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Thermoeconomic Lifecycle Cost Optimization of an Annular Fin Heat Exchanger

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Abstract

In this paper the design of annular fin heat exchanger based on economic optimization has been carried out. The optimization process targeted minimizing the lifecycle cost of annular fin heat exchanger that has the same frontal area, effectiveness and heat load of available practical standard geometry exchangers. The lifecycle cost includes both capital and operating costs. Beside the pumping cost, both the cost of exergy destruction due to irreversibilities and 10% inflation rate are included in the operating cost. The optimization process is implemented using Evolutionary Algorithm (EA). Evolutionary Algorithm is a numerical technique which is initiated by randomly generating a set of possible solutions. The optimized design has shown a significant decrease in the lifecycle cost as compared with that of standard geometry that has minimum lifecycle cost. Based on the optimized design relations for Colburn and friction factors are developed.

Key Words: Tube-fin Heat transfer, Exergy, Lifecycle cost, Evolutionary algorithm, Optimization

1. Introduction

Tube fin heat exchangers are employed for the exchange of heat between a liquid and a gas. The poor heat transfer coefficient on the gas side of the liquid carrying tubes is improved by using fins on the periphery of the gas side of tubes. Such heat exchangers have wide applications in automobile radiator, HVAC systems, and refrigeration. Two clear examples of such exchangers can be identified; the first is the radiator of motor vehicles where the hot water moves inside the radiator tubes and the forced air, which is the cooling fluid, moves outside across the tubes. While the second is the heat recovery from hot gases in which hot gases move across the exchanger tubes while cold liquid moves inside the tubes. The latter exchanger is a cross flow heat exchanger with one fluid mixed and other unmixed. These types of heat exchangers are the most successful type of heat exchangers.

The design of heat exchangers in general, requires consideration of the heat transfer occurring between the two fluids, the mechanical energy needed to overcome the frictional resistance through the heat exchanger and the useful energy destroyed (exergy destruction) due to entropy generation. The optimum design criteria of a heat exchanger should aim to achieve large heat transfer, small pressure drop (i.e small frictional resistance) and minimum exergy destruction. Moreover, this criterion should take the total lifecycle cost into consideration.

The issue of optimum design of a heat exchanger has been considered by a number of authors. Design optimization based on analytical consideration to get optimum design of a central heating radiator by varying geometrical and thermal parameters of the radiator has been done by Arslanturk and Ozugue [1]. In their design the cost optimization based on capital and operating cost has been neglected. Analytical technique is used to optimize the surface area and pumping power required for a tube-fin heat exchanger as in shown by Charyulu [2]. The study considered the effect of different materials used in the radiator like copper fins on copper tubes, brass and carbon steel tubes. However, the lifecycle cost of radiator has not been taken into account. A new model for simulating air-to-refrigerant fin-and-tube heat exchangers, with arbitrary fin sheet having varied geometric parameters has done by Singh and Aute et al.[3]. Their design has not considered the standard tube configuration but rather was based on any arbitrary configuration of tubes and fins. Their optimized design considered the surface area and pressure drop and neglected the operating cost that based on cost and exergy destruction. Some people optimize annular-finned tube heat exchanger by maximizing the heat transfer coefficient using finite difference method as shown by Chen et al.[4]. Their optimization has neglected the lifecycle cost. An optimum dimension of fins for the tube-fin heat exchanger for rectangular and equilateral triangular arrays of tubes has been done by Kundu et al [5]. The optimization has been done by using conventional analytical methods to maximize the heat transfer and minimize the surface area. The operating cost that includes pumping cost and exergy destruction costs has not been considerer in their optimization. Optimized dimensions of space radiators have been found by Arslanturk [6] which maximize the heat transfer rate per unit radiator mass. The lifecycle cost of radiator has not been considered in the optimized design. Taguchi method is used to carry out parametric study to optimize heat exchanger design is given by Gang et al.[7]. They selected fifteen samples from experimental database and optimized the design based on flow depth, ratio of fin pitch and fin thickness and number of louvers. Their optimization considered minimum surface area and pumping power but excluded the exergy destruction cost

From the previous review it can be concluded that different authors attempted to optimize the tube-fin heat exchanger based on surface area and pumping power required to overcome the frictional losses. Their analysis did not include the exergy destruction cost; and inflation effect over the lifecycle cost. In this paper annular fin heat exchanger lifecycle cost is optimized using evolutionary technique. Optimization is carried out for different geometry parameters like tube outside diameter d_o, flow passage hydraulic diameter dh, fin length L_f, fin pitch P_f, etc Lifecycle cost includes both capital and operating cost with inflationary effect over the life of heat exchanger. Capital cost include material cost and manufacturing cost, where as, operating cost include pumping cost and exergy destruction cost.

2. Problem Statement

The annular heat exchanger under consideration is a waste heat recovery radiator in which the cold liquid water is moving inside the radiator tubes while waste hot combustion gases are moving outside the tubes. The heat exchanger is a cross flow type in which one fluid (combustion gases) is mixed and the other (water) is unmixed. The layout of the heat exchanger is shown in Fig. 1 while the arrangement of tubes within the exchanger is shown in Fig. 2. The shown exchanger shares some common parameters with well known 12 standard geometry exchangers presented in London and Kays [8]. These parameters are the material of the exchanger (tubes and fins), frontal area, mass flow rate of the two fluids, inlet and outlet temperatures of the cold fluid (water) and the inlet temperature of hot fluid (combustion gases). These parameters and their values are shown in Table 1. From the given data and based on the energy balance over the heat exchanger assuming no heat loss from the whole exchanger, the thermal load and outlet gases temperature can be calculated as

$$q = \dot{m}_i C_{pw}(T_{C,o} - T_{C,in}) \tag{1}$$

$$T_{H,o} = T_{H,in} + \frac{q}{\dot{m}_o C_{pg}} \tag{2}$$

The specific heat of gases (Cp) is taken initially at the given inlet gases temperature and then the outlet gases temperature is predicted using Equation (2). Fluid properties like density (ρ), specific heat constant pressure (Cp), thermal conductivity (k), Prandtl Number (Pr) etc are calculated at mean gases temperature.

The effectiveness of heat exchanger is calculated as:

$$\varepsilon = \frac{q}{q_{\max}} = \frac{\dot{m}_i C_{pg}(T_{C,in} - T_{C,o})}{\dot{m}_o C_{pw}(T_{H,in} - T_{C,in})} \tag{3}$$

From the above analysis it is clear that the exchanger under consideration has the same effectiveness and heat load of the 12 standard common geometry exchangers given in [8]. The other design parameters which differ between these exchangers and need to be optimized for minimum lifecycle cost are free flow area/frontal area (σ), heat transfer area/total volume (α), flow passage hydraulic diameter (dh) , tube outside diameter (do), fin length (L_f), fin pitch (P_f), fin thickness (t), fin area/total area (A_f/A), center of center distance between tubes in vertical direction (h), center to center distance between tubes in the depth direction (d). These design parameters will be calculated for the heat exchanger under consideration according to the design procedure outlined in the following section.

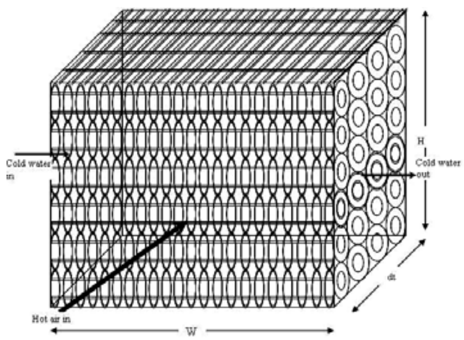


Fig. 1: A view of annular fin heat exchanger

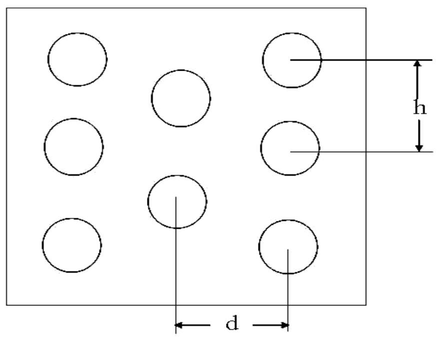


Fig. 2: Side view of heat exchanger to show tube arrangement

Table 1: Common parameters between present exchanger and 12 standard geometries given in [8]

Height of exchanger	H = 0.335 m
Width of exchanger	W = 0.6 m
Thermal conductivity of tubes (Al)	k = 237 W/m K
Thermal conductivity of fins (Al)	k = 237 W/m K
Inside/Cold Fluid	Water
Outside/Hot Fluid	Air
$\dot{m}_i$ cold fluid mass flow rate	1 kg/s
$\dot{m}_o$ outside fluid mass flow rate (hot gases)	1.25 kg/s
Water (cold fluid) inlet temperature	$T_{C,in}$ = 290 K
Water (cold fluid) outlet temperature	$T_{C,o}$ = 370K
Gas (hot fluid) inlet temperature	$T_{H,in}$ = 825 K

2.1 *Design Methodology*

The design methodology considered in this paper is based on ε-NTU approach for cross flow annular fin heat exchanger with one fluid mixed and one fluid unmixed. The heat capacity ratio C_R is first calculated as

$$C_R = \frac{C_{\min}}{C_{\max}} \tag{4}$$

For the given heat exchanger with inside fluid (water) has larger capacity C_{max}, the NTU is calculated as [11]

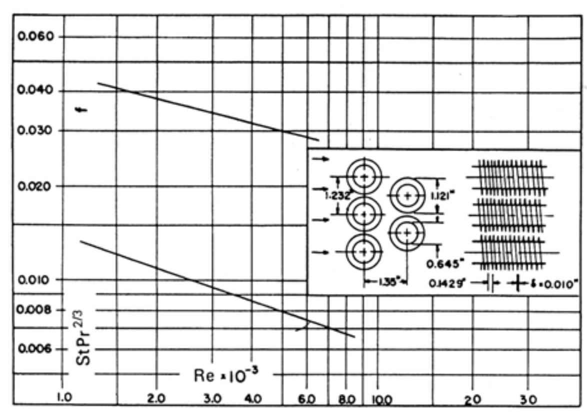
$$NTU = -(1/C_R)\ln[C_R \ln(1-\varepsilon)+1] \tag{5}$$

To proceed the calculation of the design parameter the Colburn factor, j and Friction factor *f*, are to be determined. Both j and f are estimated from the following relations which are function of Reynolds number [8];

$$j = C_j \operatorname{Re}^{mj} \tag{6}$$

$$f = C_f \operatorname{Re}^{mf} \tag{7}$$

The above relations for j and f for every standard geometry are plotted as a straight line on a log-log chart. Fig.3 shows these plots for one of these standard geometries. Below the chart (Fig. 3) are listed some of the geometrical data related to that particular geometry. Similar charts for the rest of standard geometries are given in [8]. The average values of *j and f* for each standard geometry can be approximated from *j-Re and f-Re* relations by calculating the values of C_j m_j, C_f and m_f according to the following procedure:



d_o= Tube outside diameter =16.38 x 10^{-3} m
P_{fin} = Fin pitch = 276 per m
D_h = Flow passage hydraulic diameter=6.68×10^{-3} m
T = Fin thickness = 0.25×10^{-3} m
σ = Free flow area/Frontal area = 0.449
α = Heat transfer area/total volume = 269×10^{-3} m
A_f/A= Fin area/total area = 0.83
Note: Minimum free-flow area is in spaces transverse to flow

Fig. 3: Annular Fin Standard Geometry

The value of m_j and C_j in Eq(6) are defined as follows.

Where $$m_j = \frac{\ln j_2 - \ln j_1}{\ln \mathrm{Re}_2 - \ln \mathrm{Re}_1} \quad (8)$$

And

$$C_j = e^{bj}$$

Where

$$b_j = \ln j_2 - m_j \ln \mathrm{Re}_2 \quad (9)$$

The value so m_f and C_f in Equation 7 are defined in Equation 10 and Equation 11.

$$m_f = \frac{\ln f_2 - \ln f_1}{\ln \mathrm{Re}_2 - \ln \mathrm{Re}_1} \quad (10)$$

$$C_f = e^{bf}$$

$$b_f = \ln f_2 - m_f \ln \mathrm{Re}_2 \quad (11)$$

Re_1, Re_2, j_1 j_2, f_1 and f_2 are the Reynolds numbers, Colburn factors and friction factors at any two selected points on the corresponding straight line in the chart. The above procedure is repeated for the 12 geometries.

For the geometry of the annular heat exchanger under consideration the parameters C_j, C_f, m_j and m_f are not known. Therefore, these parameters are also added with other design parameters which are considered for optimization process. The range of these parameters is taken to be within the practical limits of the standard geometries (i.e after the calculation of 12 values of certain parameters, say C_j, the range taken of Cj in optimization process is taken between the minimum and maximum values amongst the 12 values). Thermodynamic model equations are taken [9].

Length of fin is calculated using Equation 12.

$$L_f = \frac{(d_f - d_o)}{2} \quad (12)$$

Ratio of inside surface area to the outside surface area is calculated using following equations.

$$\frac{A_i}{A_0} = \frac{\text{Inside surface Area of tubes}}{(\text{surface area of fins}) \times 2 + \text{Bare surface area of tubes}}$$

$$\frac{A_i}{A_0} = \frac{Di}{0.5P_f(d_f^2 - d_0^2) + d_0 - P_f t_f d_0} \quad (13)$$

Outside Reynolds number and convection heat transfer coefficient is calculated using the following equations

$$\mathrm{Re}_o = \frac{G D_h}{\mu} \quad (14)$$

Where D_h is called hydraulic diameter and it is defined as

$$D_h = \frac{4xCross - Sectional\ Area}{Wetted\ Perimeter}$$

Where G is:

$$G = \frac{\dot{m}_o}{\sigma A_{fr}} \quad (15)$$

A_{fr} is the frontal area of heat exchanger and is calculated in terms of height and width as.

$$A_{fr} = W \times H \quad (16)$$

$$h_o = j_o \frac{G\,Cp}{\mathrm{Pr}^{2/3}} \quad (17)$$

The number of tubes in the direction of height H is calculated using Equation-18.

$$N_{tr} = \frac{H}{h} \quad (18)$$

Where h is the center to center tube distance in vertical direction as shown in Fig-2. Inside Reynolds number is calculated using Equation-19.

$$\mathrm{Re}_i = \frac{4\dot{m}_i}{\pi \mu d_i N_{tr}} \quad (19)$$

Inside coefficient is calculated using the following equation.

If Re ≤ 2300 then

$$h_i = \frac{4K_i}{d_i} \quad (20)$$

But if Re> 2300 then inside convective heat transfer coefficient is calculated using Equation-23

$$h_i = \frac{Nu \times K_i}{di} \tag{21}$$

Where

$$Nu = \frac{(f_i \times 0.125) \times (\mathrm{Re}_i - 1000) \times \mathrm{Pr}_i}{(1 + 12.7(f_i \times 0.125)^{0.5} \times (\mathrm{Pr}_i{}^{2/3} - 1))} \tag{22}$$

And

$$f_i = (0.79 \ln \mathrm{Re}_i - 1.64)^{-2} \tag{23}$$

After calculating the inside and outside Reynolds numbers, inside and outside convective heat transfer coefficients are calculated and then the overall heat transfer coefficient is calculated using following equation

$$U_o = \left[\frac{1}{h_i \dfrac{A_i}{A_o}} + \ln \frac{d_0/d_i}{2\pi K \dfrac{W}{A_o}} + \frac{1}{h_o \eta_{o,o}} \right]^{-1} \tag{24}$$

$\frac{A_i}{A_o}$ is the ratio of inside and outside surface areas and is calculated using Equation-15.

Where W is the width of heat exchanger and it is the length of tube and η_o is the overall fin efficiency and is calculated using Equation-25

$$\eta_o = \frac{\tanh(ML_f)}{ML_f} \tag{25}$$

Where M is defined as:

$$M = \sqrt{\frac{h_o P}{A_o K_{fin}}} \tag{26}$$

Where P is perimeter of fin surface

$$P = \pi(d_f + d_o) \tag{27}$$

Finally the depth of heat exchanger and the number of tubes required is really the actual achievement for the sizing problem of heat exchanger. The volume of heat exchanger required is calculated using the following equation.

$$V_{tot} = \frac{A_o}{\alpha} \tag{28}$$

Where A_o is required heat transfer surface area on the fin side and it is calculated by using NTU approach as.

$$A_o = \frac{NTU\, C_{\min}}{U_o} \tag{29}$$

Finally the depth of heat exchanger is calculated as

$$dt = \frac{V_{tot}}{A_{fr}} \tag{30}$$

Number of columns and total number of tubes are calculated as following.

$$N_{tc} = \frac{dt}{d} \tag{31}$$

$$\mathrm{N_t} = \mathrm{N_{tr}} \times N_{tc} \tag{32}$$

After calculating volume, number of passes and number of tubes required, pressure drops on both sides of tubes are calculated. Based on the pressure drop power requirement is evaluated. The pressure drop is zero on gases side if the following condition is satisfied [9].

$$(1 + \sigma^2)\left(\frac{v_{o,out}}{v_{o,in}} - 1\right) + \left(\frac{f_0\, \alpha\, V_{tot} v_m}{\sigma\, A_{fin}\, v_i}\right) < 0 \tag{33}$$

If the above condition is not satisfied then the pressure drop is calculated using Equation-34 [9].

$$\Delta P_o = \left[(1 + \sigma^2)\left(\frac{v_{0,out}}{v_{o,in}} - 1\right) + \left(\frac{f_0 \alpha V_{tot} v_m}{\sigma A_{fr} v_{o,in}}\right) \right] \left(\frac{v_{o,in} m_i^2}{2\sigma^2 A_{fr}^2}\right) \tag{34}$$

Tube side or inside pressure drop is calculated by the following equation;

$$\Delta P_i = \frac{8 m_i^2 W N_{tr} f_i}{\pi^2 (d_i)^5 \rho_i N t p^2} \tag{35}$$

Finally power requirement on both fin side and tube is calculated using the appropriate pressure drops.

$$P = \frac{\dot{m} \Delta P}{\rho} \tag{36}$$

Where $\Delta P = \Delta P_i + \Delta P_o$

2.2 *Objective Function Calculation*

Life cycle cost of heat exchanger has two components, capital and operating cost. Capital cost is the sum material cost and manufacturing cost whereas, sum of pumping and exergy destruction cost is the operating cost. After calculating the capital and annual operating cost, the lifecycle cost is calculated on the bases of present cost estimation due to inflation.

$$C_{LC} = C_{Cap\cos t} + C_{Opt.Cost} \tag{37}$$

For capital cost calculation, different methods are used [10]. In this paper capital cost is calculated as a function of surface area of the heat exchanger using the following correlation which has been considered by Vatavuk's [10].

$$C_{Cap\cos t} = 231 \times (A_o + A_i)^{0.693} \tag{38}$$

Pumping cost for both outside and inside fluids is based on fuel cost. Fuel cost in case of automobile is the cost per liter of gasoline while in case of energy recovery from hot stream gases is in kWh. Amount of pumping cost depends on number of hours used during a year. In case of energy recovery from the hot stream of gases it is calculated as follows.

$$CP_y = P \times N_{hr} \times N_{day} \times C_{fuel} \tag{39}$$

The other component of operating cost is the exergy destruction cost due to irreversibilities in the system. Exergy change on fin side and tube side fluid is calculated in the following way.

$$\Delta\psi_o = \left[C_{po}(T_{o,out} - T_{o,in}) - to \times \left(\left(C_{po} \times \ln \frac{T_{o,out}}{T_{o,in}} \right) - \left(R_o \times \ln \frac{P_{o,in}}{P_{o,out}} \right) \right) \right] \tag{40}$$

$$\Delta\psi_i = \left[C_{pi}(T_{i,out} - T_{i,in}) - \frac{\Delta P_i}{\rho_i \times 1000} - C_{pi} \times to \times \ln \frac{T_{i,out}}{T_{i,in}} \right] \tag{41}$$

Finally the exergy destruction is

$$\psi_d = -(\dot{m}_i \Delta\psi_i + \dot{m}_o \Delta\psi_o) \tag{42}$$

Exergy destruction cost is calculated in the same manner as pumping cost is calculated. For the lifecycle cost the effect of inflation is also taken into account by the present cost estimation.

$$TOPC = \frac{(CP_y + C\psi_{d,y}) \times \left(\frac{1}{(I+1)^N} - 1 \right)}{\frac{1}{I+1}} \tag{43}$$

Hence the total lifecycle cost (TLC) is the sum of capital and total operating cost.

3. Optimization Methodology

Optimization process of heat exchanger is based on minimizing surface area, pressure drop and exergy destruction. These parameters represent the three components of lifecycle cost (i.e. capital cost, pumping cost and exergy destruction cost).

As shown in the above section the lifecycle cost calculation of heat exchanger requires carrying out a comprehensive thermal and hydraulic design procedure. Thermal design is to calculate the surface area required for a given heat load under the given operating conditions. Hydraulic design includes the calculation of pressure drops on tube and fin sides. From this pressure drop power required to run the pump or fan is calculated. The details of optimization of lifecycle cost will go through the following steps.

- First particular values of geometric parameters are taken within the given range
- Then based on effectiveness of heat exchanger the surface area required for hot and cold sides is estimated using Equations (1-29)
- Pumping power required based on pressure drops on both sides is calculated using Equations (34-36).
- Exergy destruction is calculated next using Equation (42-44).
- After this objective function based on capital and operating cost is calculated.
- The optimization algorithm is applied to select new set of values for design parameters (σ, α, do, dh, L_f, P_f, t, h, d, A_f/A, C_j, C_f, m_j and m_f).
- The design parameters and objective functions for the predefined number of generations have been iterated till minimum lifecycle cost is obtained.

Design parameters for optimization are different geometry parameters like free flow area/frontal area (σ), heat transfer area/total volume (α), fin area/total area (A_f/A_t), hydraulic diameter (dh), tube outer diameter (do), length of fin (L_f), fin pitch (P_f), fin thickness (t), tube adjacent vertical height (h), tube adjacent horizontal distance (d). Range of these parameters is taken from the twelve geometries given in Kays & London [8]. These parameters are iterated to get the minimum lifecycle cost and particular values of these parameters to define new geometry which gives minimum lifecycle cost. The optimization process is shown in Fig.4.

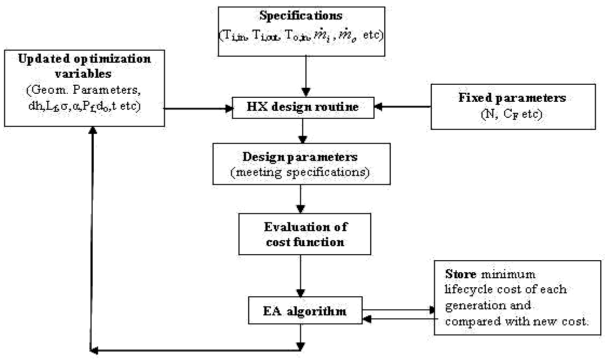


Fig. 4: Proposed Optimization Flow Chart

3.1 *Optimization Problem Description*

The aim of optimization is to minimize the lifecycle cost of heat exchanger, under the following constraints.

Objective function:
To Minimize:
C_{LC} (total lifecycle cost)

Subject to:

$\dot{m}_i, \dot{m}_o$, $T_{C,in}$, $T_{C,o}$, $T_{H,in}$ have specified values shown in Table-1

And the geometric parameters are within the specified range.

a. $0.439 \le \boldsymbol{\sigma} \le 0.642$
b $191 \le \boldsymbol{\alpha} \le 535$
c $0.001167 \le \mathbf{dh} \le 0.01369$
d $0.00965 \le \mathbf{do} \le 0.02601$
e $0.0056 \le \mathbf{L_f} \le 0.00905$
f $0.0028089 \le \mathbf{P_f} \le 0.00362318$
g $0.00025 \le \mathbf{t} \le 0.00048$
h $0.682 \le \mathbf{A_f/A} \le 0.917$
i $0.024765 \le \mathbf{h} \le 0.0782066$
j $0.02032 \le \mathbf{d} \le 0.0524$
k. $0.22239 \le m_j \le 0.51139$
l. $0.14189 \le m_f \le 0.28628$
m. $0.046346 \le C_j \le 0.4209$
n. $0.08821 \le C_f \le 0.51614$

3.2 *Evolutionary Optimization*

Evolutionary optimization algorithm is derived from Darwin Theory of evolution. The principle idea which is opposite of classical optimization methods is that, instead of using the analytical model of a function and the corresponding gradients for guiding the search along suitable directions a stochastic procedure is used. This procedure is initiated by randomly generating a set of possible solutions. Each solution is referred to as an individual and the set itself is referred to as the population. For reasonable performance of the optimization procedure it is necessary that these individuals be scattered through the entire solution space. Once initialized, these individuals are evaluated using the function. The value thus returned from the function is referred to as the fitness value of the individual. After evaluation, the individuals with the lowest fitness values are selected to form the next generation of individuals and the remaining are discarded. This process is then repeated until a stopping criterion is achieved. This could either be a fixed number of generations or a minimum value of the fitness function. Fig. 5 shows the flow chart of this method.

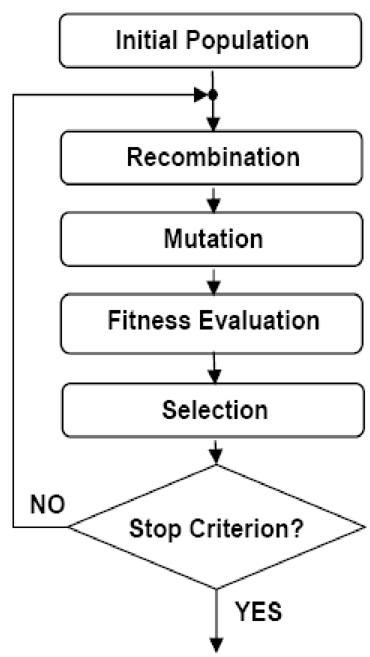


Fig. 5: Evolutionary strategies flow chart.

In this work the minimum lifecycle cost of heat exchanger is determined by the given limits of the objective function which have been taken within the range of geometrical parameters given in [8]. The optimization procedure is carried out on personal computer (PC) using evolutionary technique through a computer program that has been written in the scientific computing environment of MATLAB. In the program the number of individuals assigned to every generation is set to 35 while the maximum number of generations is selected in the range of 100 to 300. It was observed that the test of convergence was always obtained within 100 generations.

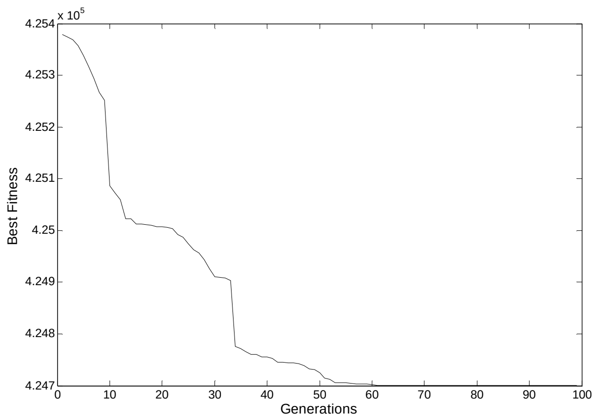


Fig.6: Convergence of fitness with respect to generations

As an example of population convergence of design parameters, Fig. 6 shows the_convergence of population with respect to generations of flow passage hydraulic diameter, dh. The populations related to other design parameters are showing more or less variation similar to that presented in Fig.6.

4. Results and Discussion

In the analysis for cost calculation the cost of energy is taken in electricity cost units that is in kWh and the rate is taken as \$0.12/kWh. Operating cost with inflation was_computed with N=10 years and annual inflation rate is 10%. The results for cost optimization are obtained for two cases related to operating cost. In the first case the results of optimized annular fin heat exchanger are obtained for the case of operating cost including exergy destruction cost while in the second case the results are obtained without including the exergy destruction cost. In the first case the obtained results for the optimized exchanger are presented in Fig. 7 and in Table 2. The results in the second case are listed in Table 3. Fig.7 shows the lifecycle cost of the present optimized exchanger along with the results of the 12 standard geometries. The Fig. 7 clearly shows that the lifecycle cost of the new optimized annular fin geometry heat exchanger has the minimum value for lifecycle cost.

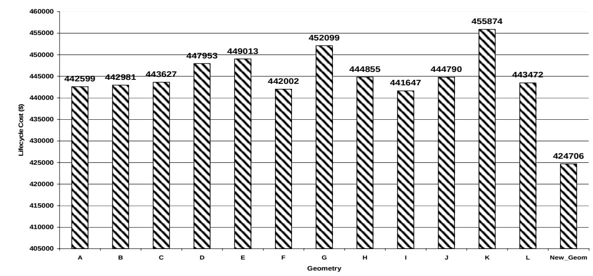


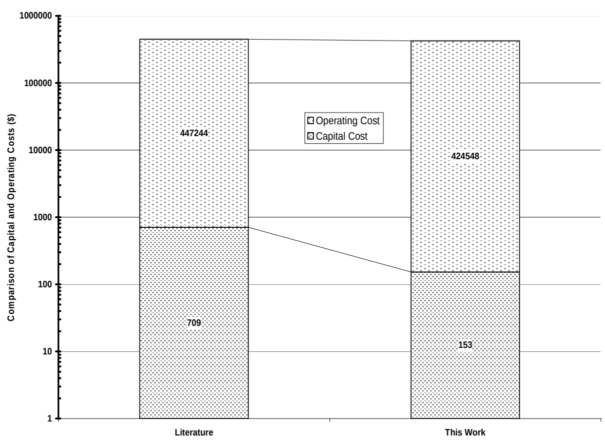
Fig.7: Lifecycle Cost of waste heat recovery radiator including exergy destruction cost

So the new obtained geometrical design parameters define a new annular fin heat exchanger with minimum lifecycle cost. These new values of geometrical parameters,_operating cost and total lifecycle cost are listed in Table 2. It also shows the operating cost and total lifecycle cost for two standard geometries of heat exchangers, the first is the standard geometry with identity D which has analyzed in [9] and the second is the standard geometry with identity I which has the minimum lifecycle cost amongst the 12 standard geometry.

Table 2: Comparison of heat exchanger design considering pumping and exergy destruction costs with true literature values

	Case Study Literature values [11]	Min. Cost Using Standard Geom.	This Work With Min Cost
Geometry Type	D	I	New Geom.
A_o (m^2)	4.0212	5.8386	0.3718
A_i (m^2)	1.76	1.297	0.15
V_{tot} (m^3)	0.02	0.0419	0.0069
N_{tc}	4	6	2
N_t (# of tubes)	57	35	7
Cap cost	709	811	153
PPy (kWh)	19189	7383	75
CPPy (\$)	2303	886	9
TPPC (\$)	15564	5996	61
ψ_{dy} (kWh)	532230	536120	523358
$C\psi_y$ (\$)	63868	64334	62803
Texd (\$)	431680	434840	424487
CostY	66880	66031	62965
C_{LC} (\$)	447953	441647	424548

As declared above the operating cost in Table 2 includes both pumping and exergy destruction cost. The table shows a significant decrease in heat exchanger surface area which is 11 times less than that of the area calculated in the literature, case D. This decrease in surface area decreases the pumping cost by $2294 per year as compared to geometry D and $877 as compared to geometry I (standard geometry which gives minimum cost amongst 12 geometries). But the exergy destruction cost reduces to only 2% as compared to geometry D. Hence, small decrease in exergy destruction cost is due to decrease in pumping cost because the temperature limits are same in both cases. This decreases the overall annual operating cost by 5% but the pumping cost reduces to $15503 as compared to geometry D and $5935 as compared to geometry I. The overall decrease in lifecycle cost is ($23,405) for new geometry. The large decrease in pumping cost and capital cost is due to large reduction in surface area of the heat exchanger. Lifecycle cost in each case is calculated by considering 10% inflationary effect. Detail parameters are shown in Table-2. The comparison of capital, operating and total cost is shown in Fig.8.



Fig, 8: Comparison of capital and operating cost (pumping + exergy destruction cost)

Table 3 shows the optimized geometry parameters as well as the lifecycle cost for the case in which the pumping cost is taken as the only component of operating cost (i.e. excluding the exergy destruction cost but 10% inflationary cost is included). The cost function in this case is the sum of capital and pumping cost. The lifecycle cost using new optimized geometry and standard geometry D is shown for the purpose of comparison. It can be seen from the table that capital cost decreases by 4 times for the new optimized geometry as compared to geometry D. It also shows that the operating cost which is the pumping cost is reduced by $15,549 as compared to geometry D. The annual savings in operating cost is ($2,300) and the overall cost is reduced by $16,085 as compared with geometry D. Further comparison of capital and operating costs in the two cases is shown in Fig.-9.

Table 3: Comparison of heat exchanger design considering capital and pumping costs with case D

	Case Study Literature values [11]	This Work With Min Cost
Geometry Type	D	New Geom.
A_o (m^2)	4.0212	0.4388
A_i (m^2)	1.76	0.19
Vtot (m^3)	0.02	0.0155
N_{tc} (passes)	4	2
N_t (# of tubes)	57	7
Capcost	709	173
PPy (kWh)	19189	21
CPPy ($)	2303	2.5
TPPC ($)	15564	15
CostY	3012	176
Lcost ($)	16273	188

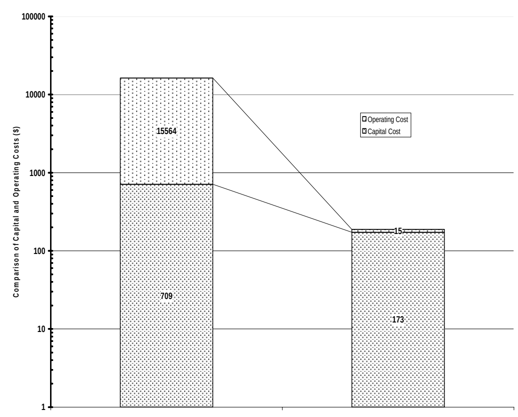


Fig.9: Comparison of capital and operating cost (pumping cost)

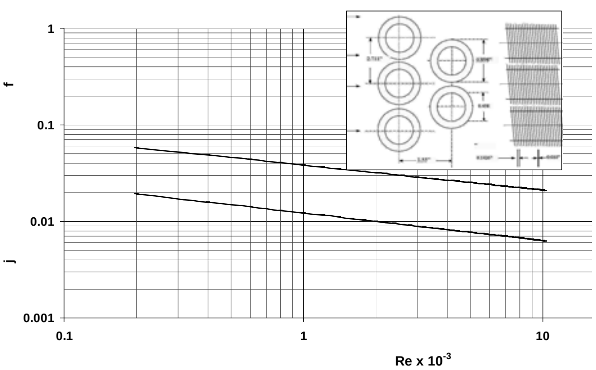
The analysis of the results in both cases shows that the capital cost is not the dominant factor as compared to operating cost. It is seen that the capital cost in both cases decreased as well as the operating cost. These cases show that the Evolutionary Algorithm (EA) gives better results as compared to literature values by finding the optimized geometric parameters for the new geometry. The new geometry not only decreases the capital cost but reduces the pumping cost to the significant amount. There is significant decrease in pumping cost but the decrease in exergy destruction cost is not significant because of same temperature limits on hot and cold fluid. Hence in both cases (cost with and without exergy destruction) has decreased.

Finally, the results obtained for optimized annular fin geometry has given optimized relations for Colburn and Friction factors as functions of Re number. These relations read as:

$$j = 0.229106 Re^{-0.259182} \tag{46}$$

$$f = 0.08821 Re^{=0.28628} \tag{47}$$

The above relations are plotted Fig.10 to give chart similar to those given in [1] for each of the 12 standard geometry. The corresponding design parameters are listed below.



d_o = Tube outside diameter $=11.63\times10^{-3}$ m
P_{fin} = Fin pitch = 276 per m
D_h = Flow passage hydraulic diameter$=12.9\times10^{-3}$m
T = Fin thickness = 0.253×10^{-3} m
σ = Free flow area/Frontal area = 0.498
α = Heat transfer area/total volume = 483 m^{-1}
A_f/A = Fin area/total area = 0.75
Note: Minimum free-flow area is in spaces transverse to flow

Fig.10: Optimized Annular Fin Geometry

5. Conclusions

In this paper an optimized design for annular fin heat exchanger that is based on economic consideration has been carried out. The heat exchanger is selected to have the same material, frontal area, effectiveness and thermal load of standard geometry heat exchangers. The optimization procedure for the design parameters has been conducted using Evolutionary Algorithm (EA). The objective function in the optimization process was the total lifecycle cost which includes both capital and operating costs. The operating cost includes pumping and exergy destruction costs plus 10% inflation rate. The optimization of the lifecycle has been carried out for operating cost with and without exergy destruction. The optimization of the lifecycle cost for the case of operating cost without exergy destruction cost has shown significant cost reduction as compared with that for standard geometry considered in the literature. The analysis of the results has shown that both capital cost and operating cost decrease as a result of the surface area reduction of the optimized design which leads to significant reduction of the total life cycle cost. On the other side relations for both Colburn and friction factors are developed for the optimized geometry design of the heat exchanger. The study has also proved that EA has good flexibility in examining good number of alternative solutions which lead to quick solution of the design problem. This flexibility of EA in solving optimization problems made it one of the recommended efficient optimization techniques in dealing with similar optimization problems.

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Nomenclature

A_{fr}	Frontal area (m^2)
A_i	Inside surface area (m^2)
A_o	Outside surface area (m^2)
A	Total area (m^2)
A_f	Area of fin (m^2)
C_{Po}	Outside fluid (air) specific heat (kJ/kg K)
C_{Pi}	Inside fluid (water) specific heat (kJ/kg K)
C_{max}	Maximum heat capacity (kW/K)
C_{min}	Minimum heat capacity (kW/K)
C_R	Ratio of Cmin/Cmax
C_{fuel}	Cost of energy ($/kWh)
$C_{Capcost}$	Capital investment ($)
C_{LC}	Lifecycle Cost ($)
$C_{Opt.Cost}$	Operational Cost ($)
C_j	Constant of Colburn factor
C_f	Constant of friction factor
d_o	Outside tube diameter (m)
d_i	Inside tube diameter (m)
d_f	Fin diameter (m)
D_h	Hydraulic diameter (m)
d	Distance between columns (m)
dt	Depth of heat exchanger (m)
f	Friction coefficient
G	Mass velocity (kg/m^2s)
h_i	Convective coefficient of inside fluid (W/m^2K)
h_o	Convective coefficient of outside fluid (W/m^2K)
h	Distance between two tubes (m)
H	Height of radiator (m)
I	Annual discount rate (%)
j	Colburn factor
K	Thermal conductivity of aluminum (W/m.K)
L_f	Length of fin (m)
$\dot{m}_i$	Inside mass flow rate (kg/s)
$\dot{m}_o$	Outside mass flow rate (kg/s)
m_j	Constant for Colburn factor
m_f	Constant for friction factor
N	Life of heat exchange (years)
N_{tc}	Number of tube columns
N_{tr}	Number of tubes in each column
Nt	Total number of tubes
N_{hr}	Number of hours
N_{day}	Number of days
NTU	Net transfer units
Pr	Prandtl number)
PPy	Pumping power per year (kW)
P_{fin}	Fin pitch (fins/m)
P	Power (kW)
q	Heat capacity (kW)
Re_i	Inside Reynolds number
Re_o	Outside Reynolds number
$T_{H,in}$	Outside fluid inlet temperature (K)
$T_{H,out}$	Outside fluid outlet temperature (K)
$T_{C,in}$	Inside fluid inlet temperature (K)
$T_{C,out}$	Inside fluid outlet temperature (K)
$T_{i,out}$	Inside fluid outlet temperature (K)
t	Fin thickness (m)
to	Ambient temperature (K)
TOPC	Total operating cost ($)
TLC	Total lifecycle cost ($)
U_o	Overall heat transfer coefficient (W/m^2K)
V_s	Volume of cell (m^3)
V_{tot}	Total volume (m^3)
W	Width of radiator (m)

Greek Symbols

ΔP_o	Outside fluid pressure drop (kPa)
ΔP_i	Inside fluid pressure drop (kPa)
υ	Specific volume (m^3/kg)
σ	A geometry constant
ρ	Density (kg/m^3)
ε	Effectiveness of heat exchanger
$\Delta\psi_i$	Exergy change inside (kW)
$\Delta\psi_o$	Exergy change outside (kW)
$\psi_{d,y}$	Exergy destruction per year (kW)
μ	Dynamic Viscosity (N-s/m)
η_o	Overall fin surface efficiency