind of heat exchangers are used in many years for their satisfactory services for industry, availability of set of symbols and excellence for design, modelling and are produced from different of materials. In this paper a novel computative technique called Jaya Algorithm is used and the aim is to lower the overall cost by designing the heat exchanger with a shell and two concentric tubes. The principle behind this strategy is that the best answer for a given problem should be sought, while the lowest standard approach should be avoided. It proves to be very efficient with a typical start and hence this algorithm would help to achieve our objective function that is in comparison to typical heat exchangers the overall cost has dropped by around 43% and 34% with GA based heat exchanger with a shell and dual concentric tube heat exchangers.

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

1.0 Introduction

There will be need for detailed and systematic research regarding decrease in total costs, increase in effectiveness of heat exchangers. Heat exchangers are devices that transmit thermal energy between two or more fluids. Heat exchangers are utilised in a variety of applications. The shell and tube heat exchanger are the most popular among the several varieties due to its well-established design and manufacturing techniques.

To meet the design restrictions, Geometric and operational factors abound in heat exchangers. A STHE is a type of heat exchanger that is built up of tube bundles compressed inside a broad cylindrical casing, or shell. A GA is powerful method for solving multivariable optimization problems. Recently many researchers have focused on the genetic algorithm is used in many heat exchanger optimization problems[1-3].

Costa *et al.*[4] based on a search of the tube count table, a design of functional most effective type of shell heat exchangers was proposed, with the purpose of reducing the heat surface area of the equipment. Kara *et al.*[5] employed a model to estimate the optimal heat transfer surface area for shell and tube heat exchangers with only monophasic fluid move on both the shell and tube sides. Caputo *et al.*[6], Baadache *et al.*[7,8] presented the shell and dual concentric tube heat exchanger and using the Genetic Algorithm, the design of heat exchanger with a shell and two coextensive tubes was developed.

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The study discusses the use of a GA to develop a heat exchanger with the goal of lowering the total cost with a shell and dual concentric tubes. Mishra *et al.*[9] is used genetic algorithm based optimization for cross-flow heat exchanger, where the objective function is minimization of number of entropy units. Ortega *et al.*[10] made use of a genetic algorithm to decrease the shell and tube heat exchangers' total operating cost at concession.

Hajabdollahi *et al.*[11] to obtain the minimum cost he used both GA and a particle swarm optimization technique that includes pumping energy. Many other researchers examined an exploration on shell and tube heat exchanger in productive sort of view.

Selbas *et al.*[12] and Babu *et al.*[13] to find the least heat transfer area they employed distinctive evolution advancement techniques with a genetic algorithm. In some studies the objective function is to help in lowering cost and region of heat transmission in a optimization problems[14–18]. Considering this research area t his is where the majority of the research has been done. Wang *et al.*[19], Rao and Patel[20] also used a GA and particle swarm to design better heat exchangers. Rao and Patel[21] employed a modified TLBO technique for model a best heat exchangers. They used this technique to minimise their cost functions when choosing variables for any of the shell and tube and plate fin heat exchangers. Tharakeshwar *et al.*[22] here for shell and tube heat exchangers, multifunctional optimization of bat algorithm is used in this work and it considers a multivariate province such as competence and total cost optimisation are carried out. Veerabhadrappa *et al.*[23] Used Cuckoo Optimisation Algorithm (COA) technique has been evolved for heat exchanger of shell and dual concentric tubes. The heat exchanger with a shell and two concentric tubes will be sized as part of the optimization programme.

The optimization approach begins with the selection of key geometrical factors such as tube numerals, interior and exterior tube diameters, baffle gradient, and so on. The technique considers the measurable and operational restrictions that are commonly advised by design regulations. The cases examined illustrate that the Jaya algorithm is a valuable tool for heat exchanger ideal design[24].

The Jaya algorithm is used to tackle heat exchanger optimization issues with several design factors. However, it is necessary to look into the possibility of using non-traditional optimization strategies. One such strategy is the Jaya algorithm. A research study was provided to illustrate the capability of the suggested technology, demonstrating that significant cost savings are possible when compared to the conventional design of shell and tube heat exchanger.

There are many handbooks available that cover the design of shell-and-tube heat exchangers. Schlunder, Hewitt, Saunders, and Shah, as well as Sekulic, compiled a collection. These sources of knowledge on heat exchanger delineation, particularly for shell and tube heat exchangers, are suggested. After review of various literature mainly by A C Caputo, optimization of outer diameter of the outer tube distance between baffles and outer diameter of inner tube to meet our objective function seem suitable. The default values which will be used in the beginning for optimization will be taken from publications from Caputo and is compared between this heat exchanger.

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 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, 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 is as follows:

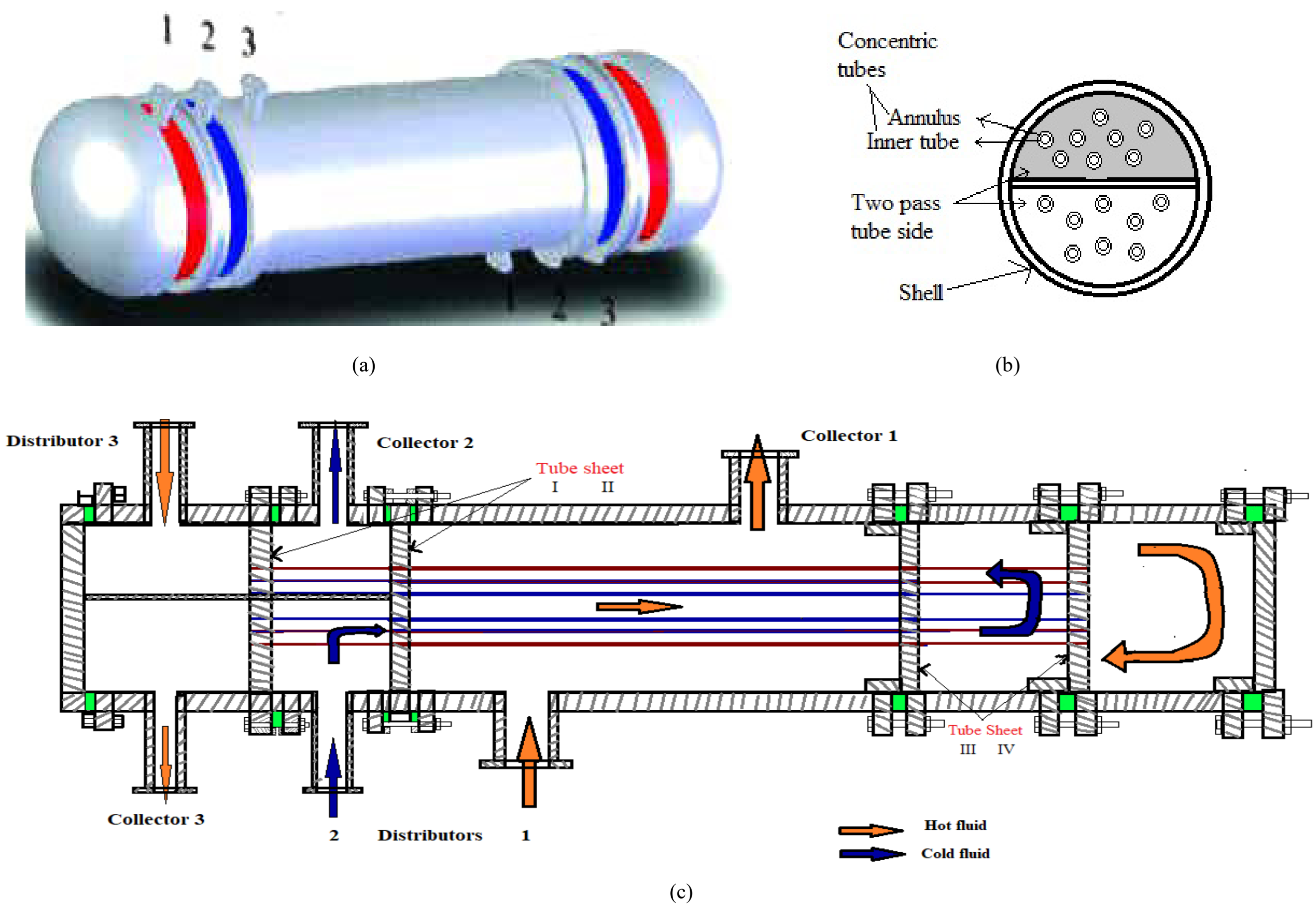

Figure 1. **a.** Diagrammatic portrayal of 3-D shell-and-dual concentric-tube heat exchanger[7]. **b.** Diagrammatic portrayal of shell-and-dual concentric-tube heat exchanger in top view. **c.** A diagrammatic view of shell-and-dual concentric-tube heat exchanger in crosses sectional view.

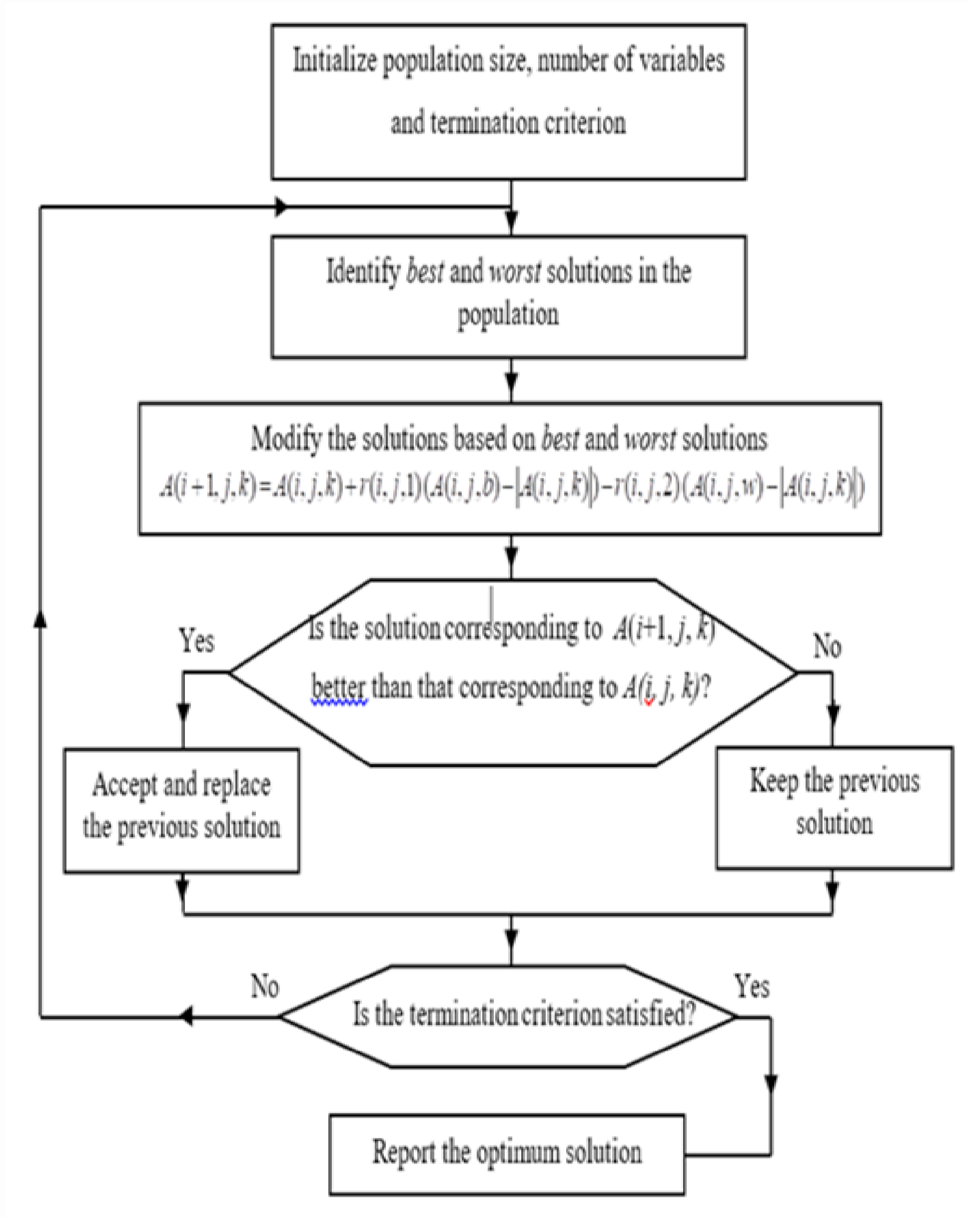

Figure 2. Algorithm of proposed program.

- Studying the literature of the previous done work on Heat Exchangers. Analysing and understanding their problem definitions and problem-solving approach.
- By considering a set of design factors, the logarithmic mean temperature difference approach is used to compute the heat transfer area.
- Investment, operational cost, and objective function evaluation.
- Using an optimisation method to choose a novel set of the layout feature's virtues.
- Iteration of the preceding phases until the target function is completed is reduced to its simplest form.

The whole procedure is catalogued in Figure 2: The mass flow of each fluid, inlet and outlet temperatures of each fluid, the models of the 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), are shown 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: tube side, annular passage side, and shell side h_1, h_2 and h_3 respectively, 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 multiple of the baffle distance "E," with the leftover total of the baffle's width determines the gap. The objective function is then estimated using the set on values from the velocities of flow and the productive features of the heat exchanger construction. The Jaya algorithm fixes the trial assess of the escalation circumstances created on the value of the goal function, which is then used to create a new decision (goal) function that describe new heat exchanger design. The technique is repeated until the goal function's minimum is obtained. Description of Algorithm Jaya algorithm is an algorithm for refinement without gradients. It can be used for Maximization or A function's minimization. It is a population-based approach in which it is able of solving both limited and incontinent optimization problems by continually modifying a population of individual solutions. The premise behind this strategy is that the ideal answer to a problem must be sought, while the thrash alternative should be eschewed. This algorithm just needs the standard control framework and none of the algorithm-specific control framework.

All the transformative and swarm intelligence algorithms are deterministic and need the same regulating framework, such as amount initiation, population expanse and best size. Each algorithm necessitates its very own algorithm certain control specifications in addition to the regular command framework. The algorithm is trying to move approach to achieving goal while avoiding failure. Jaya is the name of an algorithm that seeks to be triumphant by finding the right choice (a Sanskrit word meaning victory).

3.1 Jaya Algorithm

Let f(x) be the minimised (or maximized) goal/achieving function. Consider 'm' number of design variables (i.e. j = 1, 2,..., m) and 'n' values of possible solutions (i.e. k = 1,2,...,n) at any progression let the finest applicant get the prime f(x) value (i.e. f(x) best) applicant in its entirety solutions, and the weakest one receives a worse outcome. f(x) value (i.e. f(x) worst) across the complete postulant solutions hence finally moving towards best solution. shown in Figure 2.

4.0 Mathematical Modelling

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

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

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

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

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

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:

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

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

$$m_3 = m_f / 2 \tag{9}$$

The Reynolds number is determined by:

$$Re_3 = v_3 d_i / v_3 Re_3 = v_3 d_i / v_3 \tag{10}$$

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

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

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

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

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

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

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

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

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

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

For staggered arrangement:

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

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

$$Re_1 = v_1 D_e / v_1 \tag{20}$$

$$v_1 = m_1 / (\rho_1 sp_1) \tag{21}$$

$$m_1 = m_f / 2 \tag{22}$$

$$sp_1 = \frac{(s_t - D_o) E D_s}{s_t} \tag{23}$$

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

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

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

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

Objective function:

Current total cost

$$C_{tot} = C_{inv} + C_{0D} \tag{26}$$

Capital cost

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

$$s_{1,2} = N_t \pi D_o L \tag{28}$$

$$s_{2,3} = N_t \pi d_o L \tag{29}$$

Total discounted operating cost

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

$$CO = P C_E H \tag{31}$$

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

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.

$$\Delta P_3 = \Delta P_{friction} + \Delta P_{singular} = \frac{\rho_3 v_3^2}{2}\left(\frac{L}{d_i} f_i + p\right) s \tag{33}$$

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

$f_t = (1.82 \log_{10} Re_3 - 1.64)^{2}$ (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

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

5.0 Results and Discussions

5.1 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 heat transfer area, with the exception of the transmission of heat coefficient of the inner tube side and the number of tubes, which were calculated using Selbas *et al*[12]. To achieve our objective, a Jaya 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 utilised as input characteristics for the modelling 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) to determine the advantages of this novel form of HE. The following are the boundaries (minimum and maximum) of the optimization variables: The outer diameter of internal tube d_o ranges between 0.002 and 0.008 m; the shell interior diameter D_s ranges between 0.1 and 1.5 m; the exterior tube outside diameter D_o ranges between 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 in the regelation business, and they can be supported within the exterior separators in the tube. All operational cost

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

Table 2. Validation and comparision of other algorithms

	Case study Shell and tube heat exchanger (literature[6])	Shell and double concentric tube heat exchanger (GA) (literature[8])	Shell and double concentric tubes heat exchanger (COA) [23]	Shell and double concentric tube heat exchanger (this work validation) [Alligned arrangement]	Shell and double concentric tubes heat exchanger (this work) [Staggered arrangement]
do	–	0.005	0.005	0.0045	0.0045
Do	0.0160	0.0125	0.017	0.0160	0.0164
Ds	0.830	0.982	1.022	1.0108	0.8500
E	0.500	0.200	0.25	0.25	0.2500
Nt	1,567	3624	1,147	1,409	1,428
St	0.0200	0.0156	0.006	0.0056	0.0056
De	0.0110	0.0089	0.014	0.0134	0.0143
v_3	–	0.9261	2.57	2.5841	2.55
Re_3	–	7660.7	12684	20521	20,251
Pr_3	–	5.08	5.08	5.0821	5.0821
h_3	–	2859.5	4388	6,552	6,482.5
v_2	0.69	0.65	0.87	0.8740	0.8078
Pr_2	5.7	5.7	5.7	5.6949	5.69
d_h	–	0.0050	0.009	0.0083	0.0087
Re_2	10,936	4,038	6960	9,004	8,698
h_2	3,762.0	3,718.6	4141	4,238.6	3,945
v_1	0.44	0.47	0.39	0.0398	0.453
Re_1	11,075	9,238	1295	1,179.1	1,434.7
Pr_1	5.1	5.1	5.082	5.0821	5.0821
h_1	1,740	2,010	1989	426.35	444.14
$K_{2,3}$	–	727.2	952	962.26	944.46
$K_{1,2}$	–	691.5	298	310.52	317.62
K	660.0	-	-	599.8	–

L	3.379	1.103	1.00	1.00	1.00
$S_{1,2}$	–	156.82	63.93	70.7457	73.78
$S_{2,3}$	–	62.72	55.02	19.9224	20.19
S	262.8	218.6	168	-	-
ΔP_3	–	6,419.7	4043.743	48,615	47,462
ΔP_1	13,267.0	18,638.0	78	89.4162	88.41
ΔP_2	4,298.0	5,974.3	4202	4,573.3	3,822
C_{inv}	49,259	43,030	22289	23,665	24,183
C_{OD}	5,818	4,796.2	7117	7,867	7,393.3
C_{tot}	55,077	47,826	29406	31,532	31,575

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 dual avenues on the tube side and one passage on the shell side was maintained. Due to the shorter tube lengths, the heat exchange surface 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 shell and dual concentric tube heat exchanger.

6.0 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 Jaya Algorithm and its optimization was beneficial in economic point of view, as hot fluid comes in contact with both cold fluids, the heat transfer area is increased by a factor of one and thanks to jaya 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 in our case, the goal function considers the overall cost of the heat exchanger, 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 Jaya Algorithm competence was demonstrated when it was used to design the shell and tubes.

7.0 Conclusion

In this work the Jaya algorithm is effectively applied in the design of shell and dual concentric tube heat exchangers and the performance of the HE length is greatly regulated by the tube radii that form the heat exchanger. As contrasted to a shell-and-tube HE with the same external tube diameter and shell diameter, optimising a shell-and-dual concentric tube heat exchanger longitudinally saves a significant amount of area and resource. The

Jaya Algorithm is effectively employed to design the heat exchanger 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, the overall cost has dropped by around 43% and 34% with GA based shell and double concentric tube heat exchanger with respect to both inline and staggered arrangements.

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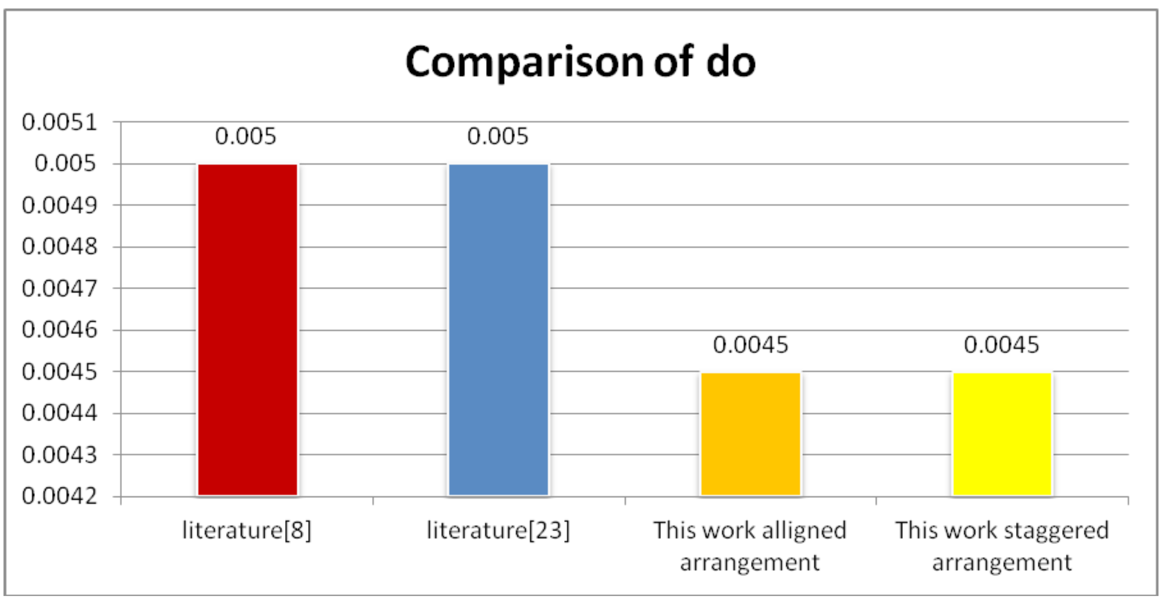
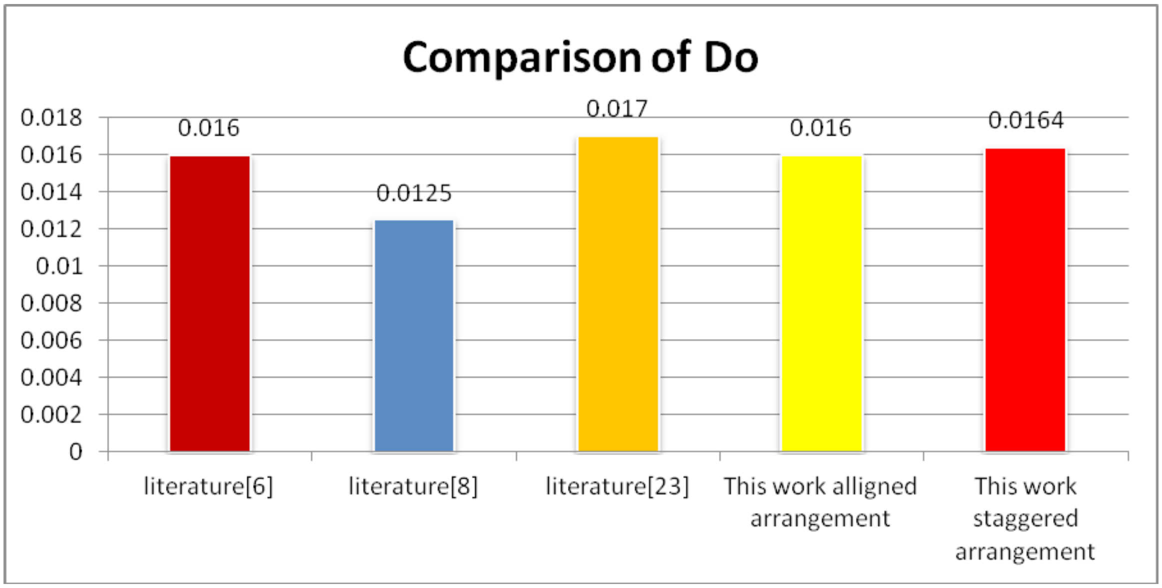
Appendix :

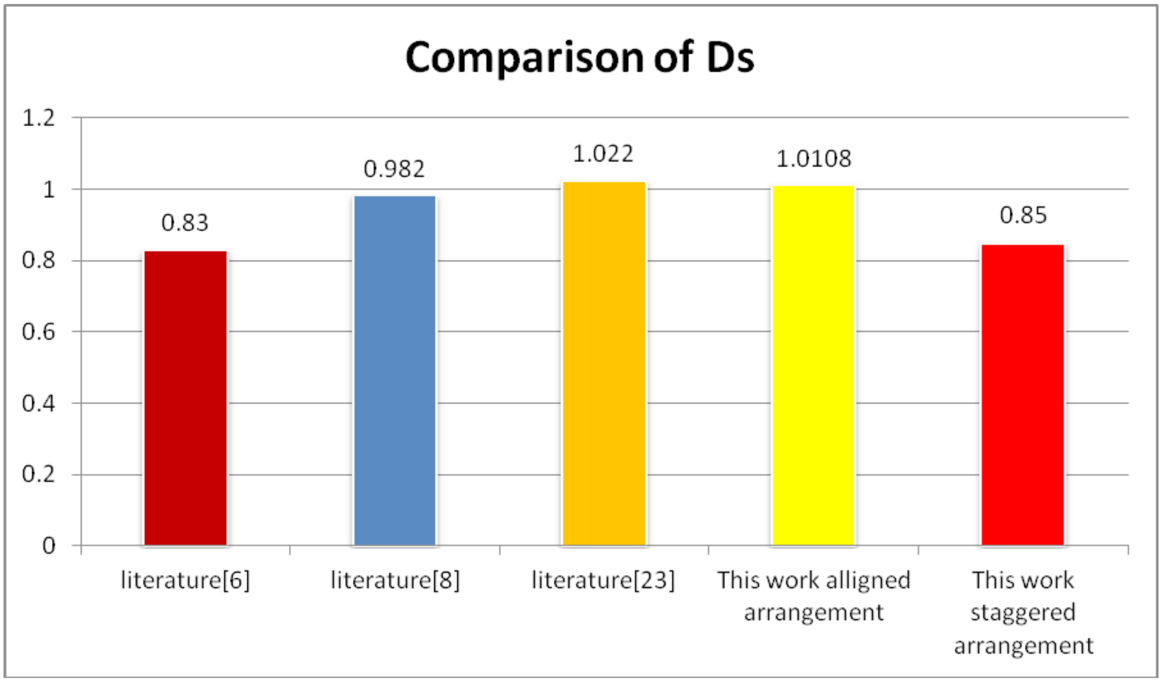
GRAPHS



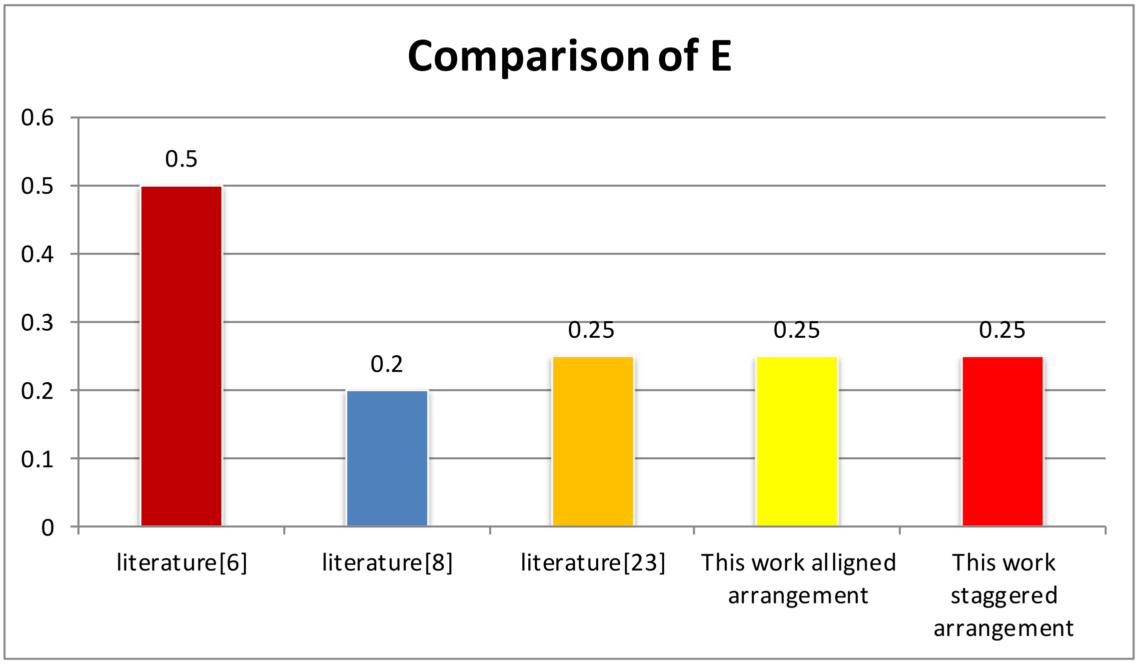
While comparing the outer diameter of the internal tubes of all cases, (single tube in case of of literature[8], and with literature[23]) it was observed that cross section of the literature[7] and [23] is slightly bigger compared to all others, with Jaya – Inline and Staggered at 0.0045 m in diameter



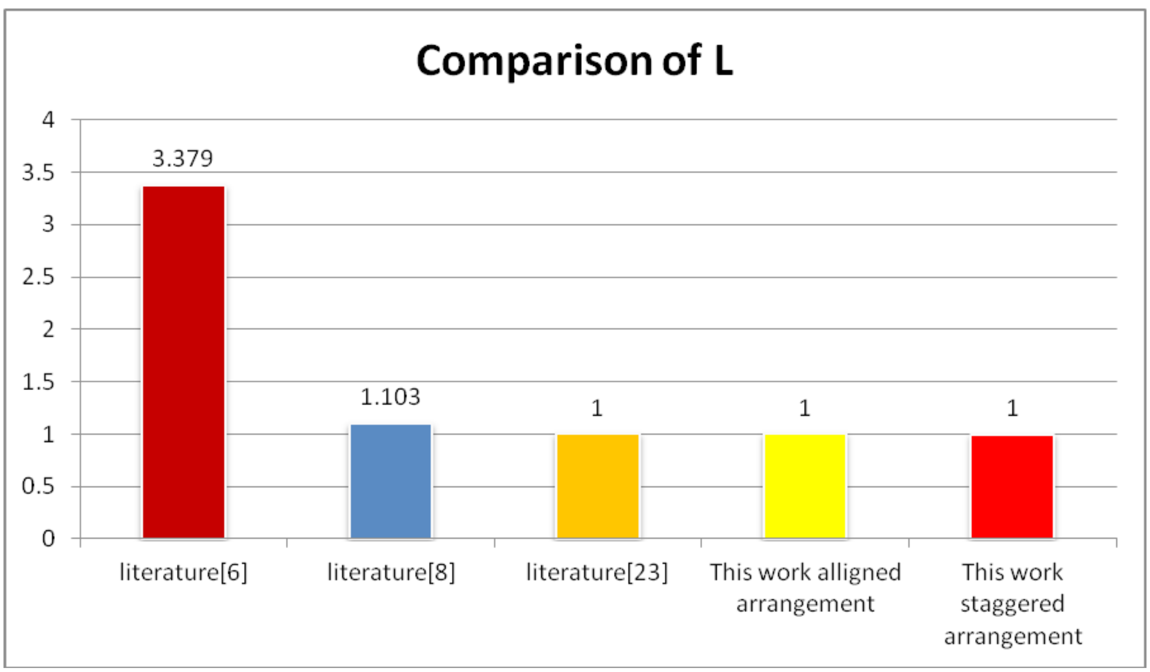
While comparing the outer diameter of the external tubes of all cases, (single tube in case of literature[6], literature[8], and with literature[23]) it was observed that cross section of the literature[6] and [23] is slightly bigger compared to our work except literature[7], with Jaya – Inline at 0.016 and staggered at 0.0164 m in diameter



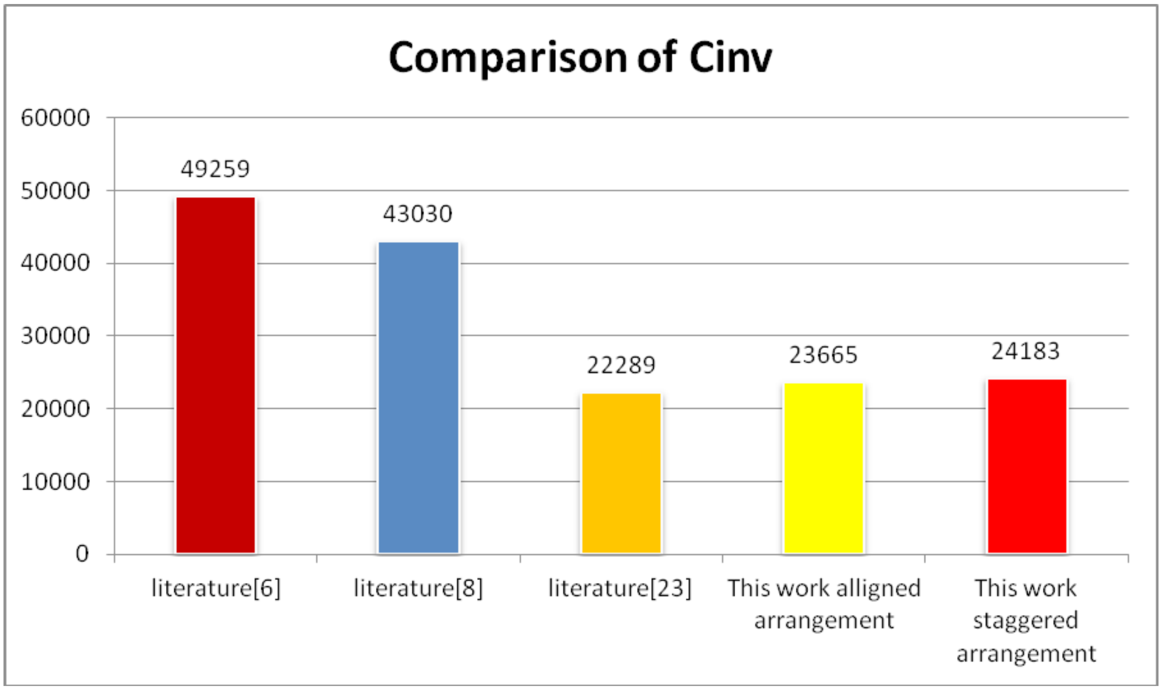
When comparing the shell outer diameter, the highest value obtained was that of literature[23] at 1.022 m, literature[6] at 0.830m lowest and with Jaya – Inline at 1.0108m and staggered at 0.850m in diameter.



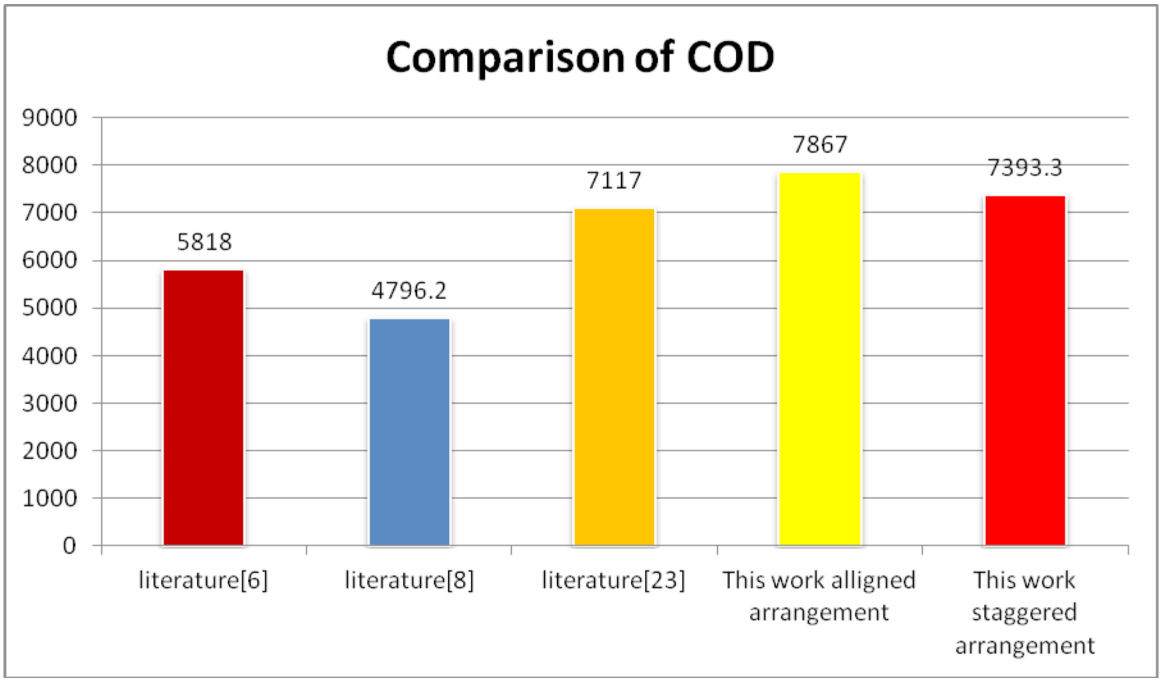
It was observed that the baffle spacing was highest in literature[6] with 0.5m, lowest in literature[8] with 0.2 m and literature[23] and our work with 0.25m. Baffle spacing has been reduced by 50 % in our work compared to literature[6].



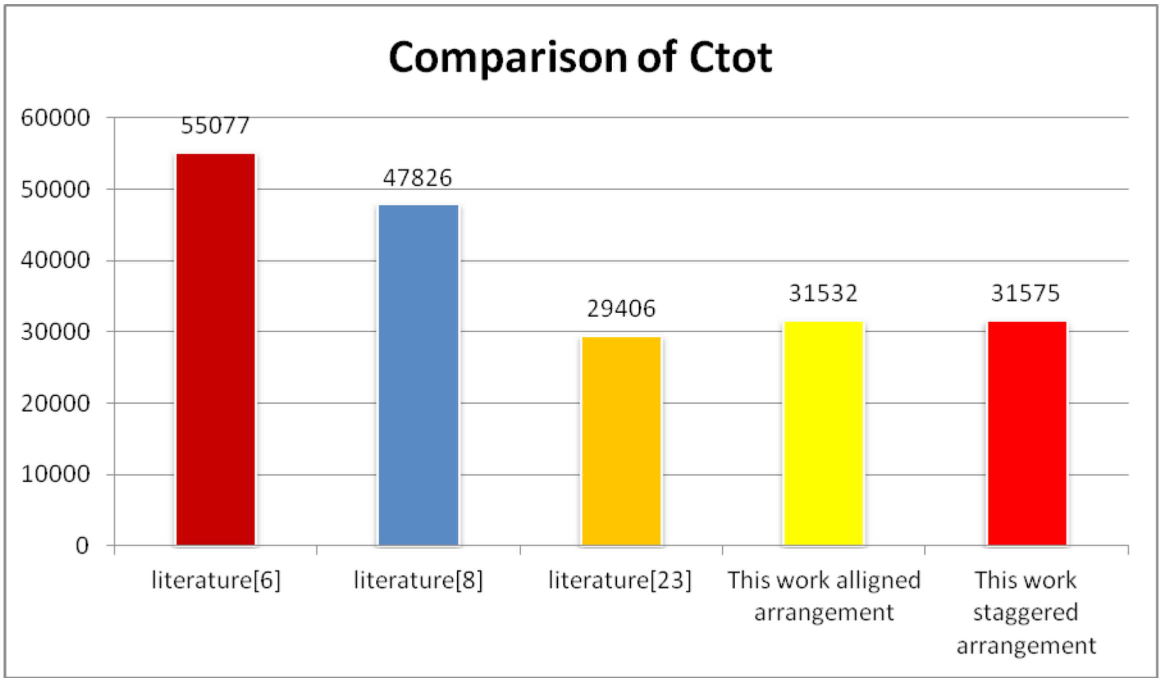
While comparing the length it was observed that the length was reduced in our work at 1m with respect to literature [6] at 3.379m, literature [8] at 1.103m and was same for literature [23] at 1m.



It was observed that the capital investment was highest in literature [6] and [23] compared to our work, however literature [23] was slightly less compared to all others.



Total discounted operating cost of our work was higher compared to other literature works; however, the total annular cost was decreased as we can see in the other graph.



Total operating cost was found to be reduced compared to that of literature [6], literature [8], and with literature [23] a slight increase with respect to both inline and staggered arrangement.

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