

A Didactic Test Rig to Analyze the Shell and Tube Heat Exchange and Stability of Control System

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Abstract. This work describes a didactic test rig both for simulation of a control system and for thermal exchange studies in a shell and tube type heat exchanger, enabling the simulation of several configuration of flow currents. Another aspect and purpose of this test rig is the possibility to study and analyze the dynamic behavior of the heat exchanger temperature control using a frequency converter as final control element instead of the conventional control valves for controlling the flow rate. Several temperature measurements in the inlets and outlets of the heat exchanger have been executed, allowing the obtainment of the overall heat transfer coefficients for concurrent and countercurrent flows. These results were compared with the theoretical ones that were calculated considering the heat exchanger dimensions, fluid and flows characteristics. Related to the control system, a temperature control optimization has been carried out by using the Ziegler-Nichols limit sensitivity of tuning method. Using the graphic resources of the supervisory software, the static and dynamic characteristics of the process were obtained enabling the elaboration of the system mathematical model. Than the simulation with the Symulink software were made, using the same tuning method. An analysis of the results have shown a good agreement between experiments and theory for both the thermal and the control viewpoint.

Key words: Heat exchanger, Temperature control and Frequency inverter.

1. Introduction

Heat exchangers are frequently used in countless continual process industries to heat, cool, vaporize or condensate fluids according to the processes needs. The continual supply of fluid at a specified temperature uses classical temperature control strategies, maintaining the variable controlled under a constant value in spite of load changes and disturbances.

To study heat exchanger thermal exchange efficiencies, the dynamic behavior of the variables involved, and temperature control, simulation may be used as a powerful tool in order to reduce project and implanting costs as well as to optimize production. Also in the educational and professional training fields, the benefits of simulation are notorious, where the student may apply and observe the theories and laws of thermodynamics and of heat transference, as well as the fundaments of automatic control and control tuning techniques for its optimization.

In articles published recently, Tyréus & Mahoney (2001), emphasize the importance of dynamic process simulations in several areas and their countless benefits. Cooper (2001), uses process control software to simulate and model processes and to analyze control tuning in closed networks. Bytronic International's electronic modules connected to pilot plants are used for training in the theory and practice of automatic control, simulating the typical flow, temperature and level processes, as well as load change and disturbance effects.

The designed test rig uses a heat exchanger as a tool for heating liquids with offers the possibilities both as concurrent and countercurrent outflow; the fluid outflow of the heater is by the side of the shell, or tubes with an automatic temperature control which are located in a closed circuit from where they can manipulate the hot water flow aiming at keeping the output temperature constant. A frequency inverter is being used as a final control element considering features like its configuration versatility, energy savings, time constant reductions and the system's load loss in itself when compared to the conventional control valves. Through simulation, the student can notice the approximation of the practical results to the theoretical ones, both in the heat transference area and in automatic process controls. Therefore this test rig is a relatively low in cost and proved to be a versatile tool to help the student to understand the types heat exchangers, the heat transfer principles and the principles and practice of automation processes.

2. Heat Exchangers

Shell and tube heat exchangers are commonly used in heat transference applications, mainly in continual process industries particularly where operation pressure and temperature are high. The thermal exchange efficiency is a function of countless constructive factors, such as the type of exchanger, number of passes, number of bafflers and also the properties of the fluids and the outflow conditions such as fluid speed and temperature.

2.1. Heat Transference in Shell and Tube Exchangers

The basic heat transference equation used in an exchanger design is given by Eq. (1) in which q (W) is a quantity of heat exchanged at a point of the exchanger with a thermal exchange surface A (m^2), ΔT (K) is the difference between the fluid temperatures and U (W/m^2.K) is the global transference coefficient that is related to internal and external tube wall resistances.

$$q = U \cdot A \cdot \Delta T \tag{1}$$

Equation (2) ignores the natural *incrustation factor* caused by the fluids on the internal and external heat exchange surfaces, reducing the value of U. The global coefficient, which includes incrustation resistance is called *project coefficient* U_D , (Kern, 1987), and it is not a constant. As in several cases U_D varies directly and non-linearly through the exchanger, it is necessary to consider U_D and ΔT with intermediary values that can be integrated numerically or graphically.

$$U = \frac{q}{\frac{1}{h_i} + \frac{1}{h_e}} \tag{2}$$

Where h_i and h_e are the heat transfer coefficient of the tube internal and external flow stream.

In several practical situations, it is possible to calculate the external thermal exchange area A_o, considering the average *global coefficient* U_{Dm} and the average temperature difference ΔT_m, as shown in Eq. (3).

$$A_o = \frac{q}{U_{dm} \cdot \Delta t_m} \tag{3}$$

The temperature difference between the two fluids varies from point to point, and the Eq. (4) provides the *Logarithmic Mean of Temperature Difference (LMTD)* for concurrent and countercurrent outflows, considering the exchanger operating in a stationary system and without fluid phase changes. The q heat transference rate may then be expressed by Eq. (5).

$$\Delta T_{med} = LMTD = \frac{\Delta T_{max} - \Delta T_{min}}{\ln\left(\frac{\Delta T_{max}}{\Delta T_{min}}\right)} \tag{4}$$

$$q = U.A.\Delta T_{med} \tag{5}$$

2.2. Typical Temperature Control Systems in Heat Exchangers.

The temperature control quality at the output of a heat exchanger will depend on the characteristics of the instruments that are used, on an adequate control strategy, and on the controller's tuning. Since the main objective of a heat exchanger is to keep the temperature constant in the continual feeding of a fluid, the main variables involved are both flow and temperature. The typical system portrayed in Fig. (1) shows the representation of a heat exchanger in crossed currents, where temperature control is performed by a simple closed loop control.

Considering a constant product flow (cold fluid), the temperature may be controlled by manipulating the flow of the hot fluid through the control valve opening. The temperature indicator controller (TIC*)* compares the measured temperature value with the set-point of the controller, it processes the error *e(t)*, executing the proportional integrative and derivative (PID) algorithm (Ogata, 1998), and adjusts the position of control valve *m(t)* maintaining heating product.

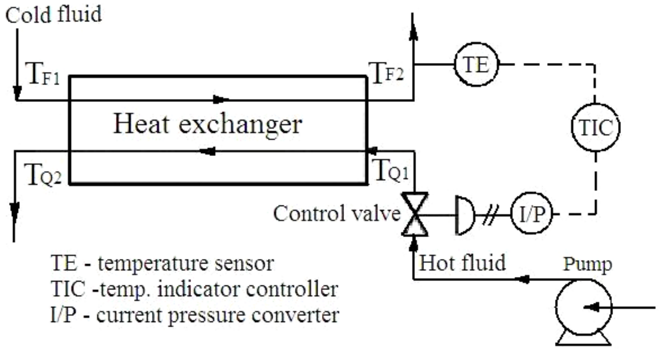


Figure 1. Typical heat exchanger temperature control with crossed currents.

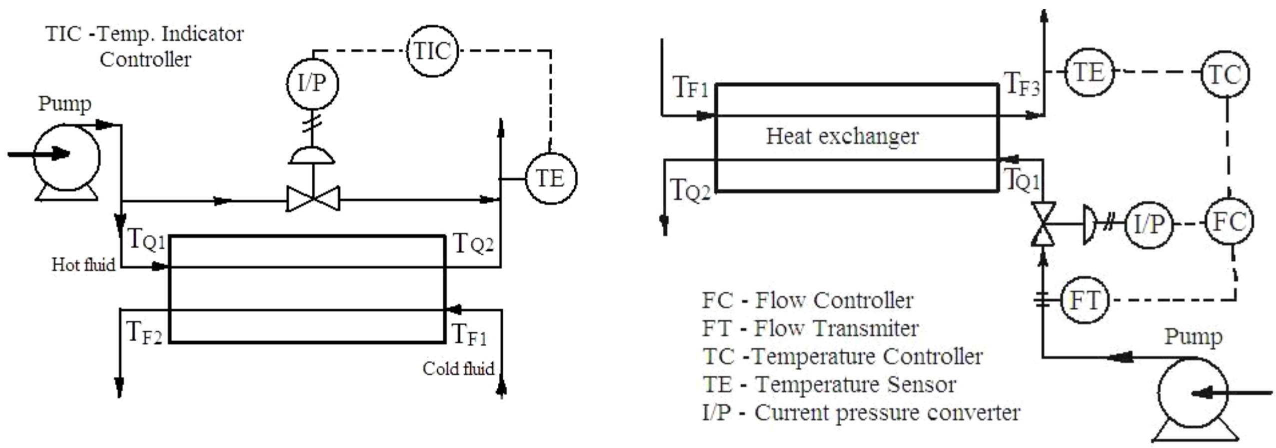
The Eq. (6) represents the Proportional-Integral-Derivative (PID) control mode in which K_p, K_i and T_d represent, respectively, the Proportional, Integral, and Derivative time adjustable constants. The correct adjustment of the PID parameters will be responsible for the control stability in the presence of disturbances and in changes of the requested temperature.

$$m(t) = K_p.e(t) + K_p.K_i\int_0^t e(t)dt + K_p.T_d.\frac{de(t)}{dt} \tag{6}$$

The relationship between the heated fluid flow and the cold fluid output temperature in this heat exchange process is not linear for low flow rates, while for high flow rates the effect on the transferred heat is very low. These characteristics cause a difficulty to control the process, making it inefficient under certain operation conditions.

In heat exchangers having high time constants, such as multiple pass heat exchangers, control speed and quantity are not satisfactory. In such cases, bypass control is used, as illustrated in Fig. (2).a, where control is made through the manipulation of the hot fluid flow bypass. This method eliminates delays by mixing hot fluids with the heat exchange output flow. However, the resistance to the heat exchange flow is placed parallelly to the control valve resistance, rendering control more difficult (Shinskey, 1979). Three-way control valves eliminate the problem when their costs are ignored.

In the example from Fig. (1) and Fig. (2). a, hot fluid flow disturbances may result in a temporary high transferred heat rate to the cold fluid, something that can cause a sudden temperature increase at the heat exchange output.



(a) (b)

Figure 2. a. Temperature control by parallel bypass and b. *Cascade* control.

Through the *cascade* control shown in Fig. (2).b, the disturbances caused by the hot fluid flow are detected before the output temperature of the heat exchanger (primary variable) is altered due to the secondary variable measurement (hot fluid flow), promoting fast performance (Buckley, 1979). The value of the secondary variable is sent to the "slave" temperature controller, which compares it to the remote set-point produced by the "master" flow controller and adjusts the control valve opening, maintaining the ideal hot fluid flow. Changes in the primary variable value or disturbances in the secondary variable are both, automatically and efficiently corrected through the *cascade control.*

In all the cases previously mentioned, temperature control is made evident by the manipulation of the flow through a control valve. The pressure drop is the mean problem caused by the control valve on the outflow.

The relationship between the fluid flow and the pressure drop caused by fluid acceleration while passing through the control valve is represented by Eq. (7) in which Q is volumetric flow in m^3/s, K_v is the constant due to the type of shutter and the number of Reynolds, A_o is the fluid passage area in m^2, Δp_v is the static pressure difference between the valve upstream and downstream in N/m^2 and ρ is the density of the fluid in kg/m^3.

$$Q = K_v \cdot A_o \cdot \sqrt{\frac{\Delta p_v}{\rho}} \tag{7}$$

Equation (8) gives the power in watts dissipated by the control valve due to pressure fall.

$$P_v = Q \,.\, \Delta p_v \tag{8}$$

Spitzer (1990) emphasizes the benefits of technologies related to speed variators to replace control valves for savings, due to the energy dissipated in the valve through the strangulation of the flow. Their lower costs when compared to control valves, besides the higher response speed, and the signal converter elimination, make viable the use a frequency inverter, as shown in Fig. (3). In this type of application, the exchanger output temperature is measured by a sensor that transmits an electronic signal to the controller which will, then, compare it to the adjusted set-point.

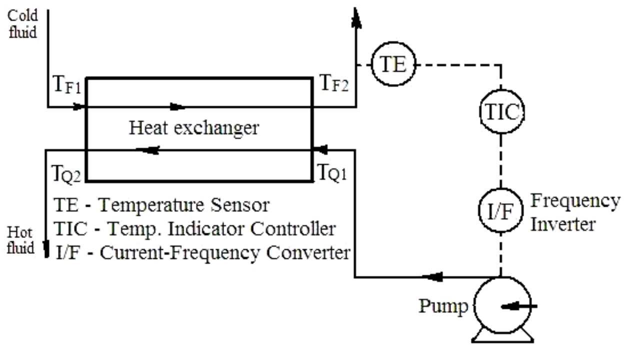


Figure 3. Heat exchanger temperature control using a frequency inverter.

The corrective effect produced by the controller is an analogical voltage signal that serves as a reference for the inverter, which will then produce a triphasic electric signal modulated in voltage and frequency causing variation in the rotational speed of the system feeding pump and, as consequence, the flow rate adjustment. Another advantage of this system is the facility to set parameters of the frequency inverter, which may be adjusted to several operation conditions with no need to replace the equipment.

3. Test Rig

The experimental part employs a test rig that uses a shell and tube heat exchanger with an exchange area of 0.1 m^2 and 3 transversal bafflers. It may be used with a 1-2 and 1-1 arrangement flow, plus and also the selection of the concurrent and countercurrent outflow via manual blocking valves, as illustrated in Fig. 4. The fluid used in the tests, on the shell side as well as in the tubes is water with the specific heat of 4185 J/kg.K and the density of 998 kg/m^3.

The hot fluid flow temperature on the tube side is obtained through an electric resistance kept under control by a digital controller whereas the flow is regulated by a frequency inverter, activating a 0.37 kW centrifuge pump. On the shell side, the flow is kept constant through a manual flow regulator valve and the heater temperature is kept at 70 ºC by a digital controller that activates a solid state relay connected to an electric resistance. The heat exchanger output temperature is controlled by the heater fluid flow at the exchange input through the analogical output of the second temperature controller that activates the circulation pump through the frequency inverter.

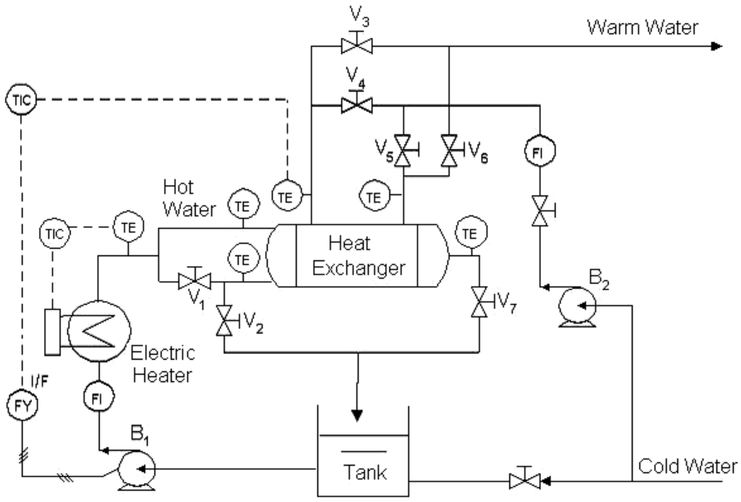


Figure 4. Schematic diagram of the test rig.

The fluid flow on the shell side and on the tube side is measured by differential pressure transmitters with an analogical output of 4 to 20 mA. The temperature and flow acquisition is performed by the input modules of a logical programmable controller that communicates with a supervisory system, as shown in Fig. (5). The process variable values are presented on screens, making the updated values available, either in numerical or graphical form, in order to make possible the configurations of the flow system.

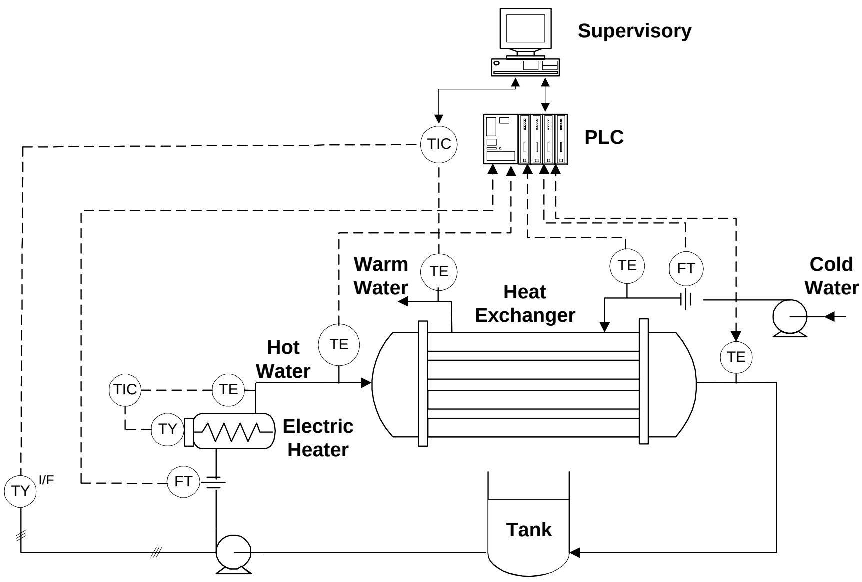


Figure 5. Temperature control and process variable acquisition.

The system's physical installation is shown in Fig. (6), which includes the heat exchanger, the hydraulic circuit, temperature sensors, flow measurers, frequency inverter, pump, heater, temperature controllers, programmable controller, and a personal computer with the supervisory software.

Figure 6. Instrument and equipment installation.

4. Results

The results presented are produced by simulations and tests, aiming at obtaining data to model the system and to analyze the results of the heat exchanger as well as that of the control system with the purpose of theoretical verification.

4.1. Thermal System

The several temperature measurements made at the heat exchanger inputs and outputs aim at obtaining the exchanged heat and the global heat transfer coefficient for the possible fluid currents in the system.

The following tests were made with hot fluid circulating through the tubes at four reference flows and with cold fluid through the shell at two reference flows, maintaining warm at 70°C the temperature of the fluid input, Whereas the cold fluid input temperature is maintained at room temperature (26ºC). The intended values and the actual practical values are presented in Tab. (1) and Tab. (2), and used to develop the calculations, having the flow of 700 l/h in the tubes and of 400 l/h in the shell as references.

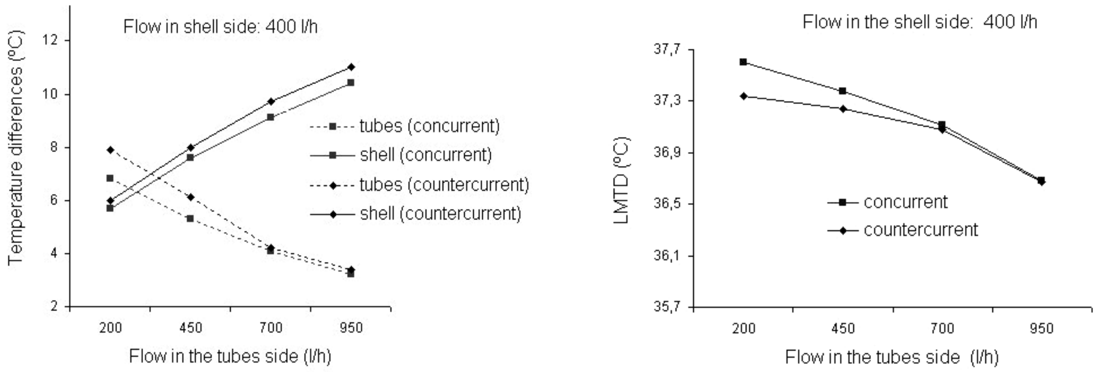
Table 1. Logarithmic mean of the temperature differences with shell flow maintained at 400 l/h for concurrent outflow.

Intended flow on the shell side, Q_f = 400 l/h (actual value = 402 l/h)										
Flow in the tubes Q_q (l/h)		Temperatures (ºC)						Logarithmic Mean of the Temperature Difference (ºC)		
		Tube side			Shell side					
Intended values	Actual values	T_{q1}	T_{q2}	ΔT_q	T_{f1}	T_{f2}	ΔT_f	ΔT_{max}	ΔT_{min}	$MLDT$
200	198	70,2	63,4	6,8	26	31,7	5,7	44,2	31,7	37,6
450	442	70,2	64,9	5,3	26	33,6	7,6	44,2	31,3	37,38
700	700	70,1	66	4,1	26	35,1	9,1	44,1	30,9	37,11
950	950	69,9	66,7	3,2	26	36,4	10,4	43,9	30,3	36,68
								Average LMTD = 37,19 ºC		

Table 2. Heat flow and global heat transfer coefficient with the flow in the shell kept at 300 l/h for concurrent outflow.

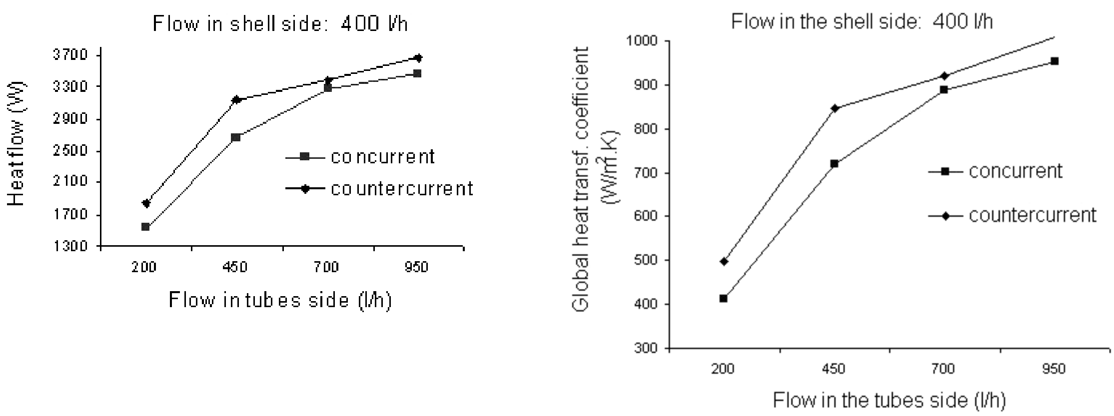
Heat flow					Global heat transference coefficient	
Q_q (l/h)	$\dot{m}_q$ (kg/s)	$\bar{c}_p$ (J/kg.K)	ΔT_q (ºC)	$\dot{q}_q$ (W)	*LMTD*	U_q (W/m^2.K)
198	0,05384	4189,6	6,8	1533,9	37,6	410,9
442	0,1202	4189,6	5,3	2669	37,38	719,1
700	0,19034	4189,6	4,1	3269,5	37,11	887,3
950	0,25832	4189,6	3,2	3463,2	36,57	953,8
Note.: Q_f = 400 l/h, $\overline{\rho}_q$ = 978,9 kg/m^3 e A_t = 0,09929 m^2					$\overline{U}_q$ = 742,8 W/m^2.K	

The graphics presented in Fig. (7). and Fig. (8) were generated through an electronic spreadsheet based on tabulated data and show the temperature variations, heat flow and the global heat transfer coefficient as a function of the input flows in the tubes and in the shell for concurrent and countercurrent outflows.



(a) (b)

Figure 7. (a) Temperature difference at heat exchanger terminal points as a function of the flow in tube side and (b) Logarithmic mean of the temperature difference



(a) (b)

Figure 8. (a) Heat flow and (b) Global heat transfer coefficient as a function of the flow in the tubes.

Having carried out all the tests scheduled , it was possible to compare theoretical and practical results and determine their points of approximations and deviations regarding the heat exchanger, as well as the effects produced by the work fluid flows on the temperature and flow configuration forms, taking into account, that the values presented were obtained considering internal and external tube walls clean, in order to determine the effect of the incrustation factor. For the global heat transfer coefficient, a strong incrustation influence representing heat losses of, approximately 20%, was observed in the calculations of both outflows.

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4.2. Control System

The static and dynamic characteristics were obtained by setting the process to operate in an open loop control system in order to define the heat exchanger and control system mathematical models. The system may be represented by the block diagrams in Fig. (9). In a closed loop system, the Ziegler-Nichols (Ogata, 1998) tuning approach was used as a test, observing the registration of the variables involved in the supervisory system's graphics screens.

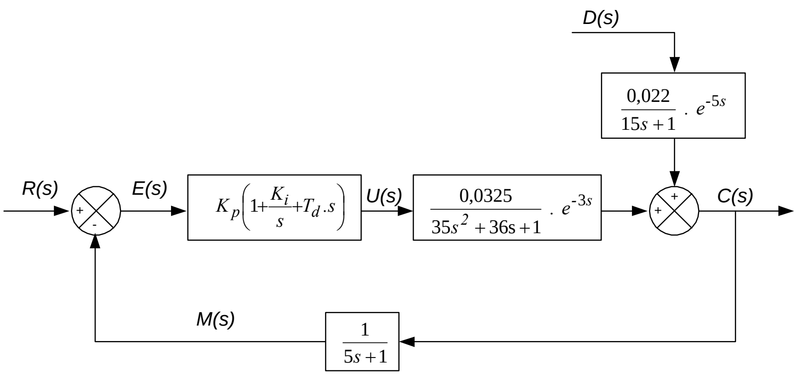

Figure 9. Control system block diagram.

By adjusting the proportional action (K_p) by 201, the integral action (K_i) in 0.077 repetitions per second, and the derivative action (T_d) in 3.25 seconds, for a step bypass of 2 ºC, an initial over-signal of 60% was obtained with a successive decline of the variable and accommodation time of approximately 140 seconds with the elimination of the off-set. After fine tuning a reduction to 90 seconds was observed in the accommodation time, as suggested by the method. Also in the over-signal value to 40% as shown in Fig. (10).

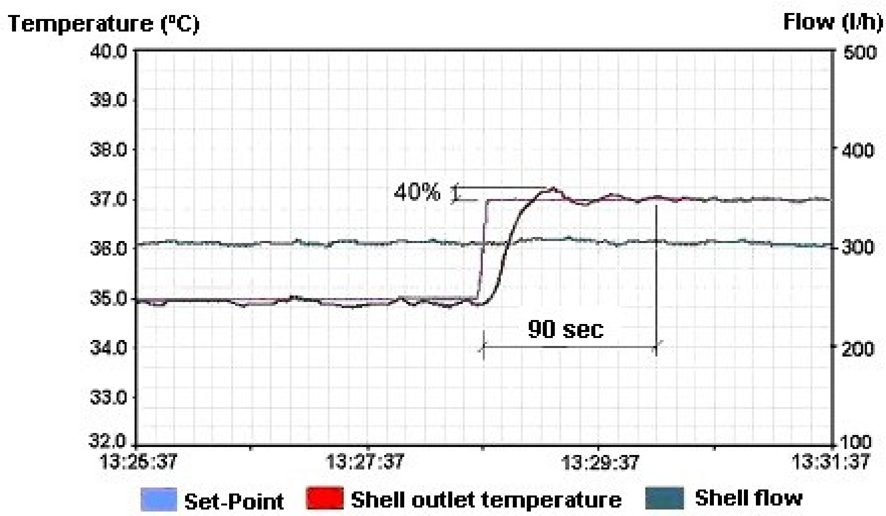

Figure 10. Controller tuning based on the Ziegler-Nichols method.

During the optimization process , having obtained the values of the *PID* parameters, it was possible to observe the control system’s response to a disturbance. During the simulation the set-point value was kept constant at 37 ºC and a flow disturbance was placed on the shell side at approximately 100 l/h, ascending and descending, obtaining as a response a deviation of 0.8 ºC and 1ºC, and a controlled variable stability with an approximate accommodation time of 80 seconds, as shown in Fig. (11).

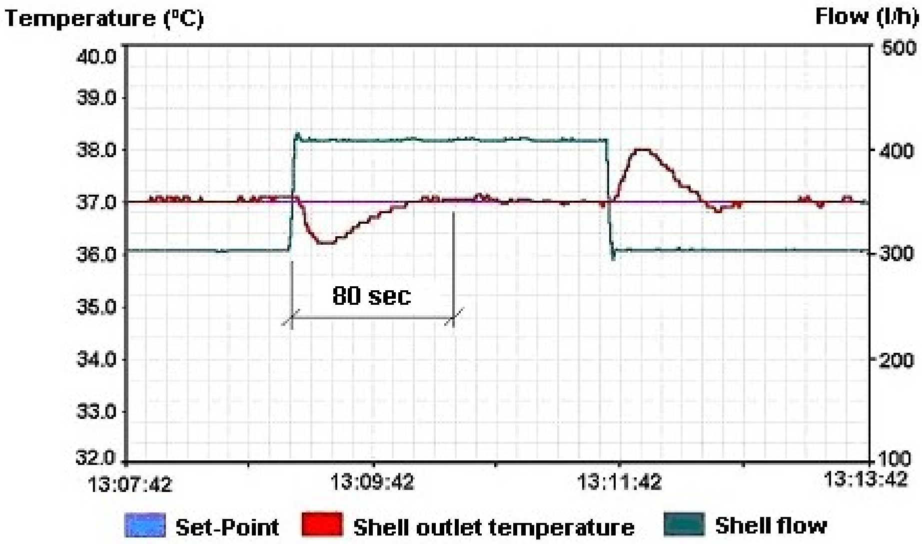

Figure 11. System response to a disturbance after control adjustment.

The results obtained by using the Ziegler-Nichols tuning method proved to be satisfactory, since the successive decline of the 25% variable was adopted as a stability criterion reached after fine tuning. If the variable’s oscillations were intolerable, a sufficiently quick buffered transitory response could be adopted as a stability criterion. Considering that the tabled values that were used are considered based on several assayed processes, an initial 60% over-signal was obtained, with an accommodation time of some 140 seconds, which is acceptable as an initial result.

Despite the auto-tuning resources of most of the industrial controllers, technical and practical knowledge regarding control action behavior becomes necessary for possible final adjustments to be made.

6. Conclusions

In the heat exchanger study, the global heat transfer coefficient and the influence of the nature of the fluid and flow currents on the temperatures were obtained successfully by the facilities offered by the system through the sensor system, the installed instruments, via the selection of a system employing manual valve circulation circuits, over and beyond through the supervisory system’s flexible resources, which allowed graphic monitoring, in real time, of the temperatures and flows, proving theoretical results.

Insofar as the control system is concerned, the use of the frequency inverter as the final control element proved to be efficient, producing a fast actuate response and contributing to control quality.

Other approaches may be used to analyze stability, such as the Bode and Nyquist diagrams in the frequency domain, or classical mathematic methods, such as state equations. The system allows the survey of each element’s transference function based on its static and dynamic characteristics and, with the assistance of a simulation software, one can compare the practical results to the theoretical ones.

The results obtained revealed the test rig adequacy for the theoretical consolidation of the heat transference and automatic control fundaments. Being able to handle a system that presents all the characteristics of the real process besides the degree of satisfaction reached by getting closer to this traditionally distinct areas elevates for academic teaching, in spite of the global cost of the equipment used as well as the cost of the powerful software tools that currently exist and are used by process simulation engineering.

7. References

Buckley, Page S., 1979, “Techniques of Process Control”, 2.ed. New York, Robert E. Krieger Publishing Co., 303 p.

Cooper, Doug. “Control Station”, Available in <http://www.controlstation.com/>. Access in may 5, 2001.

Kern, Donald Q., 1980, “Processos de Transmissão de Calor”, Translated by Adir M. Luiz, Rio de Janeiro, Editora Guanabara Koogan S.A., 671p.

Ogata, Katsuhiko, 1998, “Engenharia de Controle Moderno”, Translated by Bernardo Severo, 3.ed. Rio de Janeiro, Prentice-Hall do Brasil, 813p.

Shinskey, F. G., 1979, “Process Control Systems”, 2.ed. Mc Graw Hill Inc., 349p.

Spitzer, David William, 1990, “Variable Speed Drives: Principles and Applications for Energy Cost Savings”, 2.ed. North Carolina, Instrument Society of America,. 201p.

Tyréus, Bjorn D. and Mahoney, Donald P., “Applications of Dynamic Simulation”, Available in <http://www.hyprotech.com/support/papers/default.asp?menuchoice=su3>, Access in Jun 20, 2001.