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OPTIMIZATION OF BITUBULAR HEAT EXCHANGER USING GENETIC ALGORITHMS

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Abstract. *Optimization of a bitubular heat exchanger in countercurrent configuration was studied, seeking the minimum possible volume while subjected to design constraints. Geometric and operational parameters of the heat exchanger were varied using genetic algorithms with 50 individuals and maximum evolutionary chain of 100. Crossover and mutation were set to 0.8 and 0.005, respectively. Three different cases were analyzed and compared to a case study available in literature. By neglecting pressure drop it was possible to reduce volume by 73.9%. When pressure drop was restricted, heat exchanger volume decreased by 11% and average pressure drop was reduced by 95%.*

Keywords: *heat exchanger, optimization, genetic algorithms*

1. INTRODUCTION

Bitubular heat exchangers are composed of two concentric pipes: one fluid flows into the inner pipe and the other into the annular region. One way to classify bitubular heat exchangers regards its flow configuration. Parallel configuration means that both fluids enter the same end, flow the same direction and exit through the same end; in countercurrent configuration, fluids enter and exit opposite ends to each other. Given a surface heat transfer area, the countercurrent configuration has better performance. Thus, the use of parallel configuration is mostly limited to cases where the wall temperature must be kept (virtually) constant (Bergman *et al.*, 2011).

Primarily, bitubular heat exchanger are used in low capacity applications where the heat transfer surface area does not exceed 50 m^2, since its cost is proportional to surface unit (Shah and Sekulic, 2003). In addition to the manufacturing costs, it is also necessary to take into account operational costs, especially the power required to pump fluids. Pressure drop is directly related to heat transfer in the exchanger, as well as exchanger size, mechanical characteristics and operating conditions, among other factors. Therefore, pressure drop is a paramount parameter in a project economical viability analysis (Hewitt *et al.*, 1994).

Optimization, in which every possible candidate will be evaluated based on imposed requirements (Park *et al.*, 2004), must be a design step for most industries and even more so for heat exchangers. This is because their design is usually a complex trial and error process where all physical and geometric parameters are combined and interlaced to meet specific operational requirements such as thermal load, pressure drop, discharge temperature, etc (Xie *et al.*, 2008). Hence, applications of neural networks and genetic algorithms (GA) in Thermal Engineering have been receiving prominence in the search for solutions to real-world problems in the past years (Sen and Yang, 2000). Selbaş *et al.* (2006) proposed a new design method for shell-and-tube heat exchangers based on the economic view provided by GA. Xie and Wang (2006) optimized a finned plate heat exchanger, using GA to size the fin geometry, seeking minimum total weight or maximum efficiency. Imran *et al.* (2017) optimized thermal and hydraulic parameters of plate heat exchanger by GA and analyzed the effect of geometrical parameters of heat exchanger on thermal and hydraulic performance.

Genetic algorithm may be used in cooperation with computational fluid dynamics (CFD). Du *et al.* (2016) focused their studies on a double flow plate-fin heat exchanger and optimized seven geometric parameters parameters, such as the fin height, fin length and fin wrinkling angle, to reach the optimal overall structure and the results, obtained by CFD simulations, showed that the total heat-transfer rate was improved about 6.2% and the total pressure drop decreased by 40% comparing with the original design. Liu *et al.* (2017) used CFD simulation and GA optimization to develop a plate-fin heat exchanger and as a result the temperature of hot flow was reduced by 18.9% and that of the cold flow was increased by 16.1%.

In this paper genetic algorithms optimization technique was applied to minimize the total volume of a countercurrent bitubular heat exchanger while subjecting hot fluid to design constraints regarding its discharge temperature.

2. METHODOLOGY

2.1 Genetic Algorithms

Genetic Algorithms basic principles were first proposed by Holland (1992). Resembling natural selection mechanism, where better suited individuals survive in a competitive environment, in GA a potential solution is described by individuals formed by a series of parameters (Selbaş *et al.*, 2006). These parameters, genes of a chromosome, are structured by a binary sequence and each chromosome is evaluated for its fitness correlated to the object function of the problem. In this paper genes are expressed by inner and outer diameter, in addition to the mass flow rate of each fluid.

Figure 1 shows a simplified flow chart of the GA.

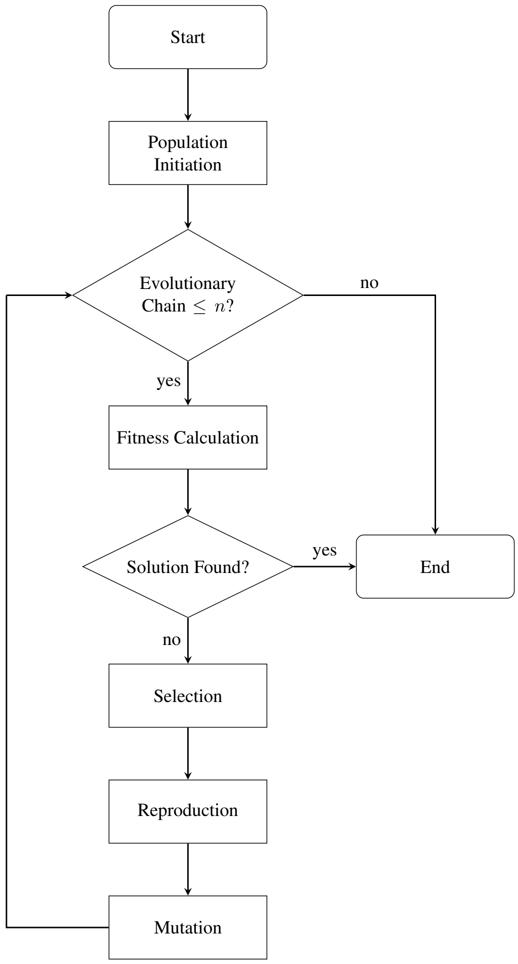


Figure 1: Flow chart of applied genetic algorithms

The algorithm begins with the creation of a random initial population. After checking whether the maximum evolutionary chain, n, has already been surpassed or not, the suitability of the individuals is assessed according to the constraints of the problem. If the solution, i.e. stop criteria, has not yet been reached, the individuals are organized according to the objective function in the Selection stage, favoring the most apt individuals.

To create the next generation in the algorithm, there are reproductive and mutagenic processes. During Reproduction, crossover operations recombine genes from two chromosomes while in the Mutation stage, one or more genes are randomly modified; which helps to overcome eventual retention at a local maximum.

For this study a population of 50 individuals and maximum evolutionary chain of 100 generations was arbitrated. Genetic operators used include roulette selection, elitism (i.e., the best individual is replicated in the new generation), uniform crossover of 0.8, and mutation probability of 0.005.

2.2 Physical Modeling

2.2.1 Thermodynamic Model

One way to analyze a heat exchanger is by logarithmic mean temperature difference method (LMTD). Assume a bitubular heat exchanger in which the cold fluid flows in the inner pipe and the hot fluid flows in the annular region. Considering only the convection between the two fluids, i.e. assuming thin tubes isolated from environment, the total heat transfer rate, q, may be determined by Eq. (1):

$$q = \dot{m}_h \, c_h \, (T_{h,i} - T_{h,o}) \tag{1}$$

Where $\dot{m}$ is mass flow rate, c is specific heat and T is temperature. The subscripts c, h, i and o stand for respectively, cold fluid, hot fluid, inside and outside.

Cold fluid outlet temperature is then calculated by Eq. (2) and mean log temperature, ΔT_m, is given by Eq. (3):

$$T_{c,o} = \frac{q}{\dot{m}_c \, c_c} + T_{c,i} \tag{2}$$

$$\Delta T_m = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\log\left(\frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right)} \tag{3}$$

Reynolds number, Re, for the annular and internal region may be obtained, respectively, Eq. (4) and (5):

$$Re_h = \frac{4\dot{m}_h}{\pi \, (D_o + D_i) \, \mu_h} \tag{4}$$

$$Re_c = \frac{4\dot{m}_c}{\pi D_i \mu_c} \tag{5}$$

Where D is the diameter and μ is viscosity.

For turbulent flow, with Reynolds above 10,000, Nusselt number,Nu, is estimated through Dittus-Boelter correlations; such as Eq. (6), which depicts a fluid being heated:

$$Nu_c = 0.023 \, {Re_c}^{0.8} \, {Pr_c}^{0.4} \tag{6}$$

Where Pr is the Prandtl number.

For laminar flows, Nusselt number is usually determined empirically. Figure 2 shows the relationship between diameter ratio and Nusselt number for oil flowing in the annular region of a bitubular heat exchanger.

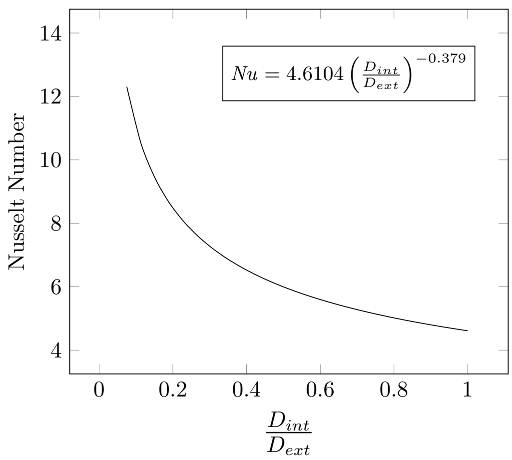


Figure 2: Nusselt as a function of ratio of diameters to oil flow

Heat transfer coefficient, h, of each fluid is defined by Eq. (7) and (8), while Eq. (9) gives overall heat transfer, U:

$$h_h = \frac{Nu_h \, k_h}{(D_o - D_i)} \tag{7}$$

$$h_c = \frac{Nu_c \, k_c}{D_i} \tag{8}$$

$$U = \frac{h_h\, h_c}{h_h + h_c} \tag{9}$$

Where k is the thermal conductivity.

Finally, the length, L, required by the heat exchange is calculated by Eq. (10):

$$L = \frac{q}{U \pi D_i \Delta T_m} \tag{10}$$

2.2.2 Fluid-dynamic Model

Friction factor depends on flow type. Equation 11 describes friction factor for laminar flows. For turbulent flows, several correlations are present in literature and, in this paper, Haaland correlation, Eq. (12), is suggested:

$$f = \frac{64}{Re} \tag{11}$$

$$\frac{1}{\sqrt{f}} = -1.8 \log \left[\left(\frac{\varepsilon/D}{3.7} \right)^{1.11} + \frac{6.9}{Re} \right] \tag{12}$$

Where f is the friction factor and ε is roughness.

For a bitubular heat exchanger in which the hot fluid flows through the annular region, the corresponding pressure drop of each fluid is given by Eqs. (13) and (14):

$$\Delta P_h = \frac{f_h\, \rho_h\, {V_h}^2\, L}{2\,(D_o - D_i)} \tag{13}$$

$$\Delta P_c = \frac{f_c\, \rho_c\, {V_c}^2\, L}{2 D_i} \tag{14}$$

In Eq. (13) and (14), ΔP is the pressure drop, ρ is density and V is fluid mean velocity.

2.3 Case Study

Geometric and operational parameters adopted in optimization are based on the following case study, adapted from (Bergman *et al.*, 2011):

> A countercurrent, bitubular heat exchanger is used to cool lubricating oil. Inner and outer pipe diameters are, respectively, 30 mm and 70 mm and pipe roughness is 0.00024 m. Mass flow of the inner pipe cooling water is 0.2 $\mathrm{kg/s}$ and oil flow rate in the annular region is 0.1 $\mathrm{kg/s}$. Oil and water enter at temperatures of 100 °C and 30 °C, respectively, and at the end of the process, oil outlet temperature must be 60 °C.

Table 1 provides all physical properties.

Table 1: Fluids' physical properties

Variable	Unit	Value
μ_c	$\mathrm{Pa\,s}$	0.000725
μ_h	$\mathrm{Pa\,s}$	0.0325
ρ_c	$\mathrm{kg/m^3}$	995.6
ρ_h	$\mathrm{kg/m^3}$	1199.7
c_c	$\mathrm{J/(kg\,K)}$	4178
c_h	$\mathrm{J/(kg\,K)}$	2131
k_c	$\mathrm{W/(m\,K)}$	0.625
k_h	$\mathrm{W/(m\,K)}$	0.138
Pr_c	–	4.85

2.3.1 Objective Function

Common design goals of a heat exchanger combine efficiency at low cost of construction and operation. This implies small volume (i.e lower weight) and controlled pressure drop; minimizing pumping costs (Xie *et al.*, 2008). Thus, in this paper, the objective function is defined by minimization of the volume of the bitubular heat exchanger, $V\!\!\!/$, Eq. (15):

$$V\!\!\!/ = \frac{\pi}{4} {D_o}^2 L \tag{15}$$

2.3.2 Limiting Bounds

Both geometric and operational parameters were considered in the optimization process. Geometric parameters consisted of inner and outer pipe diameter, in addition to pipe length. Operational parameters included mass flow control and maximum pressure drop for both fluids and minimum Reynolds number for the cold fluid (to ensure turbulent flow).

Table 2 presents range for each parameter.

Table 2: Optimization parameters

	Variable	Unit	Lower Bound	Upper Bound
	D_i	m	0.015	0.1
Geometric	D_o	m	0.025	0.15
	L	m	30	150
	ΔP_c	kPa	–	3.5
	ΔP_h	kPa	–	0.75
Operational	$\dot{m}_c$	kg/s	0.1	0.4
	$\dot{m}_h$	kg/s	0.05	0.25
	Re_c	–	10,000	–

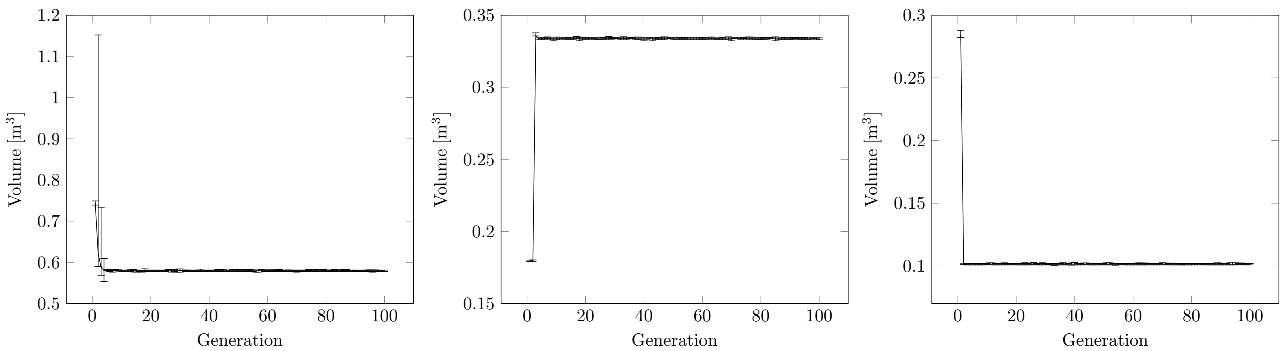
In order to evaluate the influence of the parameters on design, three different cases were proposed:

- **Case 1**: mass flow rate is kept constant at the values of 0.1 kg/s for the hot fluid and 0.2 kg/s for the cold fluid. In this case, pressure drop may not exceed maximum values specified in Tab. 2.
- **Case 2**: mass flow rate may vary within range stipulated by Tab. 2, but pressure drop is not considered in the optimization.
- **Case 3**: both mass flow rate and pressure drop are analyzed in optimization, i.e. all parameters listed in Tab. 2 are considered.

In all three cases, it is ensured that the minimum Reynolds of cooling water is 10,000.

3. RESULTS

Figure 3 shows evolutionary progression range of GA's population for cases 1, 2 and 3.



(a) Case 1 (b) Case 2 (c) Case 3

Figure 3: Evolutionary chain progression

At first glance, Fig. 3 may suggest that the choice of 100 generations was an overkill, as all results converged rapidly to the vicinity of their final value, usually near the fifth generation. This is misleading, as in reality, tight tolerances forced all simulations to the 100^{th} generation in every case. Previous attempts have taught that, depending on the random initial population, loose tolerances could lead to a local maxima entrapment which was far from the correct optimum value.

Table 3 shows the dimensions and volume of the heat exchanger for each of the three cases, in addition to the case study.

Table 3: Dimension and volume of each heat exchanger

	D_o (m)	D_i (m)	L (m)	$\forall$ (m^3)
Case Study [1]	0.070	0.030	96.8	0.372
Case 1	0.085	0.035	102.2	0.580
Case 2	0.042	0.015	71.5	0.099
Case 3	0.118	0.069	30.2	0.331

[1] Adapted from (Bergman *et al.*, 2011)

By comparing the value achieved by the objective function of the three cases to the case study heat exchanger volume, case 1 ended up with a 55% higher volume, while cases 2 and 3 reduced the volume by 73.9% and 11%, respectively. Therefore, if only the objective function is taken into account, then the heat exchanger found in case 2 is the winner.

However, this is not always the case in the industry. It is also interesting to analyze the fluid-dynamic response in each case, as seen in Tab. 4, which shows the mass flow rate and pressure drop of both fluids (hot and cold) for each case.

Table 4: Fluid-dynamic response of each heat exchanger

	$\dot{m}_c$ (kg/s)	$\dot{m}_h$ (kg/s)	ΔP_c (kPa)	ΔP_h (kPa)
Case Study [1]	0.200	0.100	5.21	1.67
Case 1	0.200	0.100	2.51	0.75
Case 2	0.400	0.063	560.61	4.43
Case 3	0.395	0.055	0.09	0.08

[1] Adapted from (Bergman *et al.*, 2011)

Table 4 makes clear why case 1 failed to minimize the heat exchanger volume. Compared to the case study, despite having the same operating parameters, case 1 must reduce the water pressure drop by 52% and oil pressure drop by 55%.

As in case 1 only the geometric parameters can be changed, the only way to meet the maximum oil pressure drop criterion is by increasing the outer diameter. Trying to make the most out of a bad situation, the first case solution minimizes volume growth by increasing the inner diameter. This increases the oil's Nusselt number and heat transfer coefficient, which in turn reduces the volume.

Regarding case 2, despite being the smallest heat exchanger found, Tab. 4 shows that the water pressure drop is more than 100 times greater than that found in the case study.

Indeed, as there was no fluid-dynamic constraint, case 2 solution lowers oil mass flow rate, since it will also lower the total heat transfer rate. Less heat transfer means a smaller heat exchanger.

A drawback of a lower oil mass flow rate is that its Reynolds number also decreases. For a turbulent flow that would mean a diminished Nusselt number, which would impair the heat transfer; but fortunately, oil's Nusselt number relies solely on inner and outer diameters (see Fig. 2).

Cases 1 and 2 are interesting because they allow us to evaluate the influence of geometric and operational parameters on the design of a heat exchanger, but the industrial application of both cases themselves is limited. Case 3 comes closer to a real case: all parameters may be modified to achieve the goal of a small and efficient heat exchanger.

Surprisingly, case 3 solution has the largest inner and outer diameter of all cases. It was seen in case 1, that increasing the outer diameter was detrimental to volume minimization, but in a sense, a price to be paid in reducing pressure drop.

To remedy the increase of the outer diameter, case 3 solution acts to shorten the pipe length and this is done threefold.

First, oil mass flow rate is reduced, which decreases the amount of total heat to be transferred. As seen in case 2, this does not affect the oil Nusselt number.

Second, the water mass flow rate is increased in order to reach a higher Reynolds number and, consequently, a higher water Nusselt number. This leads to a higher overall heat transfer coefficient.

Third, the inner diameter is increased aiming a higher oil Nusselt number and therefore a higher oil heat transfer coefficient. Increasing the inner diameter also fixes a problem that arises with increasing water mass flow rate: excessive cold fluid velocity leads to an increased pressure drop, which must be prevented.

Thus, case 3 solution ends up with a 68% shorter pipe length than the case study (which leads to a final 11% volume reduction), while reducing oil and water pressure drop by 95% and 96%, respectively.

4. CONCLUSION

Optimization of a bitubular heat exchanger was accomplished using both fluid-dynamic and thermodynamic approaches.

Geometric and operational parameters were set to fluctuate in Genetic Algorithms and three proposed cases were compared to a referential case study, in which oil flows in the annular region and cooling water flows in the inner tube.

Suggested optimization interval was satisfactory, even though some parameters reached their limit values on particular occasions: a larger range would hardly change optimization strategy or outcome.

In case 1, where limits were imposed on fluid pressure drop and only geometric parameters could be modified, minimum volume was found to be 55% greater than that of the case study. Its main solution strategy revolved around satisfying oil pressure drop condition and then enlarging inner diameter to improve heat transfer and thereby reduce pipe length.

In case 2 both geometric and operational parameters could be adjusted, while there was no restriction on the maximal fluid pressure drop. Its simple strategy of compressing both tubes diameters and slowing down oil mass flow rate guaranteed the lowest volume of 0.099 m^3, which represents 73.9% of the volume found in the case study.

Case 3 allowed geometric and operational parameters modification, while imposing limits on fluid pressure drop. Its non-trivial strategy involved speeding up water mass flow rate to ensure turbulent flow and enlargement of both diameters to meet pressure drop criteria. Despite having the largest inner and outer diameter of all cases, it was able to reduce volume in 11% in comparison to the case study.

Due to distinctive design constraints, optimization may prove to be impractical on a large scale in the heat exchanger industry. Even in this paper, in which three simple and similar cases were analyzed, there are significant differences between the three heat exchangers obtained. Therefore, optimization might ultimately be restricted on a case-by-case basis.

Moreover, gathered results are not suitable to the industry standard; it would be necessary to make geometric and/or operational changes that would divert the solution from its optimal point. Nevertheless, optimization studies should not be disregarded, because in addition to pointing the path to be chosen, they can provide good, if not optimal, industrial solutions. Even for an already active heat exchanger, it is suggested to study its operational optimization whenever possible.

Future work includes the study of non-concentric tubes, adoption of more robust thermodynamic models incorporating pipe thicknesses and heat loss with environment, as well as the analysis of the materials used in the construction of the heat exchanger.

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