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Article

Theoretical comparison of two shell-and-tube heat exchangers by applying different correlations

Comparación teórica de dos intercambiadores de calor de carcasa y tubos aplicando diferentes correlaciones

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Abstract

This paper presents the theoretical analysis of heat transfer of two different shell and tube exchangers. The theoretical study was carried out with the creation of software in the EES programming the ε-NTU method, considering different correlations for the calculations of the external and internal convective coefficients as well as their geometry for each exchanger, finally performing a mass and energy balance. The theoretical results obtained in this analysis were: operating conditions at the exit of the casing and tubes, efficiencies and total heat transferred in both exchangers, the results obtained were validated with results from other researchers.

Objetives	Methodology	Contribution
Calculate the outlet temperatures of the 2 heat exchangers with different geometric configuration using different correlations for the convective coefficients and with the ε-NTU method.	The method used was the ε-NTU to calculate the outlet temperatures of the shell and tube fluids with the help of correlations to calculate the internal and external convective coefficients and obtain the overall heat transfer coefficient.	We contributed to suggesting a method for calculating the outlet temperatures where the ε-NTU method was used, different correlations for the internal and external convective coefficients, dimensions and geometries of the heat exchangers, finally validating these results with numerical simulations in CFD .

Correlations, global transfer coefficient, internal and external convective coefficients, ε-NTU method

Resumen

En este trabajo se presenta el análisis teórico de la transferencia de calor de dos diferentes intercambiadores de tipo carcasa y tubos. El estudio teórico fue realizado con la creación de un software en el EES programando el método ε-NTU, considerando diferentes correlaciones para los cálculos de los coeficientes convectivos externos e interno, así como su geometría para cada intercambiador, por último, realizando un balance de masa y energía. Los resultados teóricos obtenidos en este análisis fueron: condiciones de operación a la salida de la carcasa y los tubos, eficacias y calor total transferido en ambos intercambiadores, los resultados obtenidos fueron validados con resultados de otros investigadores.

Objetivo	Metodología	Contribución
Calcular las temperaturas de salida de los 2 equipos de calor con diferente configuración geométrica usando diferentes correlaciones para los coeficientes convectivos y con el método ε-NTU con la ayuda del software EES y CFD.	El método usado fue el del ε-NTU para calcular las temperaturas de salida de los fluidos de la coraza y de los tubos con la ayuda de las correlaciones para calcular los coeficientes convectivos internos y externos y obtener el coeficiente global de transferencia de calor.	Contribuimos a sugerir un método para el calculo de las temperaturas de salida donde se utilizó el método ε-NTU, diferentes correlaciones para los coeficientes convectivos internos y externos, dimensiones y geometrías de los intercambiadores de calor, finalmente validar estos resultados con simulaciones numéricas en CFD.

Correlaciones, Transferencia Global de Calor, Coeficientes convectivos internos y externos, ε-NTU

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Introduction

A shell-and-tube heat exchanger is equipment used to cool a fluid that is hotter than desired or vice versa. Its design is of great importance since optimal energy transfer, size and weight are often fundamental factors that affect the economy of the process.

To design or predict the performance of a heat exchanger, it is essential to relate the total heat transfer to quantities such as the overall heat transfer coefficient, which is important as it provides the total amount of heat and depends on the external and internal convective coefficients.

Yonghua You et al., [1] carried out experimental and numerical studies applying CFD to a shell-and-tube heat exchanger with flower-shaped Baffles, within their results they obtain velocity and temperature profiles, as well as the distribution of convective coefficients as a function of the Reynolds and the global calus transfer coefficients using the k-ε turbulence model.

The error percentages between the experimental and numerical data were: 8% for the external convective coefficient and 14% for the pressure drop at a velocity in the casing of 1.2 m/s. Gh. S. Jahanmir et al., [2] performed a numerical CFD simulation for a shell-and-tube heat exchanger type by varying the inclination angles of the tubes using hot and cold water as fluids, employing the RNG-k turbulence model for the analysis of pressure drops, heat transfer, global transfer coefficient and effectiveness.

The inclination angles of the tubes that present the highest performance in the exchanger are 55° and 65°, increasing heat transfer to 69% but increasing pressure drop to 23%.

Shui Ji et al., [3] show hydrodynamic and thermal simulations performed in ANSYS CFX 12.0 for 3 types of heat exchangers with the same transfer area: the first is two passes through the shell with continuous helical baffles with one passage through the tubes; the second is with a passage through the housing with a passage through the tubes and the third is with a passage through the housing with a passage through the tubes with conventional loudspeakers.

They use the k-ε turbulence model and the equations of quantity of movement and energy, showing that the first exchanger has greater heat transfer with 25%, the second exchanger has a lower pressure drop on the shell side with respect to the others. M.M. El-Fawal et al., [4] design an economic model to minimize the area of a shell-and-tube heat exchanger using correlations to calculate the convective coefficients for shell side and tube side, and use equations for the calculation of heat exchanger geometric parameters and pressure drops. Jian-Fei Zhang et al., [5] present a numerical study in CFD and experimental of a shell-and-tube heat exchanger with helical baffles under an angle of 40°, using the k-ε turbulence model with 13.5 million cells in the mesh, at the same time that the difference between the numerical and theoretical results were 25% for the pressure drop and 15% for the Nusselt number.

Quiwang Wang et al., [6] perform a numerical CFD study of a shell-and-tube heat exchanger with helical and segmented baffles, where the working fluids are hot and cold water. Under the same mass flow, the pressure drop for the exchanger with helical baffles is 13% lower and the heat transfer is 5.6% higher than the exchanger with segmented baffles. Su Thet Mon Than et al., [7] designed a software in Mathlab and Autocad for the design of a shell and tube heat exchanger, introducing the input data which are the operating conditions of the fluids on both the shell and tube sides.

They use correlations to calculate the convective coefficients for external and internal flow, applying equations for the heat exchanger's geometric calculations and pressure drops. André L. H. Costa et al., [8] presents the minimization of the surface area of a shell and tube heat exchanger using genetic algorithms and the LMTD method, the input data are the operating conditions of the fluids and geometric data of the heat exchanger to obtain the minimum area for said process; In addition, they do not use correlations for the calculation of the global transfer coefficient.

Uday C. Kapale et al., [9] propose a model to calculate the pressure drop on the side of the casing as a function of the cut of the baffle and the arrangement of the tubes that are subsequently compared with experimental results.

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The fluids used for the model were water and oil with a Reynolds of 1x103 to 1x105 obtaining an error percentage of 2.4 to 4% with oil and 2.8 to 7.4% with water. K.C. Leong et al., [10] designed software in Delphis Programming Languaje for the design of a shell and tube heat exchanger, the input data is the operating conditions of the inlet and outlet fluids; for the calculation of the convective coefficient and the pressure drop on the shell side, they used the Bell Delaware method but do not present correlations for the convective coefficients on the tube side.

Methodology

For the model generated in the EES, the input parameters are: geometric data of the heat exchanger, mass flows, temperatures and inlet pressures of the casing and tubes. In the simulation model, the thermal properties of the fluids are determined and a calculation algorithm is applied to determine the output operating conditions of the shell and tube heat exchanger.

Four correlations were used for the analysis of internal and external flow, for internal flow are: Colburn, Petukov-Kirillov, Dittus Boelter and Gnielinski; for external flow are: Zukauskas, Kern, Hilpert and Taborek for turbulent flow conditions without phase change.

Internal Flow

The convective coefficient of heat transfer can be determined with a correlation for Nusselt's number, Colburn [11] with a Re>10000:

$$Nu_D = \frac{h_i D_{int}}{k} = 0.023\, Re_D^{4/5}\, Pr^{1/3} \qquad [1]$$

The Dittus-Boelter [12] correlation is a slightly different version than that of Colburn with a Re>10000:

$$Nu_D = \frac{h_i D_{int}}{k} = 0.023\, Re_D^{4/5}\, Pr^{0.4} \qquad [2]$$

The following expression is proposed by Gnielinski [13], where all properties are evaluated at the mean temperature with a Reynolds interval between 3000<Re<5x106:

$$Nu_D = \frac{h_i D_{int}}{k} = \frac{\left(\frac{f}{8}\right)[Re_D - 1000]Pr}{1 + 12.7\left(\frac{f}{8}\right)^{1/2}\left(Pr^{2/3} - 1\right)} \qquad [3]$$

$$f = (0.79 Ln Re_D - 1.64)^{-2}$$

The expression proposed by Petukov-Kirillov [14], shows a more complex form for this analysis, involving a friction factor but which in turn has a lower error compared to that of Dittus Boelter and Colburn, with a D>2100:

$$Nu_D = \frac{h_i D_{int}}{k} = \frac{\left(\frac{f}{2}\right)Re_D\, Pr}{1.07 + 12.7\left(\frac{f}{2}\right)^{1/2}\left(Pr^{2/3} - 1\right)} \qquad [4]$$

$$f = (1.58 Ln Re_D - 3.8)^{-2}$$

The Reynolds number is defined as:

$$Re_D = \frac{u \cdot \rho \cdot D_{int}}{\mu} \qquad [5]$$

And Prandtl's number is:

$$Pr = \frac{\mu \cdot Cp}{k} \qquad [6]$$

External Flow

The correlation of Zukauskas [15] is determined by the following expression:

$$Nu_D = \frac{h_o D_{ext}}{k} = C\, Re_D^m\, Pr^{0.6}\left(\frac{Pr}{Pr_s}\right)^{1/4} \qquad [7]$$

Where the coefficients "C" and "m" can be estimated according to the Reynolds number:

$$10 < Re \le 100 \rightarrow C = 0.9, m = 0.4$$
$$100 < Re \le 1000 \rightarrow C = 0.683, m = 0.466$$
$$1000 < Re \le 2x10^5 \rightarrow C = 0.35, m = 0.65$$

The estimation of the Reynolds number comes from the previous knowledge of the flow velocity, this in turn of a passage area involving the geometric characteristics of the exchanger, is expressed by the following equation:

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$$A_{paso} = \left[\frac{L_{carc}}{N_{baf}+1}\right]\left[D_{intcarc} - \left(\frac{D_{intcarc}}{P_t} - 1\right)D_{ext}\right] \quad [8]$$

Another methodology of analysis for the external convective coefficient on an array of tubes is proposed by Kern [16], based on an equivalent diameter, applying the following correlation:

$$\frac{h_o D_e}{k_c} = 0.36\left(\frac{D_e G_s}{\mu_c}\right)^{0.55}\left(\frac{Cp_c \mu_c}{k_c}\right)^{1/3}\left(\frac{\mu_c}{\mu_w}\right) \quad [9]$$

Where, Gs, is the mass velocity of the fluid on the casing side and the properties are evaluated at medium temperature; The equivalent diameter for a triangular array is obtained as follows:

$$D_e = \frac{4\left(\frac{P_t^2\sqrt{3}}{4} - \frac{\pi D_{ext}}{8}\right)}{\frac{\pi D_{ext}}{2}} \quad [10]$$

The magnitude of the mass velocity, Gs, can be defined over an area of cross-flow arrangement, represented by an area for maximum flow. It is obtained by the following expression:

$$G_s = \frac{m_{carc}}{A_s} \quad [11]$$

$$A_s = \left[1 - \frac{D_{ext}}{P_t}\right] D_{intcarc}\left(\frac{L_{carc}}{N_{baf}+1}\right) \quad [12]$$

Another method for the convective coefficient is proposed by Taborek [17], where the Reynolds number is based on the diameter of the tubes and the velocity of the fluid over the cross-flow area of the carcass diameter.

$$Nu_D = \frac{h_o D_{ext}}{k} = 0.2\,\mathrm{Re}_s^{0.6}\,\mathrm{Pr}^{0.4} \quad [13]$$

$$\mathrm{Re}_s = \frac{m_{carc} D_{ext}}{A_s \mu}$$

Hilpert's correlation [18]:

$$Nu_D = CRe_D^m Pr^{0.33} \quad [14]$$

Re_D	C	m
0.4–4	0.989	0.330
4–40	0.911	0.385
40–4000	0.683	0.466
4000–40,000	0.193	0.618
40,000–400,000	0.027	0.805

Representing the global coefficient with the internal and external convection phenomena, the following equation [19] is obtained:

$$U = \frac{1}{\frac{D_{ext}}{D_{int} h_i} + \frac{D_{ext} Ln\left(D_{ext}/D_{int}\right)}{2k_{mat}} + \frac{1}{h_o}} \quad [15]$$

The total heat transfer area can also be formulated in terms of the length of the tubes, number of tubes and the inner diameter of the tubes.

$$A_{total} = \pi D_{int} L N_{tub} \quad [16]$$

The thermal analysis is based on the ε-NTU method, the efficacy and NTU is estimated as follows,

$$NTU = \frac{UA_{total}}{C_{min}} \quad [17]$$

Effectiveness is determined by considering countercurrent flow:

$$\varepsilon = \frac{1 - \exp[-NTU(1-Cr)]}{1 - Cr\exp[-NTU(1-Cr)]} \quad [18]$$

$$Cr = \frac{C_{min}}{C_{max}}$$

To calculate the total heat of the heat exchanger is determined as a function of efficiency, minimum capacitance and fluid inlet temperatures, as expressed in the following equation:

$$Q_{max} = \varepsilon \cdot C_{min}(T_{ec} - T_{et}) \quad [19]$$

Once the total heat is determined, the outlet temperatures of the casing-side and tube-side fluids can be estimated using an energy balance as shown in the following equation:

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$$Q_{total} = m_{carc}Cp_{carc}(T_{sc} - T_{ec}) \quad [20]$$
$$Q_{total} = m_{tubos}Cp_{tubos}(T_{et} - T_{st})$$

The pressure drop on the side of the tubes is calculated with the following equation [20]:

$$\Delta P_t = \left(\frac{4fLNp}{Di_{tub}} + 4Np\right)\left(\frac{\rho v^2}{2}\right) \quad [21]$$

$$f = (0.79LnRe_D - 1.64)^{-2}$$

For the Shell side, the following equation is used [20]:

$$\Delta P_s = \frac{fG_S^2(Nb+1)D_S}{2\rho D_e\left(\frac{\mu}{\mu_W}\right)^{0.14}} \quad [22]$$

$$f = \exp\,(0.576 - 0.19LnRe_s)$$

To validate the model proposed in this work, a simulation was carried out in the EES software, varying the mass flows at different inlet temperatures, while the mass flow and the temperature of the mixture remain constant

Table 1 shows the geometric parameters that are required to obtain the results of the internal and external convective coefficients, the total heat transfer area and the pressure drops of both the shell or shell side and the tube side.

Box 1

Table 1

Geometric data of the heat exchanger

Internal diameter of the tubes.
Extenal diameter of the tubes.
Number of bafles
Number of tubes.
Length of tubes
Radio Pitch
Number of tube steps
Material conductivity

Source Own.

The shell-and-tube type heat exchangers shown in Figure 1 are designed to heat water with a simple and flexible installation. Heat exchangers can be connected to a boiler, heat pump, solar panel systems, or other heat source. The working fluids are cold water at the inlet on the tube side and hot water on the shell side.

Box 2

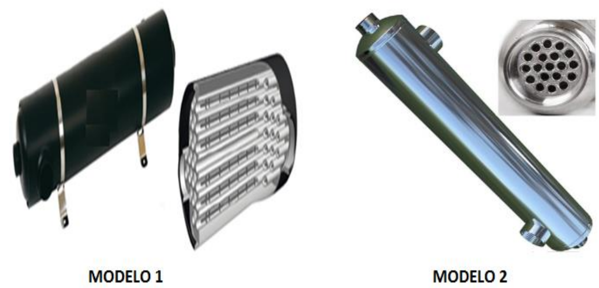

Figure 1

Shell and Tube Heat Exchangers

Source Own.

Table 2 shows the geometric data of model 1 with a transfer area of 2.65 m^2 proposed to carry out the present study and model 2 with a transfer area of 2.96 m^2.

Box 3

Table 2

Geometric data of the exchanger, shell and tubes, design 1 and 2

	Design1	Design 2
Length of the tubes (m)	1.2	1.6
Outer diameter of tubes (m)	0.02	0,031
Inner diameter of the tubes (m)	0.019	0.03
Number of Shell Steps	1	1
Number of baffles	6	8
Number of tube passes	1	1
Number of tubes	37	19
Distance between tube centers (m)	0.05	0.045

Figures 2 and 3 show a window of the program where the geometric parameters of the equipment, volumetric flow, the inlet temperatures of both fluids, and the geometric data of the heat exchanger are entered.

The simulation model determines the thermal properties of the fluids and applies a calculation algorithm to determine the output operating conditions of the shell-and-tube heat exchanger.

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Box 4

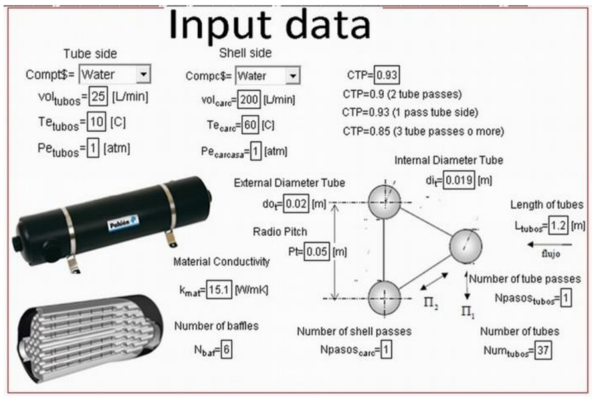

Figure 2

Shell and Tube Heat Exchangers 1

Source Own.

Box 5

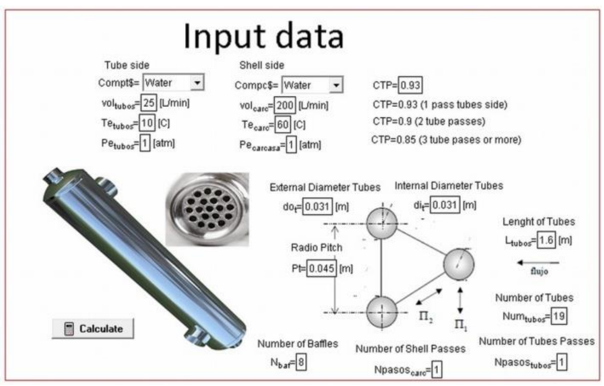

Figure 3

Shell and Tube Heat Exchangers 2

Source Own

Table 3 shows the volumetric flows and inlet temperatures of both fluids from the 3 simulations, then make comparisons of the results. It is observed that the flow on the side of the tubes is 25 L/min and on the side of the shell is 200 L/min, the 3 simulations show the different inlet temperatures.

The numerical study of the flow developed over a shell and tube heat exchanger requires a mathematical presentation of the turbulent motion of the fluid, which in turn can be transformed into an algorithm for its solution. This mathematical representation is summarized in a set of equations for the conservation of mass, momentum and energy; as well as the k-ε turbulence model [21].

Box 6

Table 3

Input data of volumetric flows and temperatures from the 3 simulations with the EES software

Shell side (L/min)	200
Tubes side (L/min)	25

Simulation 1		Simulation 2		Simulation 3	
Tet °C	10	Tet °C	20	Tet °C	10
Tec °C	60	Tec °C	70	Tec °C	45

The CFD preprocessor is used to create the geometry and meshing of the heat exchanger. Figures 4 and 5 show the geometries of the exchangers whose meshes are composed of 4 million nodes with tetrahedral elements to represent the computational domain. The computational process is carried out by applying boundary conditions such as temperature and inlet mass flows for both fluids to solve the 3D model with double precision.

Box 7

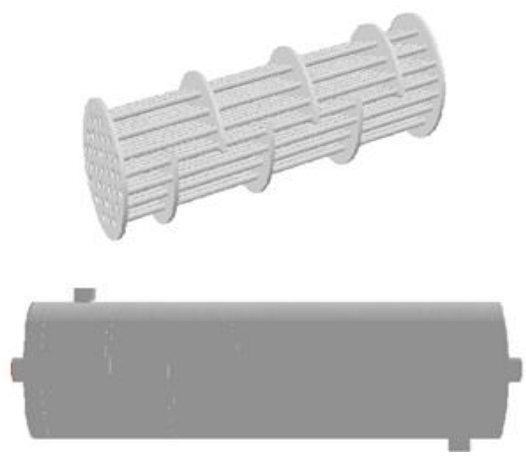

Figure 4

Shell and Tube Heat Exchangers 1

Box 8

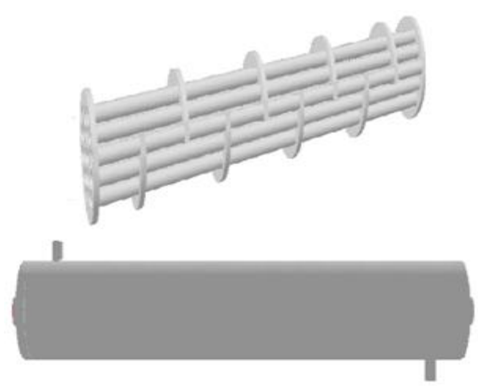

Figure 5

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Results

Simulation 1

Table 4 shows the results of simulation 1, showing the results comparisons of the outlet temperatures with the different combinations of correlations.

Box 9

Table 4

Results of simulation 1 of model 1 in EES software

Simulation 1				
hi (W/m-K)		Design 1 ho(W/m-K)		ΔP Tubes (Pa)
Petukov	535.4	Zukauskas	548.1	64.19
Colburn	409	Hilpert	576	ΔP Shell (Pa)
Gnielinski	72.83	Taborek	1881	103.4
Dittus Boelter	479.3	Kern	1097	
		U (W/m-K)	Q(W)	Efficiency (E)
Dittus Boelter-Zukauskas		246.6	26722	0.3
Gnielinski-Kern		64.94	8146	0.1
Petukov-Taborek		395	35247	0.43
Gnielinski-Zukauskas		61.31	7713	0.08
Colburn-Hilpert		230.2	25265	0.29
		Tst	Tsc	
Dittus Boelter-Zukauskas		25.32	58.08	
Gnielinski-Kern		14.67	59.41	
Petukov-Taborek		31.92	57.21	
Gnielinski-Zukauskas		14.62	59.44	
Colburn-Hilpert		24.48	58.16	

Figure 6 and Table 5 shows the temperature results and heat transfer contours of model 1 as a result of simulation 1. The temperature changes are observed on the shell side where the change is smaller, the hot fluid enters at 60 °C (333 K) and exits at 57.29 °C (330.29 K); on the other hand, the cold fluid enters at 10 °C (283 K) and exits at 31.64 °C (304.64 K) observing that the temperature change was greater inside the tubes.

Box 10

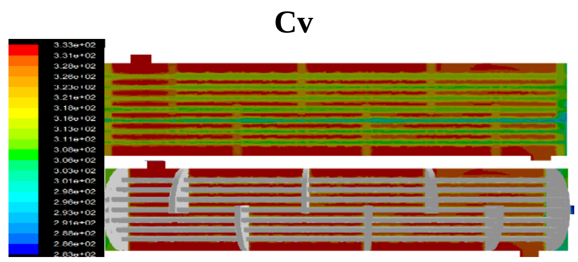

Figure 6

Figure 7 shows the flow path on the shell side as it passes through the exchanger where it can be seen that there is no stagnation near the baffles, which is very important for more effective heat transfer.

Box 11

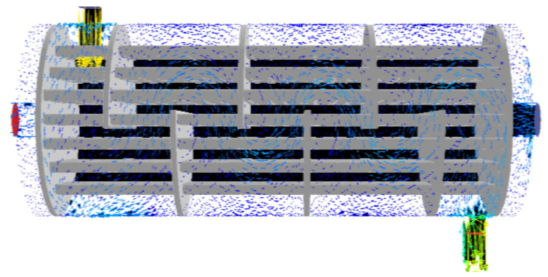

Figure 7

Shell and Tube Heat Exchangers 1. Flow trayectories

Source Own.

Box 12

Table 5

Results of simulation 1 for the model 1 in CFD.

	CFD
Tst °C	31.64
Tsc °C	57.29

The numerical and theoretical results of simulation 1 for the model 1, where the inlet temperature on the tube side is 10 °C and on the shell side 60 °C, are shown. It is observed that the theoretical results obtained with the combinations of the Petukov-Taborek coefficients are close to the numerical results in CFD.

Table 6 shows the results of simulation 1, showing the results comparisons of the outlet temperatures with the different combinations of correlations.

Box 13

Table 6

Results of simulation 1 of model 2 with the EES software

Simulation 1				
hi (W/m²-K)		Design 2 ho(W/m²-K)		ΔP Tubes (Pa)
Petukov	397.7	Zukauskas	1737	32.16
Colburn	306.6	Hilpert	1862	ΔP Shell (Pa)
Gnielinski	120.2	Taborek	2313	528
Dittus Boelter	359.9	Kern	1958	
		U (W/m²-K)	Q (W)	Efficiency (E)
Dittus Boelter-Zukauskas		286.7	32795	0.376
Gnielinski-Kern		109.4	14625	0.166
Petukov-Taborek		326.4	36108	0.41
Gnielinski-Zukauskas		107.8	14532	0.166
Colburn-Hilpert		253.6	29832	0.34
		Tst	Tsc	
Dittus Boelter-Zukauskas		28.8	57.61	
Gnielinski-Kern		18.38	58.93	
Petukov-Taborek		30.7	57.3	
Gnielinski-Zukauskas		18.33	58.94	
Colburn-Hilpert		27.1	57.82	

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Figure 8 shows the temperature contours and velocity profiles of the fluid in model 2 as a result of simulation 1. It can be seen that the temperature of the hot fluid decreases slightly to 57.26 °C (330.26 K), but the cold fluid exits at 29.31 °C (302.31 K). Making a numerical comparison between the two exchangers, the outlet temperature of the cold fluid is higher than in model 2. It can also be seen that the mass flow path from the shell side flows without any problem along the exchanger as shown in Figure 9. Making a comparison between theoretical and numerical results, it is observed that the combinations of Petukov-Taborek, Dittus Boelter-Zukauskas and Colburn-Hilpert correlations approximate the CFD results in the Table 7.

Box 14

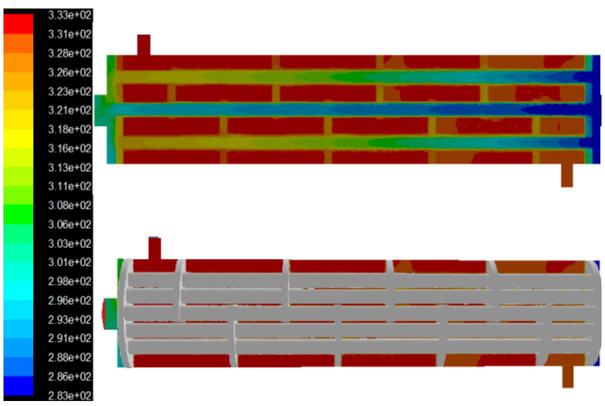

Figure 8
Shell and Tube Heat Exchangers 2. Temperature contours in K.

Source Own.

Box 15

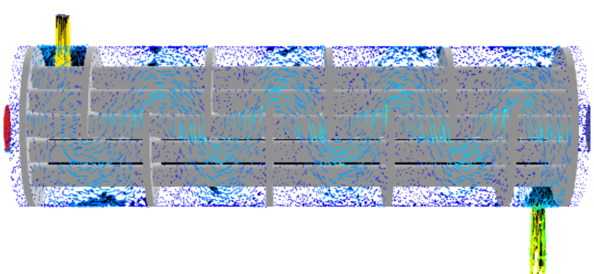

Figure 9
Shell and Tube Heat Exchangers 2.Flow trayectorie

. *Source Own*

Box 16
Table 7
Results of simulation 1 for the model 2 in CFD

	CFD
Tst °C	29.31
Tsc °C	57.26

Simulation 2

Table 8 shows the results of simulation 2 on design 1 using the different correlations; Table 9 shows the CFD results. The CFD results are 39.5 °C for the tube side and 67.5 °C for the shell side.

The outlet temperatures obtained with the combination of Dittus Boelter-Zukauskas correlations were 36.84 °C for the tube side and 67.5 °C for the shell side, these results are close to those of the CFD results.

Box 17
Table 8
Results of simulation 2 of model 1 with the EES software

Simulation 2				
		Design 1		
hi(W/m²-K)		ho(W/m²-K)		ΔP Tubes (Pa)
Petukov	611.2	Zukauskas	611.7	59
Colburn	472.7	Hilpert	604	ΔP Shell (Pa)
Gnielinski	208.3	Taborek	1945	99.54
Dittus Boelter	542.5	Kern	1146	
	U(W/m²-K)	Q(W)	Efficiency (E)	
Dittus Boelter-Zukauskas	277.1	29303	0.33	
Gnielinski-Kern	167.8	19344	0.22	
Petukov-Taborek	440.5	41217	0.47	
Gnielinski-Zukauskas	148.8	17415	0.2	
Colburn-Hilpert	255.3	27460	0.31	
	Tst °C	Tsc°C		
Dittus Boelter-Zukauskas	36.84	67.85		
Gnielinski-Kern	31.12	68.58		
Petukov-Taborek	43.7	67		
Gnielinski-Zukauskas	30	68.7		
Colburn-Hilpert	35.78	68		

Box 18
Table 9
Results of simulation 2 for the model 1 in CFD

	CFD
Tst °C	39.5
Tsc °C	67.5

Tables 10 and 11 show the results of simulation 2 on design 2. The CFD results obtained for the outlet temperatures were 41 on the tube side and 67 on the shell side.

The results obtained with the combination of correlations of the outlet temperatures and approximated the numerical ones were Dittus Boelter-Zukauskas (40.6 on the tube side and 67.37 ^oC on the shell side) and Petukov-Taborek (42.53 ^oC on the tube side and 67.13 ^oC on the shell side).

Box 19

Table 10

Results of simulation 2 of model 2 with the EES software.

Simulation 2				
		Design 2		
hi (W^2/m-K)		ho(W/^{2}m-K)		ΔP Tubes (Pa)
Petukov	454.1	Zukauskas	1939	29.81
Colburn	354.1	Hilpert	1951	ΔP Shell (Pa)
Gnielinski	213.1	Taborek	2393	508.3
Dittus Boelter	406.3	Kern	2045	
	U (W/m^2-K)	Q (W)	Efficiency (E)	
Dittus Boelter-Zukauskas	323.4	35837	0.41	
Gnielinski-Kern	186.2	23214	0.26	
Petukov-Taborek	366.7	39205	0.45	
Gnielinski-Zukauskas	185.2	23117	0.26	
Colburn-Hilpert	288.6	32936	0.378	
	Tst ^oC	Tsc ^oC		
Dittus Boelter-Zukauskas	40.6	67.37		
Gnielinski-Kern	33.34	68.3		
Petukov-Taborek	42.53	67.13		
Gnielinski-Zukauskas	33.29	68.31		
Colburn-Hilpert	39	67.59		

Box 20

Table 11

Results of simulation 2 for the model 2 in CFD

	CFD
Tst ^oC	41
Tsc ^oC	67

Simulation 3

Tables 12 and 13 show the results obtained in simulation 3 with design 1. The results obtained in CFD are close to the results obtained with the Petukov-Taborek correlations.

Box

Table 12

Results of simulation 3 of model 1 with the EES software.

Simulation 3				
		Design 1		
hi (W/m^2-K)		ho(W/m^2-K)		ΔP Tubes (Pa)
Petukov	535.4	Zukauskas	473.9	64.19
Colburn	409	Hilpert	529.7	ΔP Shell (Pa)
Gnielinski	72.83	Taborek	1770	107.6
Dittus Boelter	479.3	Kern	1038	
	U(W/m^2-K)	Q(W)	Efficiency (E)	
Dittus Boelter-Zukauskas	230.4	17699	0.29	
Gnielinski-Kern	64.72	5684	0.1	
Petukov-Taborek	389.9	26531	0.43	
Gnielinski-Zukauskas	60.25	5311	0.09	
Colburn-Hilpert	222.4	17195	0.28	
	Tst ^oC	Tsc^oC		
Dittus Boelter-Zukauskas	20.15	43.72		
Gnielinski-Kern	13.26	44.59		
Petukov-Taborek	25.21	43		
Gnielinski-Zukauskas	13	44.62		
Colburn-Hilpert	19.86	43.75		

Box

Table 13

Results of simulation 3 for the model 1 in CFD.

	CFD
Tst ^oC	26.7
Tsc ^oC	42.3

Tables 14 and 15 show the results obtained in simulation 3 with design 2. The results obtained in CFD are close to the results obtained with the Petukov-Taborek and Dittus Boelter-Zukauskas correlations.

Box

Table 14

Results of simulation 3 of model 2 with the EES software.

Simulation 3				
		Design 2		
hi (W/m^2-K)		ho(W/m^2-K)		ΔP Tubes (Pa)
Petukov	397.7	Zukauskas	1502	32.16
Colburn	306.3	Hilpert	1712	ΔP Shell (Pa)
Gnielinski	120.2	Taborek	2177	549.6
Dittus Boelter	359	Kern	1852	
	U (W/m^2-K)	Q (W)	Efficiency (E)	
Dittus Boelter-Zukauskas	279.5	22520	0.37	
Gnielinski-Kern	109.1	10209	0.16	
Petukov-Taborek	323.5	25119	0.41	
Gnielinski-Zukauskas	107.6	10084	0.16	
Colburn-Hilpert	250.6	20693	0.33	
	Tst ^oC	Tsc ^oC		
Dittus Boelter-Zukauskas	22.91	43.37		
Gnielinski-Kern	15.85	44.26		
Petukov-Taborek	24.4	43.18		
Gnielinski-Zukauskas	15.78	44.27		
Colburn-Hilpert	21.86	43.5		

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Box

Table 15

Results of simulation 3 for the model 2 in CFD

	CFD
Tst °C	25.64
Tsc °C	43.06

Conclusions

The results of the outlet temperatures of both fluids obtained with the Petukov-Taborek correlation combinations fit well with the numerical results in CFD for the proposed heat exchanger models. However, with the other combinations of correlations good results were not obtained because the internal Gnielinski and Colburn convective coefficients turned out to have very small values which did not fit any combination with the external coefficients.

The causes that affected these results were Reynolds number and the thermal properties evaluated at the operating temperatures of the exchanger, which shows that not all correlations are adjustable to the heat exchanger models. Numerical simulation was very important, since it helped to obtain reliable results from the heat exchangers and to validate the theoretical results using different correlations.

Meshing was essential, the finer the mesh, the more reliable the results, but a computing resource with a large capacity is also required to perform the modeling. Both heat exchanger models proposed to heat the water in a swimming pool are adequate; However, model 1 is more effective than model 2. When cold water enters both exchangers at the same temperature, in model 1 it comes out hotter than in model 2, with a difference of approximately 2 degrees.

Conflict of interest

The authors declare no interest conflict. They have no known competing financial interests or personal relationships that could have appeared to influence the article reported in this article.

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