New approach to design compact heat exchanger by PSO algorithm

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Abstract: In this study a plate-fin Compact Heat Exchanger is designed for microturbine applications.One of the important stages to design a plate-fin heat exchanger is optimization process.Certainly,it depends on position where it is going to employ. Firstly, eight type of heat exchangers are designed and compared for different parameters and eventually a fin is selected for optimization process based on objective function, which is minimum volume. In this paper, it has been effort to offer a practical method for selection of the fin according to operating conditions and objective function. For example, in the food industry the total annual costs is important, while in the microturbine applications heat exchanger efficiency and outlet temperature of cold fluid is significant .So , in designing of heat exchanger for any applications noting to some characteristics of fin to optimal design is necessary. Then, optimization process has been done by GA and PSO algorithm. Eventually, by using the optimization process results redesigning process has been done that at this stage a practical heat exchanger will be designed in order to use in industry.

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Introduction:

Heat exchangers play a key role on the power plant, air – conditioning, and refrigeration systems. Nowadays, compact heat exchangers are popular for saving energy, and returns to their high compactness as well as excellent heat transfer performances. Despite this unique performance, high pressure drop for both sides, cold and hot, were always a challenge for engineers.

High pressure drop is due to increased friction and transferring flow from laminar regime to turbulent. The surface geometries of strip fins are described by the fin height (h), transverse spacing (s) and thickness (t)[1].

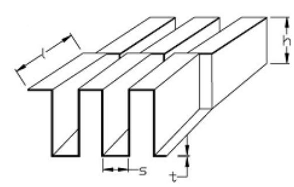

Figure 1. strip fin

Nomenclature

A_f	surface area $\left(m^2\right)$	Re	Reynolds number
b	plate spacing (mm)	S	heat transfer area
C	specific heat $\left(\frac{J}{kg.k}\right)$	S_f	fin area
C_A	price per unit area $\left(\frac{\$}{m^2}\right)$	TAC	total annual costs

C_{in}	investigation price($)		V_t	volumetric flow	
C_{op}	operation price($)		Greek symbols		
C_1	cognitive components		α	total heat transfer area /total volume $\left(\frac{m^2}{m^3}\right)$	
C_2	social components		β	total heat transfer area/volume between plates $\left(\frac{m^2}{m^3}\right)$	
f	friction factor		ρ	fluid density $\left(\frac{Kg}{m^3}\right)$	
G	mass velocity $\left(\frac{Kg}{m^2.s}\right)$		μ	dynamic viscosity $(Pa.s)$	
h	convective coefficient $\left(\frac{W}{m^2.k}\right)$		δ	fin metal thickness(mm)	
H	heat exchanger height(mm)		ε	heat exchanger efficiency	
j_H	colburn factor		τ	hours per year	
K	thermal conductivity $\left(\frac{W}{m^2.k}\right)$		$v_{i,j}$	velocity of the i-th particle	
K_c	contraction coefficient		Subscripts		
K_{el}	expansion coefficient		h	hot fluid	
K_{el}	electrical energy price $\left(\frac{\$}{MW\ \ h}\right)$		c	cold fluid	
m	mass flow rate $\left(\frac{Kg}{s}\right)$		g	best particle	
n	number of layers		max	maximum	
NTU	heat transfer unit		min	minimum	
$p_{g,j}(t)$	best previous position among all the particles				

Typically, heat exchanger design is based on trial- error process in which geometrical and operational parameters are selected in order of satisfying specified requirements and leading to an optimum solution simultaneously [2]. Due to this fact that there is always high possibility that the selected design parameters do not ensure the optimum solution, wide works have been devoted to propose optimization methods for compact heat exchanger.

2. Fin selection

One of key points about plat- fin heat exchanger is choosing suitable fin .In this study, different types of strip fins are studied and compared for different terms like Reynolds number, convective heat transfer coefficient or pressure drop. Eventually, after inspecting this fins according to objective function ,which can be minimum costs, volume or even height , the best fin will be selected. According to limitations that there is on surface area, the objective function is minimum volume, thus minimizing volume by swarm algorithm, after selecting fin, can have significant effects on heat exchanger function.

There are two basic types of thermal design problems: rating and sizing. In a rating problem, the geometry and size of the heat exchanger are fully specified. Entering flow rates and fluid temperatures are known. The jab is to calculate the thermal effectiveness and pressure drop of each stream [4].

In a sizing problem the heat exchanger requirement is specified and the designer must calculate the heat exchanger size. Normally pressure drop limits are given for each fluid stream. Since the entering flow rates, temperatures, and pressures are given, and the heat duty (or leaving temperatures) is specified, the ε and NTU (number of transfer units) are directly calculable [4].

It is assumed that working conditions for this compact heat exchanger is according to Table 1:

Table 1.Operating conditions of microturbine[3]

Variables	**Data**
Allowable pressure drop for hot side(%)	6
Allowable pressure drop for hot side(%)	3
Outlet gas temperature(k)	694
Inlet gas temperature(k)	865
Outlet air temperature(k)	670
Inlet air temperature(k)	475
Outlet pressure from compressor(bar)	4
Outlet pressure from turbine(bar)	1.06
Gas mass flow rate (kg/s)	1.4676
Air mass flow rate(kg/s)	1.45

Also, fluid properties are:

Table .2.Fluid properties

	$\rho_i\left(\frac{Kg}{m^3}\right)$	$\rho_0\left(\frac{Kg}{m^3}\right)$	$K\left(\frac{W}{m.K}\right)$	$\mu\left(\frac{N.S}{m^2}\right)$	$C_P\left(\frac{Kj}{Kg.K}\right)$	$\frac{1}{\rho_m}$	Pr
Gas	0.399	0.491	43.9	2.88×10^{-5}	1.58	0.4182	0.683
Air	2.934	2.017	0.05	3.55×10^{-5}	1.04	2.2684	0.735

In the analysis, for the sake of simplicity, the variation of physical property of fluids with temperature is neglected where both fluids are considered as ideal gases. Besides, other assumption are as follows:

1- Number of fin layers for the cold side is assumed to be one more than the hot side because of avoiding heat waste to the ambient.

2- Heat exchanger is working under steady state conditions.

3- Heat transfer coefficient and the area distribution are presumed to be uniform and constant.

4- The thermal resistance of walls and the influence of fouling is neglected.

In this work, since the outlet temperature of the fluids is not specified the ε-NTU method is used for rating performance of the heat exchanger [5].

Hence, the effectiveness of cross-flow heat exchanger, for both fluids unmixed is proposed as [6]:

$$\varepsilon = 1-\exp\left[\left(\frac{1}{Cr}\right)NTU^{0.22}\left\{\exp\left[-Cr.NTU^{0.78}\right]-1\right\}\right] \qquad (1)$$

Where, $Cr = C_{\min}/C_{\max}$. Neglecting the thermal resistance of the walls and fouling factors, NTU (number of transfer unit) can be calculated by Eq.3:

$$\frac{1}{US} = \frac{1}{(hS)_h} + \frac{1}{(hS)_c} \qquad (2)$$

$$NTU = \frac{US}{C_{\min}} \qquad (3)$$

Also, heat transfer coefficient is:

$$h = j.G.C_P.P^{-\frac{2}{3}} \qquad (4)$$

Where j is considered as colburn factor and G is mass velocity and determine by:

$$G = \frac{\dot{m}}{A_f} \qquad (5)$$

In this formula, A_f is free flow cross- sectional area which after calculating σ, contraction coefficient, can be computed.

$$A_f = \sigma \times A \qquad (6)$$

Geometric parameters of σ and α are defined by below terms:

$$\sigma = \alpha \times r_h \qquad (7)$$

$$\alpha = \frac{b_1}{b_1 + b_2 + 2\delta} \times \beta \qquad (8)$$

Where A is surface area, and so is different for each fluid. In this formula, r_h, δ and β are hydraulic radius, fin metal thickness and compactness factor respectively. In addition, b is one of the important feature in fin geometric properties, because b is the height of fin, and consequently has direct impact on total volume.

The pressure loss for each stream through the heat exchanger finned passages is calculated by

$$\Delta P_1 = \frac{G_1^2}{2\rho_{in,1}}\left[\begin{array}{l}\left(1+K_{C,1}-\sigma_1^2\right)+2\left(\frac{\rho_{in,1}}{\rho_{out,1}}-1\right)+ \\ \left(f_1\times\frac{S_1}{A_1}\times\frac{\rho_{in,1}}{\rho_{m,1}}\right)-\left(1-\sigma_1^2-k_{e,1}\right)\times\frac{\rho_{in,1}}{\rho_{out,1}}\end{array}\right] \qquad (9)$$

The terms of Eq. (9) are entrance loss, flow acceleration loss, core friction, and exit loss respectively. The K_c and K_e values depend on the cross- sectional flow geometry, σ and Re [4].

Noticeably, the entrance and exit losses are normally less than 10% of the total core loss, consequently Fig.2 can cover most situations with adequate accuracy.

After thermal design of plate- fin heat exchanger with different types of strip fin, the following result obtained.

As was mentioned previously, selecting fin for a plate- fin heat exchanger depends on situation, where it is going to employ, and so a wrong choice can lead to increase cost by 22% and decrease heat exchanger efficiency by 8%. Hence, when minimum costs is objective function, pressure drop is the most important feature in compact heat exchanger because according to G.N.Xie and B.Sunden researches [8] pressure drop has direct effect on operating cost. Where

$$TAC = C_{in} + C_{op} \quad (10)$$

$$C_{in} = C_A \times A^n \quad (10\text{-a})$$

$$C_{op} = \left\{ K_{el}\,\tau \frac{\Delta p v_t}{\eta} \right\}_h + \left\{ K_{el}\,\tau \frac{\Delta p v_t}{\eta} \right\}_c \quad (10\text{-b})$$

Here C_A and K_{el} are price per unit area and electrical energy respectively. Also, n and τ are the exponent of nonlinear increase with area increase and the hours of operation per year. $\Delta p, v_t$ and η are pressure drop, volumetric flow rate and pump/ compressor efficiency respectively[8].

Table 3.Mannual design results

Fins	Re_h	Re_c	$\Delta p_h\,(Kpa)$	$\Delta p_c\,(Kpa)$	$h_h\left(\frac{w}{m^2.k}\right)$	$h_c\left(\frac{w}{m^2.k}\right)$	U	NTU
$\frac{1}{2}-11.94(D)$	626.85	716.8	2.106	1.341	216.13	242.26	69.45	7.325
$\frac{1}{4}-15.4(D)$	531.84	590.59	2.386	0.989	256.32	281.58	68.43	8.73
$\frac{1}{6}-12.18(D)$	468	520.2	0.719	0.299	212.04	233.45	43.91	6.33
$\frac{1}{7}-15.75(D)$	436.3	484.52	2.814	1.344	288.57	340.97	59.48	9.22
$\frac{1}{8}-16.00(D)$	497.8	553.11	5.345	2.481	318.31	371.87	70.37	9.56
$\frac{1}{8}-16.12(D)$	531.84	590.59	8.669	4.028	276.04	332.78	77.06	10.16
$\frac{1}{8}-19.82(D)$	500.29	555.5	10.206	4.946	387.24	452.66	91.96	12.43
$\frac{1}{8}-20.06(D)$	496.1	551.13	9.486	4.310	376.21	437.18	93.14	12.64

Table 4. Total volume of heat exchangers

	$\frac{1}{2}-11.94(D)$	$\frac{1}{4}-15.4(D)$	$\frac{1}{6}-12.18(D)$	$\frac{1}{7}-15.75(D)$	$\frac{1}{8}-16.00(D)$	$\frac{1}{8}-16.12(D)$	$\frac{1}{8}-19.82(D)$	$\frac{1}{8}-20.06(D)$
Volume	0.218	0.190	0.322	0.278	0.234	0.190	0.189	0.185

Table .5.Cost function variables

Variables	**Data**
Price per unit area, $\left(\frac{\$}{m^2}\right)$	100
Exponent of nonlinear increase with area increase	0.6
Electrical energy price, $\left(\frac{\$}{MWh}\right)$	30
Hours of operation per year	6500
Pump/ Compressor efficiency	0.6

While for a plat- fin heat exchanger which is going to use in food industry, for example, $\frac{1}{6}-12.18(D)$ is the best choice, for some applications such as microturbine or aerospace applications the best choice is $\frac{1}{8}-20.06(D)$. It refers to this fact that in this applications total volume, outlet cold fluid temperature or even heat exchanger weight are the most important things. Thus, in this category NTU and total volume can help to designer. However, Kays and London [7] supply geometric properties of each fin as figures and tables which can be very useful for designer. For instance, a list of strip fins mentioned are according to Table.(6).

Table.6.Strip fin peroperties

Fin	$D_h(mm)$	$b(mm)$	$\beta\left(\frac{m^2}{m^3}\right)$	S_f/S
$\frac{1}{2}-11.94(D)$	2.266	6.02	1512	0.796
$\frac{1}{4}-15.4(D)$	1.605	5.23	2106	0.816
$\frac{1}{6}-12.18(D)$	2.63	8.97	1385	0.847
$\frac{1}{7}-15.75(D)$	2.07	7.72	1726	0.859
$\frac{1}{8}-16.00(D)$	1.862	6.48	1804	0.845
$\frac{1}{8}-16.12(D)$	1.552	5.23	2185	0.823
$\frac{1}{8}-19.82(D)$	1.537	5.21	2231	0.841
$\frac{1}{8}-20.06(D)$	1.491	5.11	2290	0.843

Here D_h, β and $\frac{S_f}{S}$ are hydraulic diameter, compactness factor and fin area/total area respectively. Table of (7) recommends what geometric features are important in different situations.

Table .7

Goals	Parameters
Volume	1) β 2) b
Pressure drop	D_h
Heat exchanger	1) $\frac{S_f}{S}$ 2) D_h
Total annual costs	1) D_h 2) β

Eventually, in this work fin of $\frac{1}{8}-20.06$, according to explanations mentioned, is selected for optimization process.

3. Optimization design method.

In the next subsection first, a brief overview of the GA and classical PSO algorithm is provided for optimization process; and finally a engineering approach is employed to analyze output data from this algorithm to use in industry.

3.1 GA algorithm

The genetic algorithm is maintained by a population of parent individuals that represent the latent solutions of a real- world problem [8]. For instance, the designer may encode the design prameters into corresponding binary strings, afterwards all the binary strings are connected into a binary string, that is represented as an individual. Consequently, a certain number of sets of design parameters become a population of parent individuals. Based on how well each individual fits a given environment, each individual is assigned a fitness and then fit individuals go through the process of survival selection. In other words, crossover and mutation leading to create next generation that called child individuals. By selection of good individuals from parent and child individuals, a new population is formed. More detail about description of genetic algorithms can be found in many books [9,10,11]. Besides, in this work, roulette wheel selection, uniform crossover and six- point mutation were selected.

3.2 Particle swarm optimization

Particle swarm optimization is a heuristic optimization method which was firs introduced by Kennedy and Ebehart [12]. This method is developed from swarm intelligence and is based on social – psychological principles and social behavior. Due to competitive performance of PSO algorithm on complex search spaces, using PSO approaches to solve optimization problems in engineering applications has been increased dramatically. Typically, PSO approaches are well known for their ability to deal with nonlinear and complex optimization problems. However, this method easily suffers from the partial optimism, which causes the less exact at the regulation of its speed and the direction.

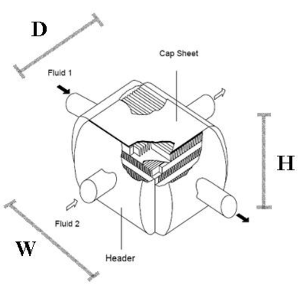

Figure .2.Plate-fin heat exchanger

In the basic particle swarm optimization algorithm, particle swarm consists of "n" particles and the position of each particle stands for the potential solution in D-dimensional space [13]. The particles change its condition based on keeping its inertia and changing the condition according to its most optimist.

The position of each particle in the swarm is affected by the most optimist position during its movement (individual experience) and the position of the most optimist particle in its surrounding (near experience) [13].In other words, PSO combines local search methods with global search methods so as to balance exploration and exploitation, where exploration is a process of visiting partially new regions of a search space and of seeing if anything promising may be found in the regions. Furthermore, exploitation is a process of employing information gathered from the previously visited points in the search space to determine which regions might be profitable to be visited next [14]. In the problem space, each particle keeps track of its coordinates that are associated with the best solution and this value is called personal best or *pbest*. Moreover, another best value that is tracked by the global version of the PSO is the overall best value, which is called *gbest* (global best).

Each particle can be shown by its current speed and position, the most optimist position of each individual and the most optimist position of the surrounding [13]. In the present work, the gbest version of PSO is adopted that is a fully connected neighborhood relation. Hence, the global best particle position for all particles is identical where the speed and position of each particle change according to the following equations:

$$v_{i,j}(t+1) = w\,v_{i,j}(t) + c_1.r_1.\left[p_{i,j}(t) - x_{i,j}(t)\right] + c_2.r_2.\left[p_{g,j}(t) - x_{i,j}(t)\right] \quad (11)$$

$$x_{i,j}(t+1) = x_{i,j}(t) + \Delta t\,v_{i,j}(t+1) \quad (11\text{-a})$$

Here $i = 1,2,....,N$ demonstrates the particles of swarm; $t = 1,2,...,t_{\max}$ indicates the iterations; w is considered as inertia weight factor; $v_{i,j}(t+1)$ is defined as the velocity of the i-th particle with respect to the best previous position of the i-th particle to the j-th dimension .In this formula, $p_{g,j}(t)$ is the best previous position among all the particles along the j-th dimension in iterations . In the equation of (11), the first term is the momentum part of the particle as well as the second term is the cognition part, which represent the independent thinking of the particle itself [14]. Furthermore, C_1 and C_2 are considered as cognitive and social components respectively. Index g represents the index of the best particle among all the particles in the swarm .Variables r_1 and r_2 are values uniformly distributed in the range [0,1] [14]. The position update are represented by Eq. (11-a) where it is based on its previous position and its velocity.

3.3 objective functions

In this work, the optimization target is the minimum total volume which is mainly associated with heat exchanger efficiency and NTU.

For the volume calculation, having H, the height of heat exchanger, W and D, which are considered as width and depth of heat exchanger respectively, is necessary.

$$V = H \times W \times D \quad (12)$$

$$H = n_1(b_1 + a) + n_2(b_2 + a) \quad (13)$$

Where n_1 and n_2 are the number of layers for hot and cold fluids respectively. In addition, a and b are parting sheet thickness and the height of fin(plate spacing)respectively .Since in practical applications the heat exchangers are operated under specified requirements, and consumption of pumping power is necessary to transfer the fluid flow through the passages in the heat exchangers, so pressure drop is inevitable [8]. In other words, the pressure drop must be below a specified maximum value. Hence, the heat exchanger optimization is considered as a constrained optimization process with following conditions:

$$\text{Constraints:}\begin{cases}\Delta p_h < \Delta p_{h,\max} \quad , \quad \Delta p_c < \Delta p_{c,\max} \\ \varepsilon > \varepsilon_{\min}\end{cases} \qquad (14)$$

Here, $\Delta p_{h,\max}$ and $\Delta p_{c,\max}$ are the maximum allowable pressure drop for hot and cold fluids respectively. Besides, for cross- flow heat exchangers, it is economic that the heat exchanger efficiency should not be less than 0.75.

4. Results and discussion

The effectiveness of the present approach using GA and PSO algorithm is assessed by analyzing a case study that was analyzed previously by M. Nasrabadi [15].

In all the experiments realized for this paper, to start GA and PSO approaches, the setup parameters were population size that was 20 for both GA and PSO, also stopping criterion equal to 1000 generations. Hence, the results are according to Table.(8).

To compare the data of different type of fins, in section 2, it was necessary that input data be uniform far all fins. Otherwise manual designing for fin of $\frac{1}{8}-20.06(D)$ is wrong because the maximum allowable pressure drop for hot and cold sides are 6.3 Kpa and 12 Kpa respectively.

As can be seen from table of (8) both GA and PSO algorithm tend to decrease the depth of heat exchanger dramatically compared with manual designing. Before explaining the reasons of this phenomenon, it is essential to discuss about pressure drop. In compact heat exchangers pressure drop consists of three terms:

$$\Delta P = \Delta P_{1-2} + \Delta P_{2-3} + \Delta P_{3-4} \qquad (15)$$

Here, ΔP_{1-2} is pressure drop in the input area, and ΔP_{2-3} and ΔP_{3-4} are pressure drop in central and output area respectively. Typically, the most important term is pressure drop in the central area- second term- because the first and third terms cancel each other. Pressure drop in the central area is because of:

1- Flow friction

2- Momentum change

Therefore:

$$\Delta P_{2-3} = \frac{G^2}{2\rho_i}\left[2\left(\frac{\rho_i}{\rho_o}-1\right)+f \times \frac{S}{A}\times\frac{\rho_i}{\rho_m}\right] \qquad (16)$$

Table .8.Optimization process results

	M.Nasrabadi[15]	Manual designing	GA	PSO
H(mm)	605.21	531.61	518.32	471.56
W(mm)	700	700	644.2	622.65
D(mm)	500	500	324.38	308.23
Re_h	399.56	496.1	554.5	605.08
Re_c	686.16	551.13	874.02	969.25
$h_h\left(\frac{w}{m^2.k}\right)$	215.95	376.21	398.14	410.47
$h_c\left(\frac{w}{m^2.k}\right)$	292.91	437.18	571.05	656.25
$\Delta p_h\,(kpa)$	6.445	9.486	6.131	6.28
$\Delta p_c\,(kpa)$	6.017	4.310	8.128	8.851
Pr_h	0.735	0.735	0.735	0.735
Pr_c	0.685	0.685	0.685	0.685
U	87.73	93.14	106.1	109.81
E	0.77	0.82	0.84	0.85
C_i	4072.19	3713.61	3280.40	2476.57
C_O	12858.34	14873.88	14447.07	15203.43
TAC	16930.53	18587.49	17727.4	17079.97

On the other hand, according to Darcy- Weisbach equation pressure drop is equal:

$$h_f = f\,\frac{L}{D_h}\,\frac{V^2}{2g} \qquad (17)$$

Where f is friction factor, and V, D_h and L are velocity, hydraulic diameter and flow length respectively. Furthermore, hot fluid enters from W (width) according to Fig.3. In other words, hot fluid traverses the depth of heat exchanger, thus it seems reasonable that both GA and PSO algorithm decrease heat exchangers depth because pressure drop for hot side is 9.486 kpa that is much more than maximum allowable pressure drop.

Since particle swarm algorithm has better performance compared with GA algorithm, redesigning process uses its results. As it is mentioned previously, the number of layers for cold side is one more than hot side. Hence, according to H=471.56 mm (from PSO algorithm) the number of layers must be modified.

$$\begin{cases} H = n_1(b_1 + a) + n_2(b_2 + a) \\ n_2 = n_1 + 1 \end{cases}$$

So, if H=471.56mm:

$$n_1 = 44.30 \Rightarrow n_2 = 45.30$$

Here, n_1 must be 45 or 44 because the number of layers must be integer ,so:

$$\begin{cases} n_1 = 45 \\ n_2 = 45 \end{cases} \Rightarrow H = 478.84$$

The width and depth of heat exchanger are also considered 625 and 310, due to economic approaches as well as difficulties caused by maintances. Eventually, after redesigning process in order to use the heat exchanger in industry, the output data are according to Table of (9).

After optimization, height ,width and depth of heat exchanger have decreased 11% ,11% and 38% respectively ,that totally has been led to decrease volume by 49%.Also,Reynolds numbers for both hot and cold fluids have increased, which rising convective heat transfer coefficient is due to this increase.What is more ,having decreased flow length for both fluids ,pressure drop for hot fluid has fallen while for cold fluid has risen .This decrease and increase is because of pressure drop function ,where that is a two-variables function. In other words, both width and depth values determine pressure drop ,and simultaneously be optimized .So ,with optimized pressure drop values ,the total annual cost has decreased by 4%.

In addition, it is useful a comparison between before and after optimization process.

Table 9. Output data from redesigning process

Variables	Optimization
Number of layers for hot	45
Number of layers for hot	46
Width, W	625mm
Depth, D	310mm
Plate spacing, b	5.11mm
Hydraulic diameter, D_h	1.491mm
Compactness factor, β	2290m^2/m^2
Fin metal thickness, δ	0.102mm
Parting sheet thickness, a	0.152mm
Fin length flow direction, L	3.175mm
Fin area/ total area, S_f/S	0.843
Allowable pressure drop for	6.3kpa
Allowable pressure drop for	12kpa
Inlet temperature of gas	865k
Inlet temperature of air	475k
Out let pressure from	4bar
Out let pressure from turbine	1.06bar
Gas mass flow rate	1.45kg/s
Air mass flow rate	1.4676kg/s

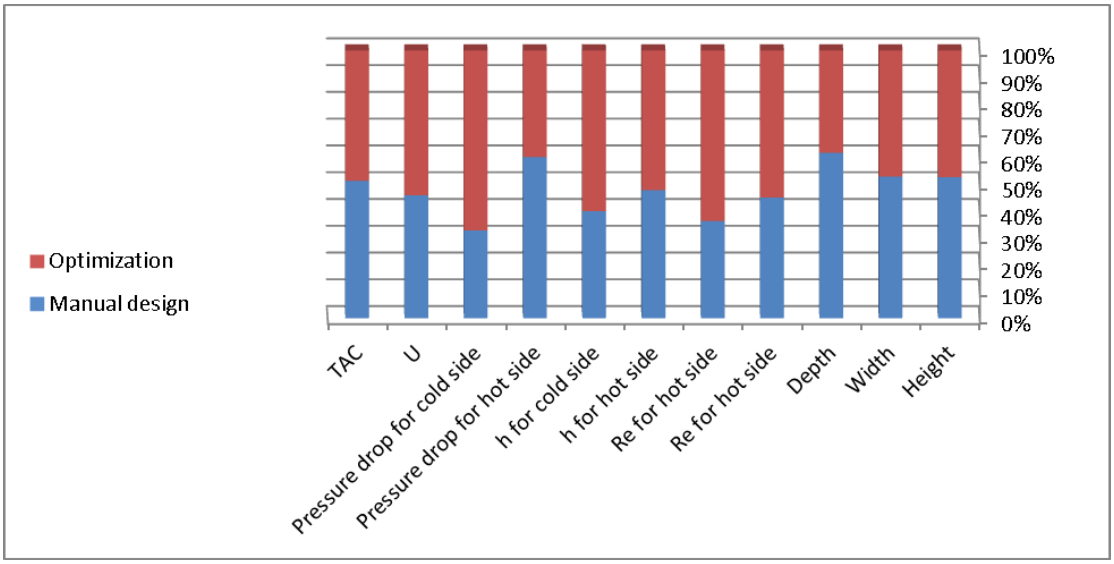

Figure.4. Comparision between mannual design and optimization process

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