FULL FACTORIAL EXPERIMENTAL DESIGN AND COMPUTER SIMULATION APPLIED TO SHELL AND TUBE HEAT EXCHANGER SETUP

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Received: July 06, 2023
Reviewed: Jule 16, 2023
Accepted: July 28, 2023

ABSTRACT

In the operation of heat exchangers there are some variables to be controlled, making it difficult to found optimized parameters. The aim of this study was to compare the experimental and simulated values of the outlet temperatures, as well as to understand the influence of operating variables for the equipment. It was used a shell and tube heat exchanger didactic module, with a constant cold fluid flow rate equal to 1.4 L.min 1. The experiments were carried out on the basis of a 2^2 full factorial experimental design with central points, as well as computer simulations in the steady state and transient regime. The higher values for heat exchange overall heat transfer coefficient determined was around 250 W.m 2.K 1. Thus, the flow regime affects the evaluated response. In addition, the computer simulation in the permanent regime presented less relative deviation. Therefore, it can be seen that although the simulations show results close to the experimental ones, there are still associated errors that should be studied and minimized, since factors such as bubble formation were not considered in the simulations. Thus, it was found that computer simulations can be used to understand the operation of heat exchangers, but they are limited to real phenomena that are not considered in theoretical mathematical models. Therefore, this study elucidates the application of statistical and computer-assisted methods as a tool to comprehend heat exchangers behavior for industrial and didactic purposes.

Keywords: design of experiments; unit operations; overall heat transfer coefficient

NOMENCLATURE

3-D three-dimensional
A heat exchange area, m²
a1-a4 finite difference method variables
c1-c5 constants of the specific heat equation
coef overall coefficient of heat exchange, J/(s.dm^2.K)
c_p fluid specific heat at constant pressure, J/(g.K)
deltat time variation, s
deltaTml logarithmic average between fluids, K
deltaz variation in space
DF degrees of freedom
F Fischer-Snedecor test
FA lack of Fit
flow_q hot fluid flow rate, dm^3/s
F_{tab} F tabulated
H0 hypothesis of a significant difference between the variables
i,j vector positions
K number of factors in the experimental design
m fluid flow rate, g/s
MLTD mean logarithmic temperature difference, K
MM_water molar mass of water, g/mol
MS mean square
z vector length
PE pure error
Q or q heat exchanged, J/s
R² coefficients of determination, %
ro density as a function of fluid temperature, g/dm^3
SS sum of squares
SV source of variation
T temperature, °C
t time, s
T_q hot fluid setpoint temperature, °C
U overall heat exchange coefficient, W/(m^2.K)
V or v volume of fluid, dm3
Vq hot fluid inlet flow rate, L/min
x differential function temperature variable
x_1 setpoint temperature, °C
$x_1.x_2$ interaction between factors
x_2 hot fluid flow, dm^3/s
Y correction factor, admensional
z vector space

Greek symbols

α significance level

Subscripts

c or C cold fluid
h or H hot fluid
in or 0 input stream
lack of fit/pure error F calculated from the lack of fit
regression/residuals F calculated from the regression
Reg Regression
Res Residuals
S or out output stream

INTRODUCTION

Heat exchangers can be used in operations involving heating, cooling and evaporation (Patel, 2023). For this reason, they are the subject of study in various technological courses, especially in the field of engineering. There are different configurations and models of heat exchangers, including the shell-and-tube type. This configuration is widely used in engineering operations and belongs to a classification called tubular heat exchangers (Sadeghianjahromi and Wang, 2021).

The shell-and-tube exchanger consists of a shell through which the cold fluid normally passes, and tubes that serve as a passage for the hot fluid (Roy and Majumder, 2019; Roy et al., 2017). Between the shell and the tubes, there are structures called baffles. These structures ensure that the fluid moves turbulently in the hull, which increases the thermal exchange between the surface of the tubes and the cold fluid. The fluid that enters the shell of the heat exchanger travels the entire length of the equipment, exiting at the other end. The fluid entering through the tubes runs the length of the tubes to the outlet end (Kücük, 2023).

When evaluating or predicting the performance of heat exchangers, it is necessary to obtain the relationship between the total heat transfer rate and the total surface area of heat exchange. The inlet and outlet temperatures of the hot and cold fluids, the overall heat transfer coefficient and other parameters. In addition, for certain systems, some authors consider that: there is no loss of energy to the surroundings, the equipment is operated in a steady state, the changes in potential and kinetic energy are negligible. There is no change in the phase of the fluids and the heat capacities (of the hot and cold fluid) do not change with temperature (Beyne et al., 2023). For the analysis of experiments, the effects that varying a factor has on the output variable are considered, with the aim of optimizing the process. In addition, it is possible to obtain a model that describes the behavior of the system within the experimental space (Carabajal et al., 2020).

Experimental design helps to optimize systems with more than one independent variable, taking into account the effect of these variables on the dependent variable. Among the types of factorial experiments is the 2^k, which occurs when you have k factors (temperature and flow rate, for example) in a system and two levels (+1 and -1, for example). This type of design is conveniently used for teaching purposes and in tests that are carried out in laboratory environments (Grangeia et al., 2020; Narenderan et al., 2019). This model is a first and/or second order polynomial equation that can contain the relationship between linear, quadratic and interaction effects (Lee, 2019). For the model to be valid, it needs to be statistically significant, not have very high errors and acceptable coefficients of determination (R^2) (Carabajal, 2020).

The main focus of this study was therefore to estimate the main effects and interaction on heat transfer between hot and cold water by evaluating the overall heat transfer coefficient (U) of a shell-and-tube heat exchanger teaching module. In addition, this study aims to promote the application of computer simulations and experimental planning for teaching purposes in equipment operation.

EXPERIMENTS

The study was carried out at the unit operations laboratory at Federal University of Jequitinhonha and Mucuri Valleys (Diamantina, Brazil). The didactic module of the shell-and-tube heat exchanger (UpControl) was used for the study. The equipment has an acrylic shell, seven U-shaped tubes made of AISI 304 steel, two passes in the shell, two passes in the tubes, two baffles and a heat exchange area of 0.15 m². The equipment is controlled by a control panel and operated by software, which records the temperature measured by the eight sensors distributed around the shell and tubes every minute. In this work, the heat exchanger was operated in counter-current flow.

There are eight temperature sensors in the heat exchanger, four for the hot fluid and four for the cold fluid. The feed water for the shell came from the cold-water tank (approximately 23ºC) and for the tubes it came from the hot tank (initial temperature approximately 23ºC). After each tank, there was an aquarium pump and a flow meter (L/min). The fluid directed to the tubes passed through the heater, previously programmed for setpoint heating.

The heat exchange process was evaluated by experimental design with the variables: hot fluid setpoint temperature (T_q) and hot fluid inlet flow rate (V_q). The design used was the 2² Full Factorial, with three central points and one replicate, in order to discuss the associated pure error. The independent variables were evaluated according to the levels (1, 0 and 1) shown in Table 1. The design carried out for this work is shown in Table 2. The dependent variables were the overall heat transfer coefficient (U) and the amount of heat lost by the hot fluid (Q). For each response variable, a statistical analysis was carried out with the data calculated using the inlet and outlet temperatures during 20 min of heat exchange (the time when heat exchange stability between the fluids was achieved). The experimental design, mathematical model, statistical analysis and contour curves were obtained using the student-licensed software Protimiza Experimental Design. The results were evaluated with 95% reliability (α of 0.05). In this study, the cold flow rate was constant and equal to 1.4 L/min in order to simplify the planning.

Table 1. Coded and decoded levels used for the experimental matrix.

Independent variable	Units	Level		
		-1	0	1
Setpoint temperature (x_1)	°C	45	55	65
Hot fluid flow (x_2)	dm^3/s	0.6	1.0	1.4

Table 2. Experimental matrix for heat exchange with coded and decoded values.

Test	x_1	x_2	Setpoint temperature	Hot fluid flow
1	-1	-1	45	0.6
2	1	-1	65	0.6
3	-1	1	45	1.4
4	1	1	65	1.4
5	-1	-1	45	0.6
6	1	-1	65	0.6
7	-1	1	45	1.4
8	1	1	65	1.4
9	0	0	55	1.0
10	0	0	55	1.0
11	0	0	55	1.0

The response variables were determined using mathematical formulas, considering that the heat exchanged came exclusively from the hot fluid. The mean logarithmic temperature difference (MLTD), expressed in Kelvin (K); and the Q values, expressed in W (J/s), considering the specific heat at constant pressure (c_p) of the hot fluid to be 4.18 J/(g.K), were determined according to the literature. The heat exchange design equation was used to determine the U values, expressed in $W/(m^2.K)$, considering the heat exchange area (A) equal to 0.15 m² and the correction factor (Y) for the heat exchanger configuration equal to 1 (İnan et al., 2023; Kern, 1950; Kücük, 2023).

The heat exchange process was simulated using the free software Coco Simulator version 3.7. In order to compare the experimental values, predicted by the mathematical model of the experimental design, and the values obtained from the simulation, showing the effectiveness of the heat exchange. The heat exchanger configuration parameters were: no pressure drop, counter-current mode, hot water entering through the tubes, cold water entering through the shell, and the Water properties package (CAPE-OPEN 1.1) available in the software's default configuration. The heater was configured to heat the inlet water from approximately 23ºC to the setpoint temperature. The pumps were configured with no pressure increase and an adiabatic efficiency of 0.75 (software default).

Another way of comparing experimental data is with mathematical models that vary over time (transient regime) (Novazzi, 2007). To this end, a code language was developed in the open-access software Scilab 6.1.1. In developing the code language, the heat transfer equations, obtained from the energy balances of hot and cold fluids, were taken into account. To solve the mathematical model with partial derivatives, it was assumed that the temperature was uniform over the entire heat exchange area, there was no heat loss to the surroundings, constant inlet temperatures, no fouling and the arithmetic mean temperature difference was taken into account.

The method used to solve the partial derivatives was that of delayed finite differences (Pinto and Lage, 2001). In the method, the temperature was considered to be variable with the flow time and in the fluid flow direction (axial position). The physical properties were obtained as described above for the experimental matrix.

RESULTS AND DISCUSSION

Table 3 shows the results from the calculations involving Q and DTML to determine U, using the inlet and outlet temperatures of the water in the tubes and shell over the course of 20 min of operation of the heat exchanger. The table shows that the U values were higher than 100 $W/(m^2.K)$, which indicates that there was considerable heat transfer in the small heat exchange area. In addition, the DTML and Q values (which are not present in this work) were greater than 15 K and Q greater than 290 W (in module), respectively.

Table 3 shows that the highest U values were obtained in tests 3 and 8. For test e, the heat exchange was greater than 250 $W/(m^2.K)$ from 10 min of flow, with a peak of 285 $W/(m^2.K)$ in the first 10 min of the process. Tests 6 to 8 showed values similar to those of test 5 in the first 5 min of flow. Based on the above, it can be seen that the maximum heat exchange recorded occurs at the setpoint values: 45ºC and hot fluid flow rate equal to 1.4 L/min. It can therefore be seen that the flow time is fundamental for thermal exchange.

Table 4 shows the statistical results of the effect of the variables achieved for U with the experimental matrix for after 20 min of flow (system stability).The effect of the interaction of the independent variables was negative, indicating a drop in the heat exchange coefficient. On the other hand, the effect of the hot fluid flow rate was positive and greater than the other effects. In addition, the variables were significant (p-value<0.01). It is worth mentioning that the flow time altered the behavior of the system, although the statistical results are not present, Figure 2 clearly represents this observation.

Table 3. Results for U evaluated by experimental planning as a function of flow time.

Test	Global Heat Exchange Coefficient (W/(m^2.K))			
	5 min	10 min	15 min	20 min
1	115.58	117.44	105.17	105.17
2	161.04	140.40	145.55	142.51
3	244.58	285.12	257.81	252.03
4	253.41	216.29	222.21	225.19
5	109.22	116.54	121.45	105.17
6	158.82	148.32	145.68	141.30
7	266.37	208.45	259.43	251.21
8	255.46	199.82	222.58	225.95
9	187.91	178.64	191.50	186.54
10	185.53	158.33	188.80	186.54
11	182.46	166.21	187.67	186.54

Table 4. Results for the analysis of the effects of the independent variables (x_1, x_2 and $x_1.x_2$, referring to setpoint temperature, hot fluid flow and the interaction between both variables, respectively) on U after 20 min of flow.

Name	Coefficient	Standard error	t calculated	p-value
Mean	182.56	0.93	196.22	<0.01
x_1	2.67	1.09	2.45	0.04
x_2	57.53	1.09	52.73	<0.01
$x_1 \cdot x_2$	-15.70	1.09	-14.39	<0.01

Table 5 shows the results obtained after reparametrizing the model at a 5% significance level, with the most significant effects, with a considerably satisfactory coefficient of determination (R^2>80%)and equal to 99.77%. This indicates that the model fits most of the experimental data plotted, leading to good repeatability of the heat exchange experiments.

Table 6 shows the analysis of variance (ANOVA) for the response variable U. According to the Fischer-Snedecor test (F-Test), the F calculated from the regression ($F_{regression/residuals}$) was higher than the F tabulated from the regression (F_{tab}); and the F calculated from the lack of fit ($F_{lack\ of\ fit/pure\ error}$) was higher than the F tabulated from the lack of fit (F_{tab}); however, this occurs when there is little variation between the central points, as can be seen for U in Table 3. Therefore, based on the R^2 values and the F-test, it can be inferred that hypothesis H0 was rejected. Therefore, the variance of temperature and flow rate are statistically different, at 5% significance, for the response variable U.

Table 5. Mathematical models and coefficients of determination for U after 20 min of flow.

Mathematical model
$Y = 182.56 + 2.67\, x_1 + 57.53\, x_2 - 15.70\, x_1.x_2$

Table 6. Analysis of variance (ANOVA) for U, with the F-test for Regression/Residuals and Lack of Fit (FA)/Pure Error.

SV	SS	DF	MS	F_{calc}	F_{tab}	p-value
Reg	28505	3	9501.84	997.94	4.35	<0.01
Res	66.65	7	9.52	-	-	-
FA	65.30	1	65.30	290.77	5.99	<0.01
PE	1.35	6	0.22	-	-	-
Total	28572	10	-	-	-	-

Res – Residuals; Reg – Regression; PE – Pure Error; SV – Source of variation; SS – Sum of squares; DF – Degrees of freedom; MS – Mean square.

Figure 1 shows the response surface for U, in which it can be seen that at higher flow times, higher flow rates of the hot fluid are required to result in higher overall heat transfer coefficient values. In addition, the setpoint temperature has a considerable effect on U, but at lower levels higher heat exchange values are achieved.

Finally, the optimum operating point for the shell and tube heat exchanger, operated in countercurrent, cold fluid passing through the shell, hot fluid passing through the "U" tubes, with two passes in the shell, two passes in the tubes and for 20 min of flow, was 45ºC and 1.4 L/min. The U value in this region was close to 250 W/(m^2.K). Therefore, as the object of study considers the heat exchanged over the heat exchange area (U), the optimum operating point for this heat exchanger is at low setpoint temperatures, but with high hot fluid flow rates (or equivalent to cold fluid flow rates).

In order to compare the results achieved with the experimental matrix and those predicted by computer simulators, a simulation was carried out in COCO Simulator, considering a permanent regime (flow rate entering the system unchanged), that the mass of fluid leaving the exchanger was equal to the mass entering, without heat loss to the environment outside the hull and without the fouling term (Kapustenko et al., 2023).

The conditions simulated for U were: feed water (23.14 g/s; 23.1ºC), heater (45ºC) and cold-water feed (23.37 g/s; 23.1ºC). The output currents showed T_{Co} values of 30.11ºC and T_{Ho} values of 37.92ºC. The transient simulation conditions for U were: feed water (23.36 g/s; 45ºC), and cold-water feed (23.20 g/s; 23.1ºC). The output currents showed T_{Co} values of 32.45ºC and T_{Ho} values of 43.21ºC. The experimental values, after 20 min of flow and with heating at 45ºC, for the outlet temperature of the hull stream was 27.2ºC, and of the tubes was 38.4ºC.

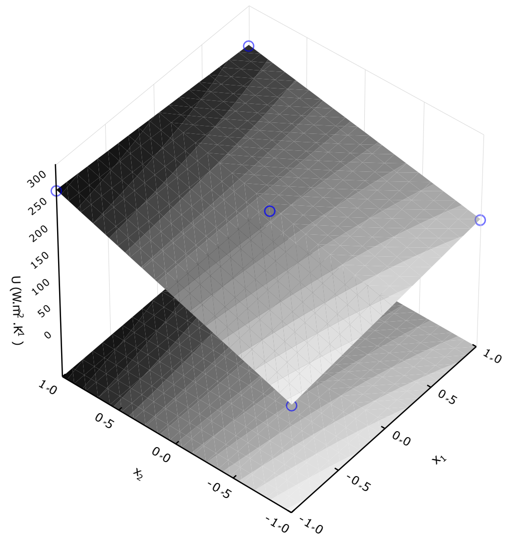

Figure 1. Contour curves for U in 20 min of flow. Where x_1 is the coded level of the setpoint temperature; and x_2 is the coded level of the hot fluid flow.

It can therefore be seen that there is a relative deviation of less than 10% between the temperatures observed for the simulated outlet currents in the permanent regime, but between 14 and 10% for the simulation in the transient regime. The value simulated in COCO Simulator in the optimum condition observed for U was 307.33 W/(m^2.K), 18% higher than the value predicted by the mathematical model at 20 min of flow. The value simulated in Scilab in the transient regime under the optimum condition observed for U was 415.95 W/(m^2.K), 66% higher than the value predicted by the mathematical model at 20 min of flow. It can therefore be seen that the simulation in the COCO Simulator software was relatively close to the value obtained with the experimental matrix, indicating the viability of computer simulations in practical applications and for educational use. Furthermore, although the transient simulation showed greater relative deviations for the outlet temperatures, they represent a way of checking the system's behavior over long periods of time, if you know the input parameters for iterative calculation and resolution of the theoretical mathematical models.

The code language used in the transient computer simulation was evaluated with the mathematical models, resolution methods and parameter determination via information available in the literature, as previously reported.

```
/*          HEAT EXCHANGER Finite difference method Model: Novazzi, 2007 */
/*     Delayed finite differences     */

//Note: METHOD SIGNAL INVERTED IN COLD TEMPERATURE
clc
clear
mode(-1);
lines(0);

function f=finites(t, x, nz, a1, a2, a3, a4, t0, deltat)
//     Applying the boundary conditions

    for j=1:length(t) //Variation over time
        for i=1 //Variation in space
         time=t0+j*deltat;
         x(i,j)=-6E-13*time^4 + 5E-09*time^3 - 2E-05*time^2 + 0.02*time + 328.24;//TH0 equation obtained by approximation using the graph
         f(i,j)=-a1/deltaz*(x(i+1,j)-x(i,j))-a2/2*(x(i,j)-x(nz+i+1,j)+x(i+1,j)-x(nz+i,j));
         f(nz+i,j)=a3/deltaz*(x(nz+i+1,j)-x(nz+i,j))+a4/2*(x(i,j)-x(nz+i+1,j)+x(i+1,j)-x(nz+i,j));
    end
        for i=2:length(z)-1//Variation in space
         f(i,j)=-a1/(2*deltaz)*(x(i+1,j)-x(i-1,j))-a2/2*(x(i,j)-x(nz+i+1,j)+x(i+1,j)-x(nz+i,j));
         f(nz+i,j)=a3/(2*deltaz)*(-x(nz+i-1,j)+x(nz+i+1,j))+a4/2*(x(i,j)-x(nz+i+1,j)+x(i+1,j)-x(nz+i,j));
    end
         for i=length(z) //Variation in space
        //x(nz+i,j)=TC0;
        time=t0+j*deltat;
        x(nz+i,j)=3E-14*time^4 - 5E-10*time^3 + 2E-06*time^2 - 0.0022*time + 294.93; //TC0 equation obtained by approximation using the graph
        f(i,j)=-a1/deltaz*(x(i,j)-x(i-1,j))-a2/2*(x(i,j)-x(nz+i-1,j)+x(i-1,j)-x(nz+i,j));
        f(nz+i,j)=a3/deltaz*(x(nz+i,j)-x(nz+i-1,j))+a4/2*(x(i,j)-x(nz+i-1,j)+x(i-2,j)-x(nz+i,j));
    end
end
endfunction

function specificheat=cp(T)
/* Calculating specific heat (Green and Perry, 2007, Table 2-153)
            [cp]= J/g.K              */
MM_water=18.015;
c1=276370;
c2=-2090.1;
c3=8.125;
c4=-0.014116;
c5=9.3701e-6;
specificheat=(c1+c2*T+c3*T^2+c4*T^3+c5*T^4)/(MM_water*10^3);
endfunction

function density=ro(T)
```

```
/* Density calculation (Green and Perry, 2007, Table 2-32)
        [ro]= g/dm3                */
MM_water=18.015;
c1=-13.851;
c2=0.64038;
c3=-0.00191;
c4=1.8211e-6;
density=(c1+c2*T+c3*T^2+c4*T^3)*MM_water;
endfunction

function coef=u(Thot_in, Thot_out, Tcold_in, Tcold_out, flow_q, area)
/* Calculating the overall heat exchange coefficient, U
        [U]=J/s.dm2.K              */
f_correction=1;
deltaT1=Thot_out-Tcold_in;
deltaT2=Thot_in-Tcold_out;
TMED=(Thot_in+Tcold_in)/2;
deltaTml=(deltaT1-deltaT2)/log(deltaT1/deltaT2)
q=cp(TMED)*ro(TMED)*flow_q*(Thot_in-Thot_out) //J/s
coef=q/(f_correction*area*deltaTml);
printf("Coefficient of thermal exchange: %f\n",coef);
printf("PROCESS DATA \n AVERAGE TEMPERATURE: %f; \n AVERAGE DENSITY: %f; \n cP AVERAGE: %f; \n OVERALL COEFFICIENT: %f \n", TMED,ro(TMED),cp(TMED),coef);
endfunction

/*      MAIN PROGRAM       */
// INTERVALS
  // TIME
  t0=0;
  tf=20*60;//24*60;//
  deltat=20;
  nt=(tf-t0)/deltat;
  t=[t0:deltat:tf];
  disp(length(t));
    // SPACE
  z0=0;
  zf=100;
  deltaz=10;
  nz=(zf-z0)/deltaz;
  z=[z0:deltaz:zf];

/*.......PARAMETERS.......*/
TH0=45.00+273.15; //K
THS=35+273.15; //K
TC0=23.10+273.15; //K
TCS=25+273.15; //K
printf('Final hot temperature (in the exchanger) = %f \n',THS);
printf('Final cold temperature (in the exchanger)= %f\n',TCS);
A=15/nz; //dm2
roh=ro(TH0); //g/dm3
roc=ro(TC0); //g/dm3
flow_h=1.4/60 //dm3/s
flow_c=1.4/60//dm3/s
mh=flow_h*roh;  //g/s
mc=flow_c*roc; //g/s
Vh=0.155; //dm3
Vc=5.114; //dm3
cph=cp(TH0); //J/g.K
cpc=cp(TC0); //J/g.K
vh=Vh/nz;
vc=Vc/nz;
U=u(TH0,THS,TC0,TCS,flow_h,15); //J/s.dm2.K
a1=(mh)/(roh*vh);
a2=(U*A)/(roh*Vh*cph);
a3=(mc)/(roc*vc);
a4=(U*A)/(roc*Vc*cpc);
/*........................*/
nz=length(z);
xinicial=[TH0*ones(nz,1);TC0*ones(nz,1)];
lista=list(finites,nz,a1,a2,a3,a4,t0,deltat)
x=ode(xinicial,0,t,lista);
disp(x);

/*..........PRINTING.........*/
colors=['r';'g';'b';'y';'k';'m';'c';'r-.';'g-.';'b-.';'y-.';'k-.';'m-.';'c-.';]
scf(1); clf();
for i=1:length(z)
   plot(t,x(i,:),colors(i));
end
xtitle('Hot temperature');
h1=legend(['z=0';'z=10';'z=20';'z=30';'z=40';'z=50';'z=60';'z=70';'z=80';'z=90';'z=100';],[1]);
scf(2);clf();
for i=1:length(z)
   plot(t,x(nz+i,:),colors(i));
end
xtitle('Cold temperature');
h2=legend(['z=0';'z=10';'z=20';'z=30';'z=40';'z=50';'z=60';'z=70';'z=80';'z=90';'z=100';],[4]);

disp("Start of the position vector")

//Vector with the desired points
   for i=1:length(t)
      vet(i,1)=x(1,i);
      vet(i,2)=x(4,i);
      vet(i,3)=x(8,i);
      vet(i,4)=x(11,i);
      vet(i,5)=x(22,i);
      vet(i,6)=x(20,i);
      vet(i,7)=x(14,i);
      vet(i,8)=x(12,i);
   end
disp("Hot sensor TI 04");
disp(vet(:,4));
disp("Cold sensor TI 08");
```

```
disp(vet(:,8));
printf(" HOT TEMPERATURE INPUT : %f; \n HOT TEMPERATURE OUTPUT: %f; \n COLD TEMPERATURE INPUT: %f; \n COLD TEMPERATURE OUTPUT: %f; \n HOT DENSITY: %f; \n DENSITY COLD: %f; \n HOT MASS FLOW: %f; \n COLD MASS FLOW: %f; \n cP HOT: %f; \n cP COLD: %f; \n",TH0,THS,TC0,TCS,roh,roc,mh,mc,cph,cpc);//K

disp('PROGRAM END')
```

The results obtained in this study are divided in two parts: (i) experimental validation of 3-D numerical results for finned arrangements, and (ii) global optimization results with respect to tube-to-tube spacing, eccentricity and fin density.

Heat flow in heat exchangers depends on the thermal conductivity of the surfaces in contact with the hot and cold fluids. For this, it is necessary that the heat exchange area is significant, maximizing contact with the fluids, enabling heat transfer (Hu et al., 2023). Heat transfer can be influenced by various factors and parameters, such as the turbulence of the fluids in the hull and the flow rate, providing higher transfer coefficients (Barewar et al., 2023). This was observed in this study, where the flow time and flow rate of the hot fluid were essential for increasing heat transfer.

The fundamental theory of thermal exchange states that there is a natural flow of heat from a hotter surface to a colder one, until the system reaches thermal equilibrium. This means that the colder bodies increase in temperature and the warmer ones decrease as the surfaces flow and come into contact. However, the warmer body does not transfer enough heat for its temperature to be lower than that of the other (İnan et al., 2023). This was observed in the tests carried out and in the computer simulation, highlighting the limitation of heat transfer.

In addition, heat transfer in heat exchangers is influenced by the area available for heat exchange, the specific heat of the fluid, the dimensionless correction factor, the flow rate, temperature, and others (İnan et al., 2023; Kücük, 2023). In this work, the specific heat of water was considered constant, but the density varied with the temperature of the hot fluid. Therefore, taking into account the number of variables involved in heat exchange, computer simulations can facilitate heat exchange experiments, reducing the number of experiments and maximizing heat exchange results (Khan et al., 2023).

Experiments involving heat exchangers can therefore be evaluated using computer simulations, based on a model reported in the literature. Computer simulations result in expressive results about the operation of equipment during an experimental or industrial routine. It is clear that the results of mathematical and computational models deviate from the experimental results, but they can provide conclusions and alternatives to overcome heat exchange problems (Alperen et al., 2023).

The modeling and computer simulation of shell-and-tube heat exchangers reported indicate that by increasing the mass flow entering the heat exchanger, the heat exchange coefficient can be increased (İnan et al., 2023). In addition, the existence of baffles associated with high flow rates can improve heat transfer, as turbulence is essential in heat exchange processes (Khan et al., 2023). This phenomenon was observed in this work, where the optimum heat transfer point occurred at high hot fluid flow rates.

Therefore, computer simulations associated with empirical mathematical modeling can provide valuable information for understanding the operation of equipment such as shell and tube heat exchangers. However, mathematical models are approximations that have errors associated with them, but they can be studied in order to understand the practical limitations of heat exchangers (Kapustenko et al., 2023). This can be circumvented with the use of statistical tools, such as contour curves, which provide optimal heat exchange work regions, making it possible to study the interactions of independent variables on heat exchange, minimizing the number of experiments (Alperen et al., 2023; Wang et al., 2023).

CONCLUSIONS

This study explored the relationship between the control variables of a shell and tube heat exchanger on the overall heat transfer coefficient as a function of flow time. It was found that optimization by experimental planning provided the optimum heat exchange point at high hot fluid flow rates (1.4 L/min) and low setpoint temperatures (45ºC), where the effects change with flow time, especially the heater temperature. In addition, the higher overall heat transfer coefficient value determined was around 250 W/(m^2.K). The experimental matrix therefore made it possible to investigate the effects of the variables on the U values as a function of time. When comparing the experimental outlet temperature values with the simulated values, there was a relative deviation of less than 10%, indicating that computer simulations in the steady state can provide results very close to the real thing, although the heat exchange effectiveness was less than 0.4. Therefore, this work provides a way of conducting heat exchange experiments, which makes it possible to better understand the phenomena involved and the influence of process variables, through experimental design and computer simulations.

ACKNOWLEDGEMENTS

The authors acknowledge the support from the Universidade Federal dos Vales do Jequitinhonha e Mucuri.

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