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Technical note

OPTIMIZATION OF COMPRESSED HEAT EXCHANGER EFFICIENCY BY USING GENETIC ALGORITHM

M. GHORBANI*
Technical Manager of Inspection at ARAD Engineering & Inspection Company
IRAN
E-mail: mahdi.ghorbani@aradco.ir

S.F. RANJBAR
Department of Mechanical Engineering, University of Tabriz
Tabriz, IRAN

Due to the application of coil-shaped coils in a compressed gas flow exchanger and water pipe flow in air-conditioner devices, air conditioning and refrigeration systems, both industrial and domestic, need to be optimized to improve exchange capacity of heat exchangers by reducing the pressure drop. Today, due to the reduction of fossil fuel resources and the importance of optimal use of resources, optimization of thermal, mechanical and electrical devices has gained particular importance. Compressed heat exchangers are the devices used in industries, especially oil and petrochemical ones, as well as in power plants. So, in this paper we try to optimize compressed heat exchangers. Variables of the functions or state-of-the-machine parameters are optimized in compressed heat exchangers to achieve maximum thermal efficiency. To do this, it is necessary to provide equations and functions of the compressed heat exchanger relative to the functional variables and then to formulate the parameter for the gas pressure drop of the gas flow through the blades and the heat exchange surface in relation to the heat duty. The heat transfer rate to the gas-side pressure drop is maximized by solving the binary equation system in the genetic algorithm. The results show that using optimization, the heat capacity and the efficiency of the heat exchanger improved by *15*% and the pressure drop along the path significantly decreases.

Key words: heat exchanger, pressure drop, heat capacity, optimization, genetic algorithm.

1. Introduction

Considering that the optimization of heat exchangers is important in terms of performance, it is very important to find a suitable method for optimization. In most engineering issues, including the design of compressed heat exchangers, many relationships and parameters are involved that have a nonlinear relationship [1, 2]. The combination of these relationships brings together much more complicated functions that the usual methods of math solving to optimize such functions are either very difficult or in some cases impossible. Several studies have been carried out on the optimization of heat exchangers [3-6]. Xie *et al.* [3] performed a heat exchanger optimization of the plate-fin type of compressed heat exchangers. Using the geometric algorithm method, geometric dimensions of compressed heat exchangers have been optimized. Minimizing the total volume and total annual cost of compressed heat exchanger is considered as a function of the hexadecimal. Optimization results show that by reducing the volume of heat exchanger by about *30*%, the total annual cost would be reduced up to *15*%. In the research by Hao Peng and Xiang Ling [4], the method of genetic algorithm and reverse neural network is used to design an optimized compressed heat

* To whom correspondence should be addressed

exchanger plate-fin. The main objective of the plate-fin compressed heat exchanger design was to minimize the total weight and total annual cost under certain constraints. Manish Mishra *et al*. [5] examined the compressed plate-fin heat exchanger with cross flow and discontinuous blades by the genetic algorithm. The optimization program is aimed at minimizing the entropy production unit for the specified thermal task and with specific space constraints. Sanaye and Hajabdollahi [6] examined thermal modeling and optimized compressed heat exchanger design. Six parameters, blade stroke, blade height, gap between blades, cold flow length, non-flow length and hot flow length are considered as design parameters. The genetic algorithm is used to maximize efficiency and minimize the total annual cost (total initial and current costs). Rao and Patel [7] used the PSO algorithm for optimization of thermodynamic and cross-flow performance of plate-fin. The minimization of the entropy production unit for the required heat requirement under the specified space constraints, the minimization of the total volume, and the minimization of the total annual cost are considered as the objective function. The results obtained from the PSO method were consistent with the results obtained from the genetic algorithm method. Najafi *et al*. [8] examined the plate-fin heat exchanger with air flow (as the ideal gas) on both sides. Several geometric variables are considered as optimization parameters with topical constraints. Two objective functions are defined including the total heat transfer rate and total annual cost for the system. Their results show that any attempt to increase the heat transfer rate would lead to an increase in total system costs, which would be undesirable. Therefore, the genetic algorithm was used for optimization. Also, Ghosh *et al*. [9] presented how to arrange the currents in a multi-threaded plate-fin heat exchanger. In this research, an attempt was made to find the proper flow pattern for achieving the maximum heat load for the specified number of properties, including the mass flow rate and the incoming temperature. Ali Khodayari Bavil, Seyed Esmail Razavi [10] to achieve high thermal performance prposed an effective way with considring a variety of techniques. They found that regarding heat transfer enhancement, circular ribs are in preference to block ones. Their numerical research investigated the heat transfer, fluid flow and thermal performance factor in the rib-roughened straight rectangular channel. Yavuz Özçelik [11] examined a large number of shell-pipe heat exchanger variables by varying a large number of heat exchanger parameters. He presented several designs of shell-pipe heat exchangers. Jiangfeng Guo *et al*. [12] presented a shell-tube heat exchanger with an optimization by the genetic algorithm. In this optimal design, the production rate of impenetrable entropy with the deduction of the entropy production ratio in relation to the heat transfer relative to the cold flow fluid temperature is used as a function of the objective to be optimized by considering the geometries of the shell-pipe heat exchanger as a constraint function. For a particular case in which the heat duty is constant, the optimal design can increase the efficiency of the heat exchanger, while the heat transfer surface is constant. Firstly, in this research, we attempted to determine the dynamical computation of a heat exchanger, a definite function for the ratio $\frac{Q}{\Delta P}$, then by maximizing the ratio $\frac{Q}{\Delta P}$ we tried to find the length and width of the compressed heat exchanger (W, L). Of course, since the genetic algorithm is set to find the minimum of functions, the minimum value for the function of the pressure drop ratio is calculated relative to the heat transfer rate.

2. Method

In the design of the heat exchanger, the heat transfer between the fluxes and also the mechanical pumping power used to overcome the fluid friction and to move it into the heat exchanger are important. In these heat exchangers that work with dense fluids, the frictional power of the worn out device is usually low compared to the heat transfer trajectory, and as a result, the frictional power consumption rarely has a decisive effect. But in the case of low-density fluids such as gases, the amount of energy used to overcome frictional power is transmitted in the form of heat, and it should be remembered that in most thermal power systems the value of mechanical energy is 4 to 10 times greater than the value of thermal energy. The surface arrangement for a double-acting heat exchanger is a round tube bundle as shown in Fig.1A. Of course, this arrangement has long been used for low-density, high-density fluids, but the only way to increase the surface density is to reduce the diameter of the pipes. An effective way to increase the surface density is to use secondary surfaces, i.e. blades, in one or two surface designs. Figure 1B shows a few rounded bevel pipes in which the annular blades are installed outside the pipes. This arrangement is often used in gas heat exchangers where the optimal design requires that the surface area on the gas side is maximized. The blades can be used in the exchange of liquid-liquid to liquid or on the liquid side of the exchange of heat to the liquid.

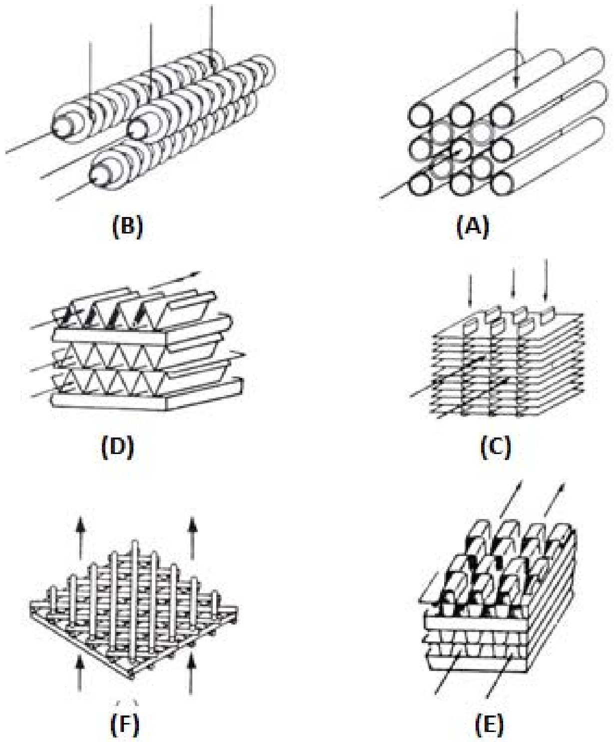

Fig.1. Sample of compressed heat exchanger surfaces [1].

In gas heat exchangers, it is necessary to have a superficial density on the sides of both fluids, and one of the methods of using the blade for this purpose is shown by testing the flat blades of Figs 1D and E. This heat exchanger is made up of layers of flat plates joined by intermediate blades. In Fig.1E the blades are replaced continuously, thus modifying the characteristics of the convective heat transfer and the friction of the flow in the way described. In the periodic heat flow exchange, energy is transmitted through convection and stored in a matrix to be released later in another charge. Figure 1E shows such a compact matrix that can be created by bundling solid bars or wired networks together. Matrices can be made by placing panels and blades, or more simply by using closed tubes. Some of the most commonly used matrix materials are made of ceramic glass materials. The heat transfer track at the unit level and for one degree of temperature difference is the same as the heat transfer coefficient h, which is calculated for the specific coefficient of the flux from the following equation

$$h = \frac{C_p \mu}{\mathrm{Pr}^{2/3}} \frac{1}{4r_h} \left(St\, \mathrm{Pr}^{2/3} \right) \mathrm{Re}\,. \tag{2.1}$$

The frictional power at the unit level can easily be obtained in terms of the Reynolds number, the coefficient of friction, and the specific characteristics of the fluid from the following equation

$$E = \frac{1}{2g_e} \frac{\mu^3}{\rho^2} \left(\frac{1}{4r_h} \right)^3 \mathrm{Re}^3\,. \tag{2.2}$$

Given the essential characteristics of the heat transfer and friction transition in terms of the Reynolds number, we can obtain the curve h in terms of E. In this case, each surface arrangement is represented by a curve for the air profile at 1atm and $500°C$ $(260°F)$. Another way of improving the efficiency of the geometric deformation of the flow pipes is to use curved or wave guides in which the separation of the boundary layer is performed. The tube bundle in which the fluid flowing along the perpendicular to the pipes has a high efficiency surface because of the formation of a new boundary layer on each tube and, as a result, the heat transfer coefficients are much larger than the coefficients of flow inside the tube. Various types of internal appendages (turbulent amplifiers) are often applied inside the tubes to increase the heat transfer coefficient, but this design does not affect the level of direct detachment under the layers of the boundary layer at the heat transfer surface.

3. Design of heat exchangers according to the effects of heat transfer and pressure drop

Designers use the theory and design data to determine the size of the heat exchanger core in a given application of heat transfer and pressure drop. It also provides designers with equations that enable them to improve their designs based on any criteria they can choose for this purpose. There are three types of exchange system as follows:

1. The common type of direct transmission, in which the two fluids exchanging heat energy are separated from each other by the heat transfer surface.
2. Transmission type - indirect with intermediate fluid, which essentially consists of two direct transfer units, which are related to each other by a heat transfer pump intermediate. The heat circulates between the heat exchangers, in which the heat energy is supplied, which is used to heat the cold chill.
3. Periodic flow type, such as the pre-heater of the common air ion-gustrom, which is a rotating matrix heat transfer surface, in which a component is periodically flowing from the front of warm and cold flames. With warm water passing through the matrix, the cold flames and the matrix are heated. On the cold side of the cycle, the cold chill is hot and the matrix is cooled down.

Figure 2 shows the desired flow system. In the case of gas exchange, the pressure variations in the intervals 1 to b, a to 2 are very low relative to the overall pressure; so $v_a \approx V_1$ and $v_b \approx V_2$.

Hence, from defining the inlet and outflow coefficients of *Ke*, *Kc* and integrating the motion-kinetic equation, the relation between the charge loss flux decay and the frequency of heat exchangers is equal to

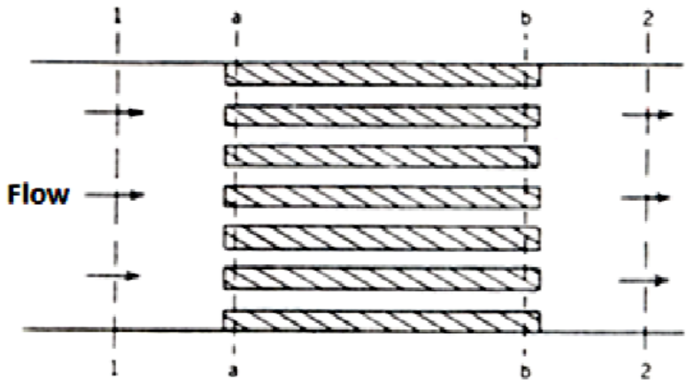

Fig.2. Model of the heat exchanger nucleus for analyzing the pressure drop. *G* is based on the lowest free flow level at the core. [1]

$$\frac{\Delta P}{P_1}=\frac{G^2}{2g_r}\frac{v_1}{P_1}\left[\left(K_c+1-\sigma^2\right)+2\left(\frac{P_2}{P_1}-1\right)+\int\frac{Av_m}{A_cV_1}-\left(1-\sigma^2-K_e\right)\frac{P_2}{P_1}\right]. \tag{3.1}$$

On the other hand, the effect of inlet and outlet drops on the friction coefficient is considered for the permeate flow on the pipe or at the wired matrix surfaces, which may be used in the periodic flow heat exchangers, is given by Eq.(3.2)

$$\frac{\Delta P}{P_1}=\frac{G^2}{2g_c}\frac{V_1}{P_1}\left[\left(1+\sigma^2\right)\left(\frac{V_2}{V_1}-1\right)+f\frac{Av_m}{A_c v_1}\right]. \tag{3.2}$$

In the case of matrix levels, the degree of porosity P is replaced by σ.
In multi-pass formations, declines in return stations should be calculated separately.
From the definition of the hydraulic radius $A/A_c=L/r_h$ and combining with Eq.(3.2)

$$\frac{G^2}{2g_c}\frac{V_1}{P_1}=\frac{\left(V_1^2/2g_c\right)}{\left(P_1/\rho_1\right)} \tag{3.3}$$

where the inlet flow rate of the core is based on the lowest free flow level that determines G. The correct mean value of the special volume for use in Eqs (2.1) and (2.2) is equal to

$$v_m=\frac{1}{A}\int_0^A v\,dA. \tag{3.4}$$

Consider the flow temperature conditions in Fig.3. For a unit amount of Cmin / Cmax, the flow rates of flux in a non-identical true flux arrangement and with a good approximation in any arrangement other than the parallel flux, are changed linearly by changing the area. So

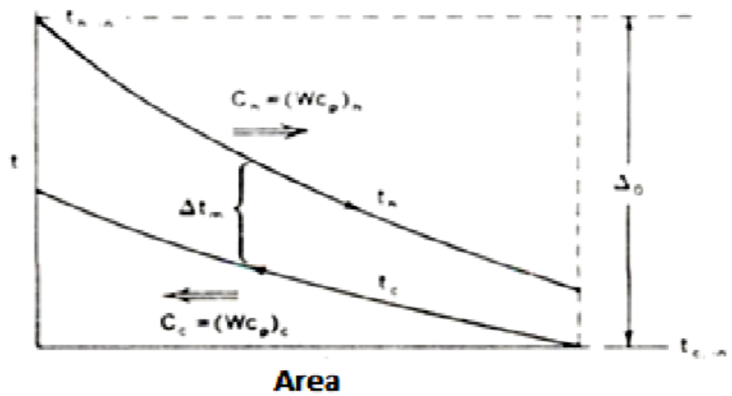

Fig.3. Distribution of temperature in a heat exchanger [1] with each flow arrangement and *Cc* >> *C*.

$$\frac{V_m}{V_1}\approx\frac{P_1}{P_{av}}\frac{T_{av}}{T_1}, \tag{3.5}$$

$$P_m\approx\frac{V_1+V_2}{2}, \tag{3.6}$$

in which T_{av}, P_{av} are the arithmetic mean of the end values. If the wall temperature is essentially uniform (Fig.3), for example, in a water cooled condenser coil with evaporation, Eq.(3.6) takes the form

$$\frac{V_m}{V_1} \approx \frac{P_1}{P_{av}} \frac{T_{lma}}{T_1}, \tag{3.7}$$

in which P_{av} is the mean of the arithmetic values of the end values and T_{lma} is related to the logarithmic mean of the temperature difference between the charge with the variable temperature and the constant temperature of the charge Δt_{lma}

$$T_{lma} = T_{const} \pm \Delta t_{lma}. \tag{3.8}$$

Here, in the case of the conditions specified in the figure, a positive sign is used

$$\Delta t_{lma} = \frac{\left(t_{h,in} - t_c\right) - \left(t_{h,out} - t_c\right)}{In\left[\left(t_{h,in} - t_c\right) / \left(t_{h,out} - t_c\right)\right]} = \frac{t_{h,in} - t_{h,out}}{N_{tu}}. \tag{3.9}$$

If t_h is constant and t_c is changed (C_h>>C_c), and the negative sign is obtained in Eq.(3.8), a similar relationship is obtained where t_c is replaced by t_h. The mass velocity G in Eq.(3.9) is based on the lowest free flow level, which is consistent with the definition of friction coefficient here. The input and output effects in Eq.(3.9) usually have a small amount of overall pressure drop in a typical swap design, because of the large size of A/A_c, the friction clause controls the size of the core. Consequently, K_c, K_e does not need to be precisely determined. In this chapter, the input and output drop coefficients are shown as a function of the flow arrangement and Reynolds number. The friction coefficient f, used in Eq.(3.9), is influenced by the change in the specification μ, ρ of the fluid in the cross-sectional flow and its variation in flux.

3.1. Method of selecting the level

In the schematic display of the heat exchanger design (Fig.4), surface features are shown as a major input in the design theory. St/f curves according to Re or $St\mathrm{Pr}^{2/3}/f$ in Re, make the geometric shaping of the surface possible, resulting in a heat exchanger with a smaller flow level.

$$G = \frac{W}{A_c} \approx \left[2g_c \frac{P_1}{V_m} \eta_\circ \left(\frac{\Delta P / P_1}{N_{tu}} \right) \left(\frac{St}{f} \right) \right]^{1/2}. \tag{3.10}$$

Clearly, the surface with the highest St/f will have the smallest level of AC flow. Therefore, using the "Chart" method, the "good coefficient" of the surface area is obtained. However, it should be noted that the front-to-side ratio to the flow level, $1/\sigma$, is present in converting A_c to the front of the core. Now, let's consider a good coefficient, that is, a good coefficient of heat transfer surface (or core volume or core mass). Equations (2.1) and (2.2) and Fig.2 are sufficient for this purpose. The geometric shape chosen for the surface located at the top of the graph h in E (for a given set of $\rho, \mathrm{Pr}, \mu, C_p$ fluid characteristics and $h\beta$ hydraulic diameter, in terms of $E\beta$ method for choosing geometric shapes, creates a surface that gives a small and often small mass core.

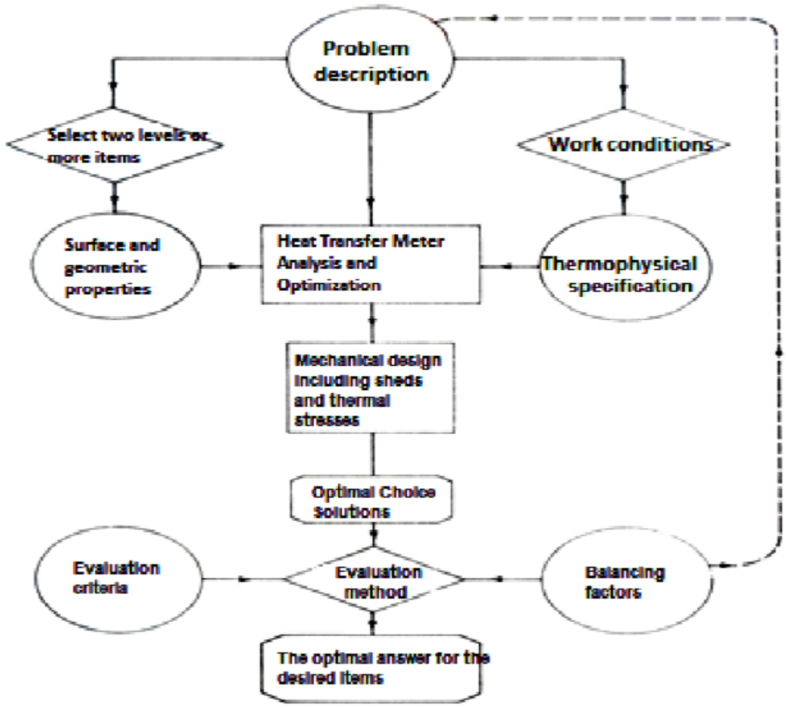

Fig.4. Heat exchange design and optimization method [1].

3.2. Geometry of the surface and core

In order to apply the basic data of heat transfer and friction of flow in the design problem, certain geometric relationships are required. Here is a special set and a special form of these relationships that prove their usefulness.

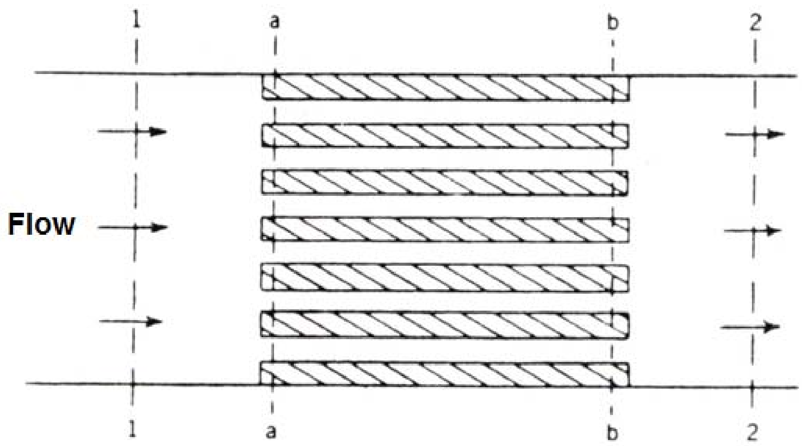

Fig.5. Heat exchange model for pressure loss analysis. *G* is based on the lowest free flow level at the core.

In fact, these dimensions are part of the basic design data for surface selection shown in Fig.5, and are:
b: the distance between the pages (only in the case of flat surfaces), *ft* or *m rh*, the hydraulic radius of the flow passage *ft*, *AcL / A* or *m;*

β: the total surface transfer ratio of one side of the exchanger to the volume between the sides of the page (only for flat panel surfaces), *ft^2/ft^3* or *m^2/m^3;*
α : the total surface transfer ratio of one side of the exchanger to the total volume of the exchanger (given only for matrix, pipe and prefabricated surfaces), *ft^2/ft^3* or *m^2/m^3;*
P: porosity, total volume / cavity volume (only for matrix surfaces).

Additionally, for a plane with a flat band, the thickness of a (*ft* or *m*) should be given separately. The following geometric coefficients are considered as the design result for each side of the complete heat exchanger core:
A - total transfer surface of one side of the exchanger, ft^2 or m^2;
A_c, - the free flow level of one side, ft^2 or m^2;
A_{fr}, - the front surface of one side, ft^2 or m^2;
L - the flow length on the one hand, *ft* or *m*;
V - the total volume of swap, ft^3 or m^3;
α : The total surface transfer ratio of one side of the exchanger to the total volume of the exchanger (this quantity is the basic given data of the matrix, tubular and tubular surfaces), ft^2/ft^3 or m^2/m^3;
σ : The exchange the free flow to the front of the one side.

The following equations give the relation between the surface and core coefficients on one side of the exchanger. The bottom 1 index points to one side and 2 to the other; coefficients without a low index are common to both sides. These relationships are applied by placing the lower index 1 instead of 2 and 2 instead of 1 on the side 2.

$$r_h = L\left(\frac{A_c}{A}\right), \tag{3.11}$$

$$\sigma_1 = \left(\frac{A_c}{A_{fr}}\right)_1 = \left(\frac{Ar_h}{LA_{fr}}\right)_1 = \frac{(Ar_h)_1}{V} = (\alpha r_h)_1, \tag{3.12}$$

$$\sigma_1 = \frac{b_1\beta_1 r_{h_1}}{b_1 + b_2 + 2a}, \tag{3.13}$$

$$\sigma_1 = \frac{A_1}{V} = \left(\frac{A}{LA_{fr}}\right)_1 = \left(\frac{\sigma}{r_h}\right)_1, \tag{3.14}$$

$$\sigma_1 = \frac{b_1\beta_1}{b_1 + b_2 + 2a}, \tag{3.15}$$

$$A_{c_1} = \left(\sigma A_{fr}\right)_1 = \left(\frac{Ar_h}{L}\right)_1 = \left(\frac{A\sigma}{L\alpha}\right)_1, \tag{3.16}$$

$$\left(\frac{r_h}{L}\right)_1 = \left(\frac{A_c}{A}\right)_1 = \left(\frac{\sigma}{L\alpha}\right)_1, \tag{3.17}$$

$$P = \frac{A_c}{A_r} = \alpha r_h, \tag{3.18}$$

$$\alpha = \frac{P}{r_h}, \tag{3.19}$$

$$A_c = PA_{rr} = \alpha r_h A_{fr} = A_L^{r_h}, \tag{3.20}$$

$$A = \frac{\alpha}{P} LA_c, \tag{3.21}$$

$$A = \alpha LA_{fr}. \tag{3.22}$$

The following relationships hold true only for matrices with cross rods and square grids

$$P = 1 - \frac{\pi}{4x_t}, \tag{3.23}$$

$$\sigma = \frac{\left(x_t - 1\right)^2}{x_t^2}, \tag{3.24}$$

$$\alpha d = \frac{\pi}{x_t}, \tag{3.25}$$

$$\frac{r_h}{d} = \frac{x_t}{\pi} - \frac{1}{4} = \frac{1}{4}\frac{\mathrm{Re}}{\mathrm{Re}_d}. \tag{3.26}$$

4. Results

The heat exchanger model is shown in Fig.5. The values of heat transfer rate and pressure drop and geometric dimensions before optimization

$$Q = 2340\ kW \qquad \Delta P = 14364\ n/m^2,$$

$$L = 0.48m \qquad w = 0.61m \qquad H = 1.62\ m.$$

The values of heat transfer rate and pressure drop and geometric dimensions after optimization

$$Q = 2760\ kW \qquad \Delta P = 5746\ \ N/m^2,$$

$$L = 0.62m \qquad w = 0.47m \qquad H = 1.62m.$$

By comparing the above results it can be seen that with the constant volume of exchanger and its height, the ratio has increased $\frac{Q}{\Delta P}$ about 4.46 times. In fixed volume, with only a small change in the width and width of the exchanger, we have been able to increase the heat transfer and reduce the air pressure drop (high pressure drop). Of course, the genetic algorithm results in different answers at each run time, which are calculated and compared only for a result of $\Delta P, Q$ values. Further, other results from the genetic algorithm are presented in Tab.1. With some reflection on the results of the genetic algorithm, we conclude that the heat transfer rate has increased by about *18*%, which means that the compressed heat exchanger's thermal efficiency has increased. Increasing the thermal efficiency can be seen at cold and hot fluid outlet temperatures, in this way, the cool-out temperature of the cold fluid is higher than the first one (i.e. before optimization) and the warm-out temperature of the hot fluid is reduced more than the first state (i.e. before optimization), which is, of course, favorable for us because we have achieved a better result with the same energy. The simulation results for the friction coefficient and Kelibron in function the Reynolds number are shown in Figs 6 and 7, respectively.

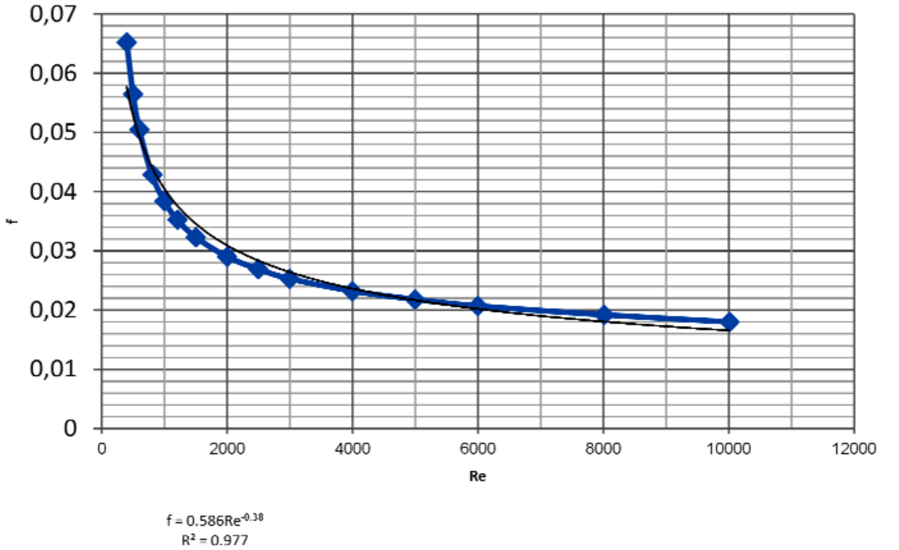

Fig.6. Changes in friction coefficient versus the Reynolds number in optimal model.

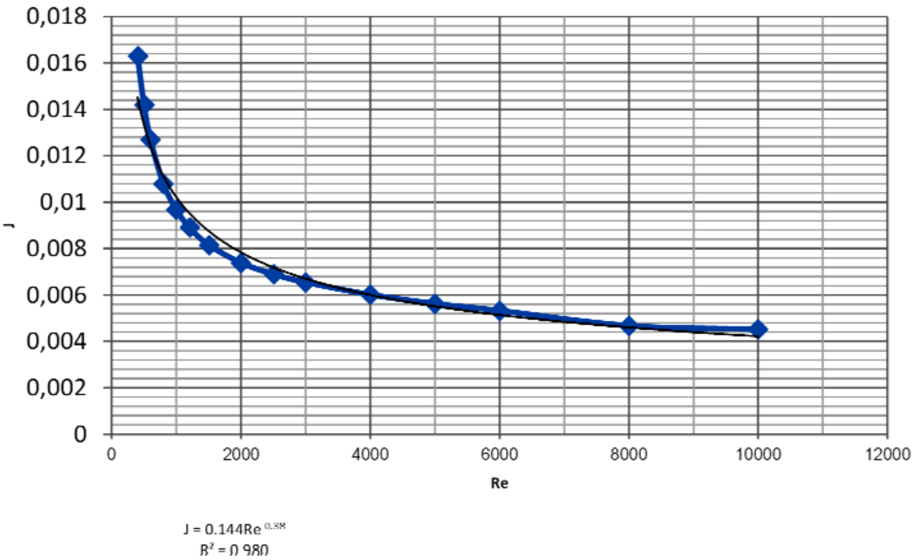

Fig.7. Changes in the Kelibron coefficient versus Reynolds number in the optimal model.

The results of the genetic algorithm show that the air pressure drop was also reduced by about *58*%, which is a stunning result. The main or, in other words, the only energy source are pumps and compressors, and since the compressors in comparison to pumps uses more energy (compressors are used for fluids in the gas phase and pumps are applied for fluids in the liquid phase. Since the specific volume of fluids in the gas phase is much greater than the specific volume of fluids in the liquid phase, so the specific work required to compress the gases is much greater than the specific work of the liquids. So the demand for the required power of compressor is much higher than the pumps.) In this paper, the air pressure drop has been checked (i.e., the side that used the compressor). Given that the compressed sweep width of the heat has decreased, the Reynolds number has increased and the friction coefficient has been reduced.

Table 1. Comparison of heat capacity results before and after optimization.

	$Q(Kw)$	$\Delta P\left(N/m^2\right)$	$L(m)$	$W(m)$	$H(m)$
Before Optimization	*2340*	*14364*	*0.48*	*0.61*	*1.62*
After Optimization	*2760*	*5746*	*0.62*	*0.47*	*1.62*

5. Conclusion

In this paper, the design and optimization of the heat exchanger has been performed to improve system efficiency. For this purpose, the thermodynamic model of the system was first introduced and then, by using the genetic algorithm, the thermal system performance was optimized. The results showed that considering the heat transfer mechanism in heat exchangers (heat exchange between two cold and hot fluids without applying external thermal energy, however, regardless of the heat transfer between the exchanger body and the environment), compressors and pumps are the main consumers of energy. Therefore, the proposed scheme has a special economic significance in constant volume and subsequently constant weight (A fixed weight indicates a constant initial cost of production). Since we know that the optimization of the current cost is primarily prior to the optimization of the initial cost (it should be noted, however, that in this scheme the initial cost of construction has remained constant), so it has been important for us to optimize the current cost. Optimization with genetic algorithm increased the exchange capacity of heat exchangers up to *15*% by significantly reducing the pressure drop.

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