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Antonio Augusto Torres Maia
Federal University of Minas Gerais

Marconi de Assis Silva
Federal University of Minas Gerais

Ricardo Nicolau Nassar Koury
Federal University of Minas Gerais

Luiz Machado
Federal University of Minas Gerais

Alexandre Carlos Eduardo
Federal University of Minas Gerais

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Control of an Electronic Expansion Valve Using an Adaptive PID Controller

Antônio A. T. Maia*, Marconi A. Silva, Ricardo N. N. Koury, Luiz Machado, Alexandre C. Eduardo

Federal University of Minas Gerais, Mechanical Engineering Department,
Belo Horizonte, Minas Gerais, Brazil

Phone: + (55 31) 3409 6667, Fax: + (55 31) 3443 3783, e-mail: aamaia@ufmg.br

ABSTRACT

In many refrigeration systems, electronic expansion devices have been used to replace the conventional expansion devices like capillary tubes and thermostatic expansion valves. The electronic expansion devices are usually provided with an automatic controller that is responsible for determining the valve opening that keeps the superheat at the outlet of the evaporator within the desired limits. Most of these controllers permit only the adjustment of the desired superheat, the proportional, the integral and the derivative gains. After being adjusted for one operating point, these parameters do not suffer any automatic correction even when the operating conditions changes. This could penalize the system efficiency because the controller parameters defined initially may not be the most suitable for the system when operating in this new condition. Within this context, in this work it was developed an adaptive PID-controller to regulate the superheat degree at the outlet of the evaporator. A dynamic model obtained from experimental tests was used in the controller design. The controller effectiveness was evaluated by means of computer simulation and through experimental tests. The results obtained showed that the employed technique is effective in regulating the superheat degree at the outlet of the evaporator with an acceptable performance.

1. INTRODUCTION

The increase in the energy prices in the last decades has motivated many research works to identify great energy consumers and ways to improve the efficiency of these systems. In this context, refrigerating machines have a representative participation in the daily energy consumption.

Most of the domestic refrigerating systems are equipped with a capillary tube or a thermostatic valve as expansion device. These expansion devices are not able to deal with wide range of operation conditions and they also present some response lag. Bearing in mind the same installation, energy savings can be obtained by replacing the conventional expansion device by an electronic expansion valve (EEV). The employment of this valve can be advantageous when compared with the conventional expansion devices because it has shorter response time and the controller used in most of these systems is generally able to keep the superheat close to the optimal value under every condition, which contributes to improve the refrigerating capacity (Lazzarin & Noro, 2008; Fallahsohi, 2010). The PID controller that comes with the electronic expansion valve normally permits adjustments in the proportional, integral and differential gains. If it is not properly adjusted the system can display a less efficient response or even an unstable response. On the other hand, when the controller gains are correctly defined it is generally done for one operating point and these settings do not suffer any automatic change even when the machine is working in a different operation point, which also may reduce the system efficiency. A strategy that could be used to overcome these problems is to use a controller with an auto tuning algorithm. To improve its efficiency, this algorithm should be executed continuously and the controller gains updated at every change in the operating point.

Several control methods are available for controlling the superheat at the outlet of the evaporator using an electronic expansion valve. Outtagarts et al. (1997) proposed two control algorithms (PD and Qualitative Optimal Regulation), based on the system characteristics obtained from experimental data, to control an EEV. Ekren and Küçüka (2009) proposed a fuzzy logic control to regulate the speed of a scroll compressor and to adjust the opening of an EEV. In both works it was emphasized the importance of an effective controller.

In this work it was developed an adaptive PID-controller to regulate superheat degree at the outlet of the evaporator. The automatic robust tune rule proposed by Vilanova (2008) was employed calculate the controller gains. Mathematical relations extracted from experimental data allowed to calculate the system gain and time constant for each operating point and this information was used in the tune algorithm. The whole control algorithm was tested by means of computer simulations and experimental tests.

2. EXPERIMENTAL APPARATUS

The experimental test bench (Figure 1) consists of a vapor compression refrigerating system, which has R134a as refrigeration fluid and pure water as secondary fluid. The system is basically composed by a reciprocating compressor, a condenser, a sub-cooler, an evaporator, three expansion valves and systems to do measurements and data acquisition. The compressor is alternative type and it has a piston displacement of 157 cm³. A three-phase electrical motor is employed to drive the compressor. This electrical motor is powered by a frequency inverter that enables the variation of the revolution speed of the motor-compressor assemblage. The condenser is shell and tube type and it has a 6 kW capacity. The secondary fluid temperature in the condenser is adjusted by mixing warm water that comes from the condenser itself with room temperature water, coming from the feeding system. The sub-cooler is coaxial type, made of an envelope tube and of an internal tube in "U". The evaporator is coaxial type and it is composed by a PVC envelope tube and three inner cooper tubes through which flows the refrigeration fluid. Water flows in counter flow in the space between the PVC and cooper tubes. The evaporator was designed to provide a maximum refrigeration capacity of 3 kW. In the evaporator, the secondary fluid temperature is maintained within the desired limits by an electrical heating system. The experimental bench has three expansion valves placed in parallel (manual, thermostatic and electronic type). A blockage valve permits the isolated operation of each expansion device. In this work, only a manual expansion valve was used. Eleven T-type thermocouples were implanted inside the tubes at the inlet and outlet of each system component. Two piezoresistive pressure sensors were installed at the inlet and outlet of the expansion devices. The refrigerant mass flow was measured with a Coriolis flow meter. The signals generated by the different sensors of the test bench are received and treated by a data acquisition system.

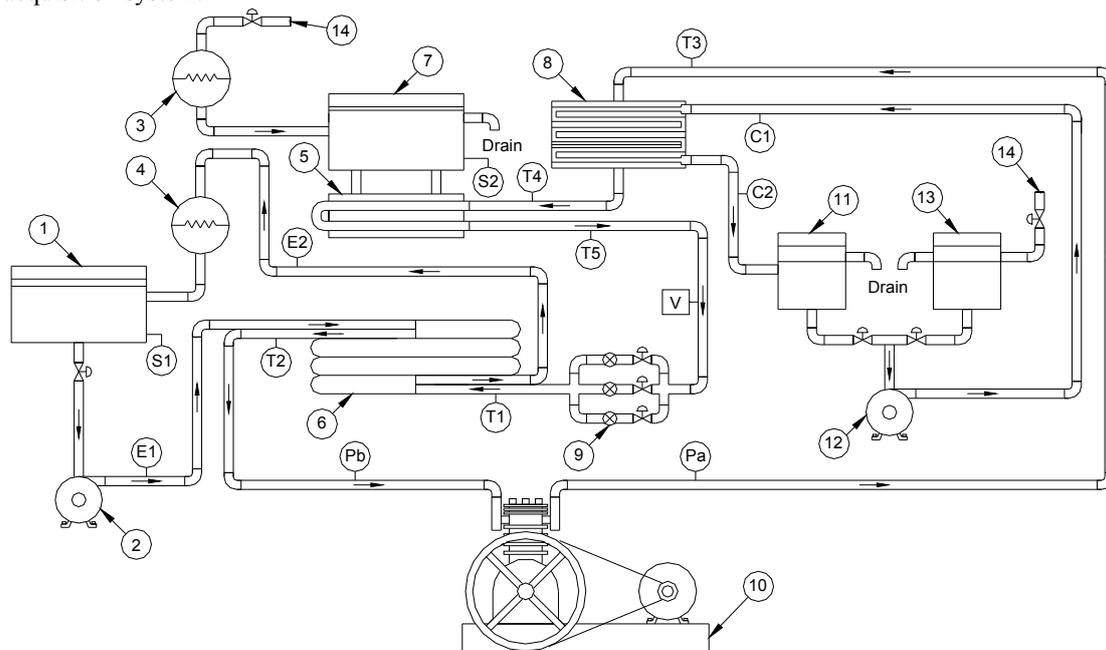


Figure 1: Experimental bench schematic configuration.

Legend:

- | | | | |
|-------|--|---------|--|
| 1. | Water reservoir (evaporator) | 12. | Water pump (condenser) |
| 2. | Water pump (evaporator) | 13. | Cold water reservoir (condenser) |
| 3, 4. | Electrical heater (sub-cooling / evaporator) | 14. | Water feed |
| 5. | Cooler | V. | Flow meter |
| 6. | Evaporator | Pa. | Pressure sensor (condensation) |
| 7. | Water reservoir (cooler) | Pb. | Pressure sensor (evaporation) |
| 8. | Condenser | S1, S2. | Temperature sensor (PID) |
| 9. | Expansion devices | T1-T5. | Temperature sensor (refrigeration fluid) |
| 10. | Reciprocating compressor | E1, E2. | Temperature sensor (evaporator water) |
| 11. | Hot water reservoir (condenser) | C1, C2. | Temperature sensor (condenser water) |

3. MODEL IDENTIFICATION

In order to obtain the experimental data needed to identify the system model, several tests were performed, to evaporating temperatures of -5°C , 0°C , 5°C and 10°C . The experimental tests consisted in to adjust the refrigerating machine to operate with a superheat at the outlet of the evaporator close to 7°C . This adjustment was obtained controlling by hand the opening of the manual expansion valve. Once in steady state, the mass flow rate at the inlet of the evaporator was reduced by closing the expansion valve. During these tests, the condensing temperature was kept close to 50°C and the compressor speed in 650rpm. In Figure (2) is presented the superheat (ΔT_s) response due to the mass flow rate (\dot{m}_f) reduction at the inlet of the evaporator for an evaporating temperature of -5°C . The superheat corresponds to the difference between the temperature at the outlet and the inlet (evaporation temperature) of the evaporator ($\Delta T_s = T_2 - T_1$).

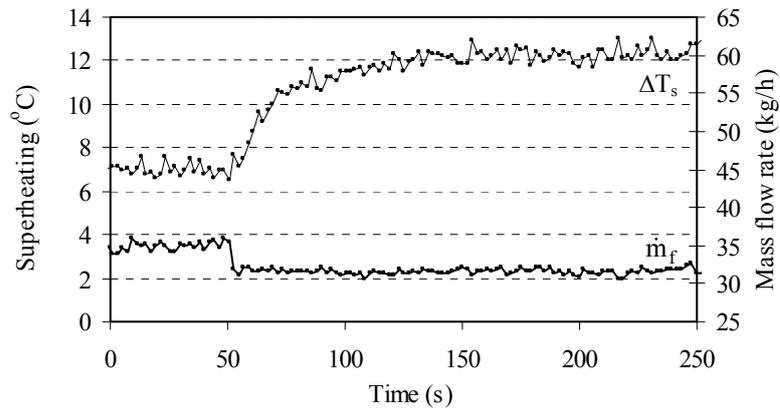


Figure 2: Superheat response due to the mass flow rate reduction at the inlet of the evaporator for an evaporating temperature of -5°C .

The superheat response can be represented by a first order function plus time delay (Outtagarts, 1997; Maia, 2005):

$$G(s) = \frac{K e^{-\theta s}}{\tau s + 1} \quad (1)$$

Where K represents the static gain in Kelvin.h/kg, τ the time constant in seconds and s the Laplace operator. The static gain and time constant were estimated for each operating point regarding the evaporation temperature, and they are being presented in Figure (3). The static gain was estimated using the relation $\Delta(\Delta T_s) / \Delta \dot{m}_f$ and the time constant correspond to the time instant in which the superheat reached 63% of its final variation. The data presented in Figure (3) was employed to define mathematical relations that describe the gain and time constant behavior according to evaporation temperature.

$$K = 0.0027 T_1^2 + 0.0321 T_1 - 1.7163 \quad (2)$$

$$\tau = -0.0399 T_1^2 - 0.3338 T_1 + 20.254 \quad (3)$$

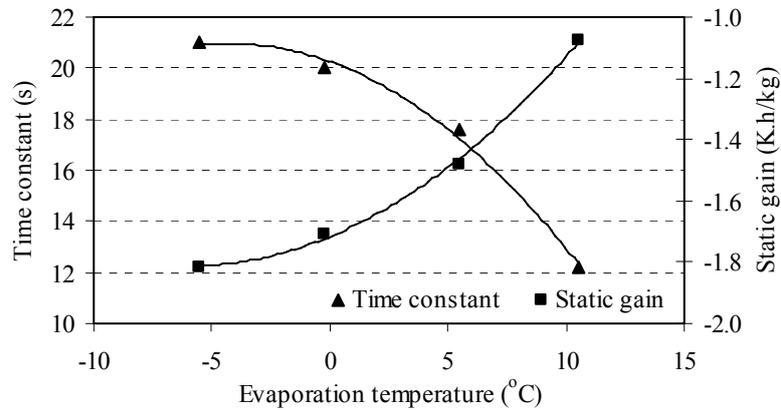


Figure 3: Time constant and static gain evolution as regards evaporation temperature

4. CONTROLLER DESIGN

There are several methods of tuning a PID controller. In the study performed by Cominos and Munro (2002) it was presented a summary of the most used PID tuning methods and some of the more recent techniques. Many of these strategies define the controller gains for one operating point. In view of the fact that the evaporator dynamic behavior is influenced by operating conditions, a simple PID controller operating with static settings can fail in providing a system response with the desired performance specifications. To overcome these problems it was employed a standard PID controller with the automatic robust tune rule proposed by Vilanova (2008). This method is based upon a first order plus time delay model and aim to achieve a step response specification while tanking in to account robustness considerations. This strategy allows calculating and updating the controller gains at each operating point, providing a completely automatic tuning determined by the process parameters. The control tuning rule can be summarized by the following set of equations:

$$T_i = \tau + 0.03 \theta \quad (4)$$

$$K_p = \frac{T_i}{2.65 K K_v \theta} \quad (5)$$

$$N + 1 = \frac{\tau}{T_i} \quad (6)$$

$$\frac{T_d}{N} = 1.72 \theta \quad (7)$$

Where K_p is the proportional gain, T_i and T_d are the integral and derivative time constants, K_v is the valve gain and N represents the ratio between T_d and time constant of an additional pole introduced to assure the properness of the controller. For this parameter it is usual to assume $N \approx 10$, but it was considered as being a design parameter, as can be seen in the Equation (6) (Vilanova, 2008). It is important to emphasize that almost all parameters needed to solve Equation (4) to Equation (7) are present in Equation (1). These equations working together with Equation (2) and Equation (3) can provide satisfactory control for a different operating point of the system. One observable variable, evaporating temperature in this case, is used to determine what operating point the system is currently in and to enable the suitable controller. The controller obtained using this strategy is called adaptive controller with gain scheduling (Åström and Wittenmark, 1995) and it is being illustrated in Figure (4). The system has two feedback loops. The first is the conventional feedback loop. The second is the adaptation feedback loop used to adjust the controller gains according to the current operating conditions.

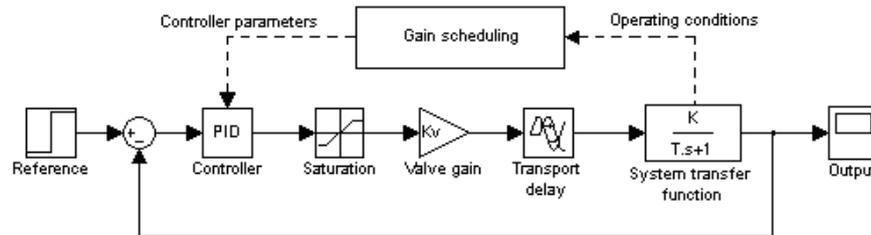


Figure 4: Schematic representation of an adaptive controller with gain scheduling

The controller performance using the presented tuning method was evaluated by means of simulations using Matlab-Simulink. The expansion valve gain was obtained from experimental tests by linearizing the expression that relates valve opening with mass flow rate. This variable corresponds to 1.7 when working with an evaporation temperature of 10°C, corresponding to the operating point in which the system provides the maximum refrigerating capacity. The delay time was considered to be 2s in all simulations. To evaluate the improvements obtained with the adaptive control, the controller gains were defined using the tuning rule presented in this work for an evaporating temperature of 10°C. Keeping these gains static, the evaporating temperature was changed from 10°C to -5°C in steps of 5°C. The results obtained in these simulations are being presented in Figure (5).

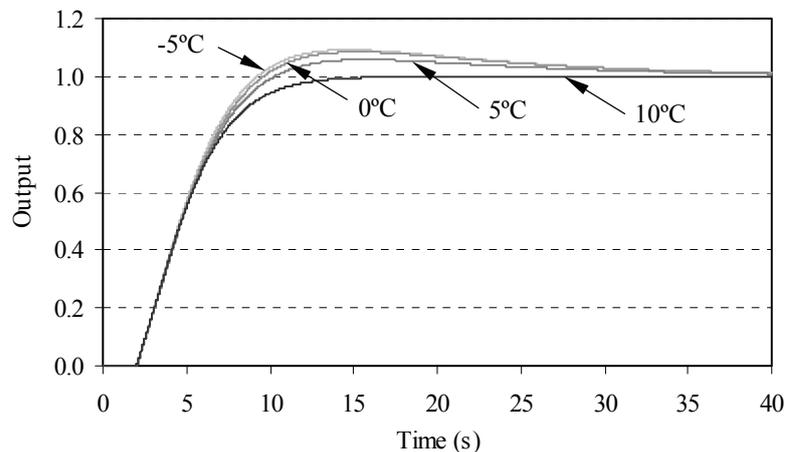


Figure 5: System responses to a step input for evaporating temperature varying from 10°C to -5°C in steps of 5°C and controller tuned for evaporating temperature of 10°C.

In spite of being acceptable, for evaporating temperature different from 10°C the results obtained are not as good as the results obtained for 10°C. They present overshoot and longer settling time. Considering that refrigerating systems use to work continuously, these small differences can become significant after long operation time.

5. CONTROLLER IMPLEMENTATION

In order to put into practice the proposed methodology, the control algorithm was programmed in a microcontroller. To meet all the requirements of this project it was necessary a microcontroller with two analog inputs to read the signals from the temperature sensors (LM335), one timer to generate the time base for the numerical integrals and derivatives and seven digital I/O to be employed to control a LCD display (1602B). One extra digital I/O will also be necessary to control a servo motor (Futaba S3305) that will be used as the valve actuator. Considering these project demands, it was chosen the PIC16F877A.

5.1. Controller algorithm

There are several ways to express the PID control law (National, 2001; Cominos and Munro, 2002; Ogata, 2003; Vilanova, 2008). In this work was considered the typical PID control law.

$$u(t) = K_p \left(e(t) + \frac{1}{T_i} \int_0^t e(t) dt + T_d \frac{de(t)}{dt} \right) \quad (8)$$

$$e(t) = SP - PV \quad (9)$$

Where $e(t)$ represents the error, SP is the set point and PV is the process variable, $u(t)$ is the controller output. To solve the controller equation using the microcontroller it was necessary make use of numerical methods to solve the integral and the derivative part. The integral part was solved using trapezoidal integration and derivative part was approximated by a finite difference equation. Since the set point can change instantly in a step change, the error will also have a step change whenever the set point is changed. This sudden change in the error makes the derivative of $e(t)$ to be infinite (derivative kick). In order to avoid a sharp spike on the control signal at the time of a reference change, the derivative action was based in the in the process variable rather than the error. In this way, the derivative part (u_d) presented in Equation (8) can be written as:

$$u_d(t) = K_p \left(T_d \frac{d(SP(t) - PV(t))}{dt} \right) = K_p \left(-T_d \frac{d(PV(t))}{dt} \right) \quad (10)$$

Considering all these factors, Equation (8) can be written as:

$$u(t) = K_p \left\{ e(k) + \frac{1}{T_i} \sum_{i=1}^k \left[\frac{e(i) - e(i-1)}{2} \right] \Delta t - \frac{T_d}{\Delta t} [PV(k) - PV(k-1)] \right\} \quad (11)$$

The proportional term is also based on error and it will also respond strongly to a step change in $e(t)$. To eliminate a possible sudden spike in the control signal the proportional term can also be based on the process variable instead of the error. Nevertheless, as the response of the proportional action is much less severe than the derivative action, the proportional action will be kept as it is presented in Equation (11).

6. EXPERIMENTAL RESULTS

The experimental tests to evaluate the controller performance consisted in programming the superheat set point to 10°C. After about 400s, the set point was changed to 7°C. These tests were performed considering the evaporating temperature of 10°C, 3°C e -3°C and a delay time of 0.5s in all tests. For all operating conditions, the superheat at the evaporator outlet is equal to the set value. In Figure (6) is presented the superheat response when the set point is changed form 10°C to 7°C, for an evaporating temperature of -3°C.

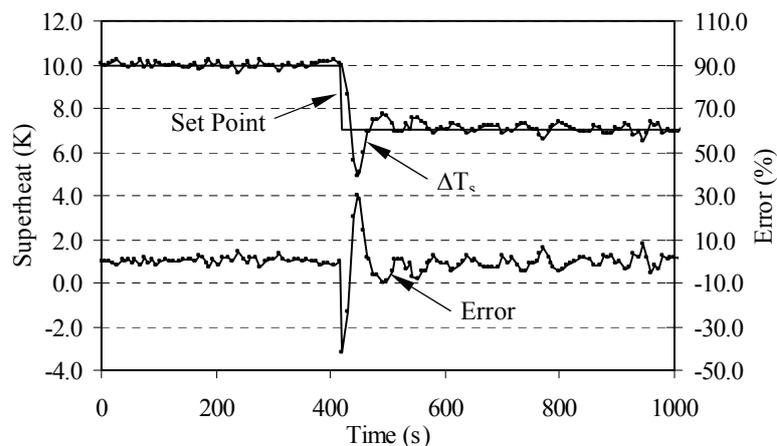


Figure 6: Superheat response to a step change in the superheat set point from 10°C to 7°C, for an evaporating temperature of -3°C.

The control algorithm succeeds in keeping the superheat in 10°C with a maximum percentage error of about 4%. After the step, the superheat percentage error becomes less than 8% in approximately 120s. This difference was not observed when working with higher evaporating temperature. The result observed shows that the conditions in which this test was performed are very close to the conditions for the minimum stable superheating. Another test was performed starting from a superheat of 10°C and then changing the set point to 5°C. The results obtained are presented in Figure (7).

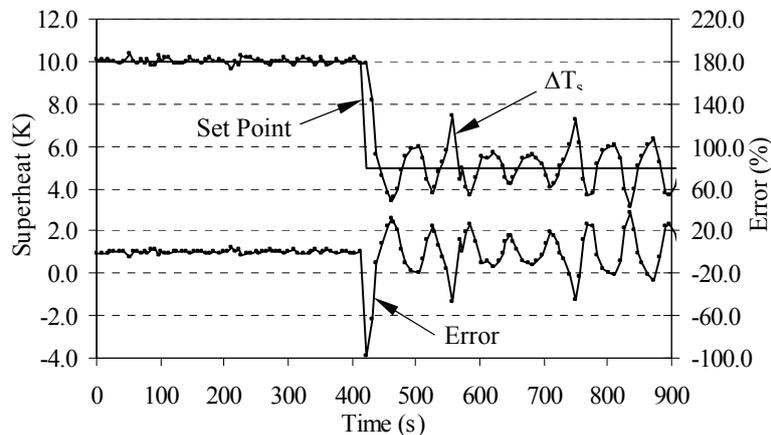


Figure 7: Superheat response to a step change in the superheat set point from 10°C to 5°C, for an evaporating temperature of -3°C.

Initially the controller maintained the superheat equal to the set value (10°C). When the set point was reduced to 5°C, the superheat value started to oscillate. The behavior observed after the step disturbance shows that the set point of 5°C is less than the minimum stable superheat, and the controller was not able to perform an accurate control. It has been shown by Chen et al. (2002) that the minimum stable superheat is influenced by the change of the heat transfer mechanism when the temperature gradient is very low and it has little relationship with the expansion valve type and the control method. In the work presented by Fallahsohi et al. (2010), some tests were performed to investigate the minimum stable superheat. The controller proposed (predictive functional control) also failed in providing an acceptable response when operating with a superheat smaller than the minimum stable superheat. In these tests it was observed that the temperature oscillations occurred for setting value equal to 4.8°C, that is close to the set point value employed in the test presented in Figure (7).

The last test to be presented consisted in a comparison between the adaptive and non adaptive controller. The controller gains were defined using the tuning rule presented in this work for an evaporating temperature of 10°C. Keeping these gains static, the evaporating temperature was changed to -3°C. The results obtained in this test are presented in Figure (8).

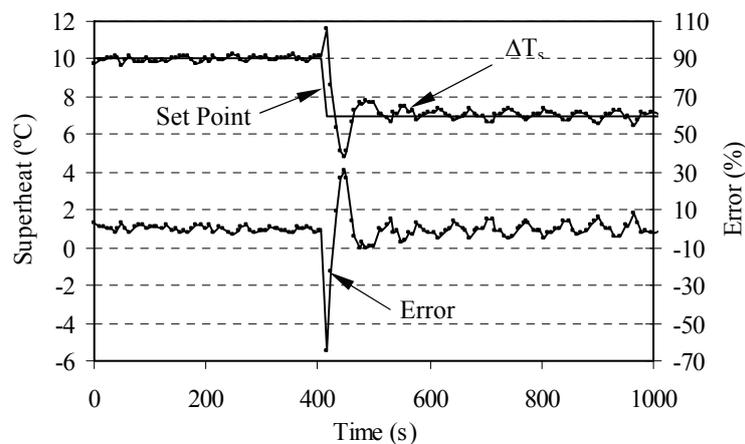


Figure 8: Superheat response to a step change in the superheat set point from 10°C to 5°C, for an evaporating temperature of -3°C using a non adaptive controller.

The results presented in Figure (8) using a non adaptive controller are similar to the results obtained with the adaptive controller (Figure 6). Before the disturbance, the sum of the absolute errors for the non adaptive controller was 5.9 and for the adaptive controller was 5.4. After the disturbance, it was considered that the system achieved the steady state when the error became inferior to 10% at the first time. The sum of the absolute errors for the non adaptive controller was 14.7 and for the adaptive controller was 13.5. During the initial seconds after the disturbance, firstly the superheat rises to about 11.5°C and then it decreases to about 5°C. This initial augment contributed to increase the percentage error at this point. Despite the fact that in both situations the controller was adjusted using the same algorithm, the adaptive controller worked better.

7. CONCLUSIONS

Most of PID controllers employed in electronic expansion devices do not suffer any automatic adjustment when the refrigerating system suffers a change on its operating point. In this context, in this work it was developed an adaptive PID controller using the control tuning rule proposed by Vilanova (2008). Experimental data were utilized in order to identify a mathematical model and to determine a mathematical relation for the static gain and time constant. These mathematical relations working together with the control rule allowed calculating the controller gains for each operating point. The controller was programmed in a microcontroller and evaluated by means of experimental tests. The results showed that the proposed controller is effective in controlling the superheat with good performance. The adaptive controller was compared with a non adaptive controller adjusted with the same control rule considering a different operating point. The results showed that the adaptive controller provided a good response with an inferior percentage error.

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