DEVELOPMENT OF AN ADAPTIVE PID CONTROLLER FOR AN ELECTRONIC EXPANSION VALVE

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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 which is responsible for determining the valve opening that keeps the superheating at the outlet of the evaporator within the desired limits. Most of these controllers permit only the adjustment of the desired superheating, the proportional, integral and derivative gains. These adjustments do not suffer any automatic correction, even when the equipment works in a different operation point, what may reduce the control system efficiency due to an increase in the settling time and/or in the overshoot. In this work it was developed an adaptive PID-controller to regulate superheating degree at the outlet of the evaporator. A dynamic model obtained from experimental tests was used in the controller design. The controller efectiveness was evaluated by means of computer simulations. The results obtainned indicated that the employed technique is effective in regulating the superheating degree at the outlet of the evaporator with an acceptable performance. At the end is presented the strategy that will be used to implement the proposed controller in a PIC microcontroller.*
Keywords: *Adaptive control, PID controller, Refrigerating system, Electronic expansion valve.*

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. Additionally, these devices are produced in large numbers meaning that any improvement achieved in these units can lead to significant energy savings.

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, considerable energy savings can be obtained by replacing the conventional expansion device by an electronic expansion valve. 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 superheating close to the optimal value under every condition, witch contributes to improve the refrigerating capacity (Maia, 2005; Lazzarin & Noro, 2008).

The PID controller that came with the electronic expansion valve normally permits to do 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, what 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.

In this work it was developed an adaptive PID-controller to regulate superheating 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 form 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. In the final part of this work is presented some aspects that will be considered when implementing the proposed controller.

2. EXPERIMENTAL APPARATUS

The experimental device (Fig. 1) consists of a vapor compression refrigerating system, which has R134a as refrigeration fluid and, as secondary fluid, pure water in the evaporator and in the condenser. 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 cm^3. 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 a multiple tube 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.

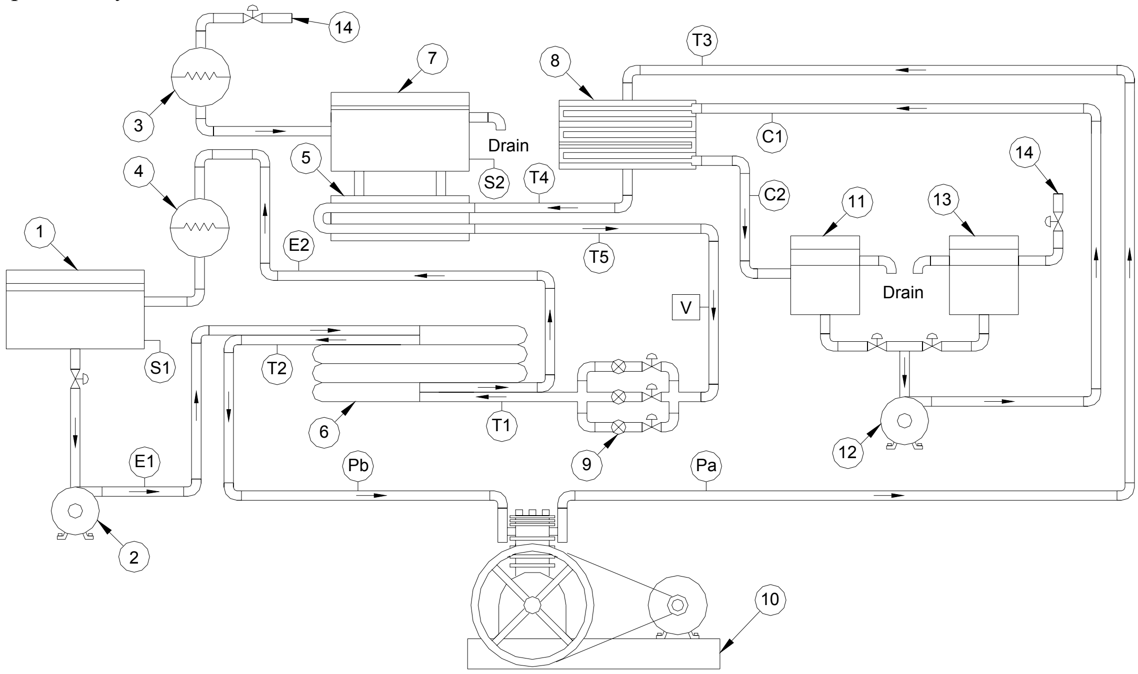

Figure 1 – Experimental bench schematic configuration.

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

It was performed several experimental testes to obtain the experimental data needed in the process of model identification. The testes carried out covered the evaporating temperature of -5^{o}C, 0^{o}C, 5^{o}C and 10^{o}C. During these testes the condensing temperature was kept close to 50^{o}C and the compressor speed in 650rpm. The experimental tests consisted in firstly to adjust the refrigerating machine to operate in steady state, presenting a superheating at the outlet of the evaporator close to 7^{o}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 in about 5% by closing the expansion valve. The system dynamic response was stored in a data file until a new steady state was reached. In Fig. 2 is presented the superheating (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 superheating corresponds to the difference between the temperature at the outlet and the inlet (evaporation temperature) of the evaporator ($\Delta T_s = T_2 - T_1$).

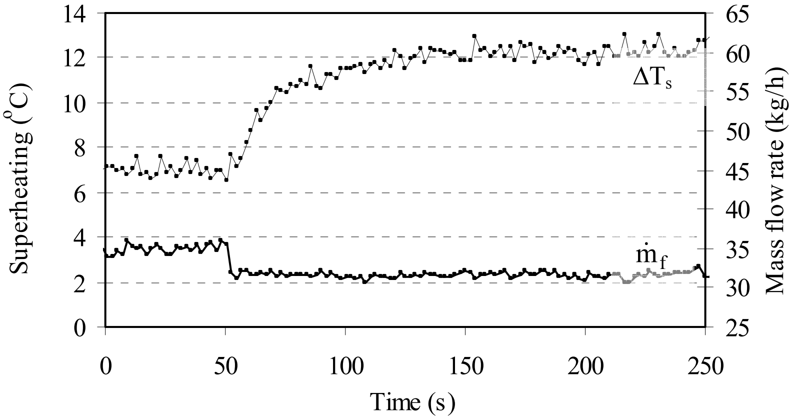

Figure 2. Superheating response due to the mass flow rate reduction at the inlet of the evaporator for an evaporating temperature of -5°C.

The superheating response can be represented by a first order function with time delay (Eq.(1)) (Outtagarts, 1994; Maia, 2005). The time delay values observed in the experimental data were 6s, 4s, 4s and 2s which corresponds to the evaporating temperature of -5°C, 0°C, 5°C and 10°C, respectively.

$$G(s) = \frac{Ke^{-\theta s}}{(\tau s + 1)} \tag{1}$$

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 as regards of the evaporation temperature, and they are being presented in Fig. 3.

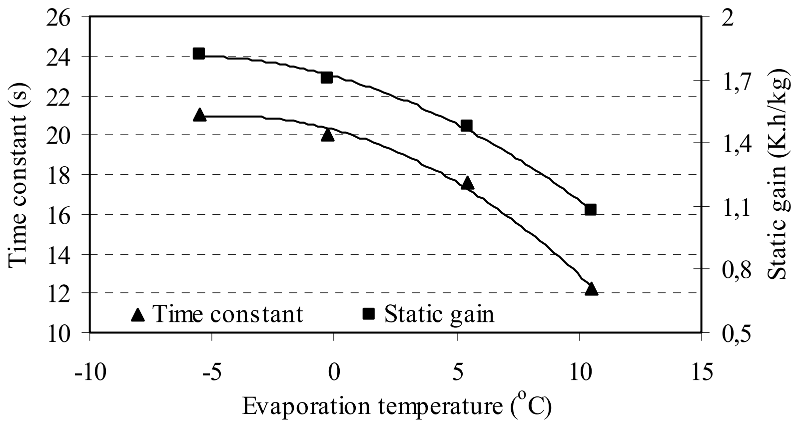

Figure 3. Time constant and static gain evolution as regards evaporation temperature

The static gain was estimated using the relation $\Delta(\Delta T_s)/\Delta \dot{m}_f$. As can be noticed in Fig. 3, the static gain value diminishes with the evaporation temperature growth. This behavior is justified by mass flow rate augment as a result of the evaporation temperature increase. Once the mass flow rate is the denominator of the expression that defines the static gain, its reduction is natural. The superheating did not change enough to interfere in this tendency.

The time constant correspond to the time instant in which the superheating reached 63% of its final variation. The time constant behavior observed in the Fig. 3 is also related to the mass flow rate augment with the evaporation temperature. Higher mass flow rate contributes to improve the heat transfer coefficient inside the evaporator, which promotes more efficient thermal exchanges reducing the time needed to achieve the steady state after a disturbance.

The data presented in Fig. 3 was used to define mathematical relations that describe the gain and time constant behavior according to evaporation temperature.

$$K = 0.0027 T_1^2 + 0.0321 T_1 - 1.7163 \tag{2}$$

$$\tau = -0.0399\,T_1^{\,2} - 0.3338\,T_1 + 20.254 \tag{3}$$

4. CONTROLLER DESIGN

The process of controller tuning is an important step in the controller implementation and if it is not performed in the right way the system can become less efficient or even unstable. There are several techniques to tune a PID controller; however, 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 strategy allowed calculating and updating the controller gains at each operating point.

The tuning method proposed by Vilanova (2008) is derived from optimization approaches and it provides a completely automatic tuning determined by the process parameters. This method was developed considering a first order plus delay time transfer function as a system model (Eq.(1)). The optimization problem was solved for this system resulting in the following set of equations:

$$T_i = \tau + 0.03\,\theta \tag{4}$$

$$K_p = \frac{T_i}{2.65\;K\,K_v\,\theta} \tag{5}$$

$$N + 1 = \frac{\tau}{T_i} \tag{6}$$

$$\frac{T_d}{N} = 1.72\,\theta \tag{7}$$

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≈10, but it was considered as being a design parameter, as can be seen in the Eq.(6) (Vilanova, 2008). It is important to emphasize that all parameters needed to solve Eq.(4) to Eq.(7) are present in Eq.(1). These equations working together with Eq.(2) and Eq.(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 Fig(4). As can be seen in Fig(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.

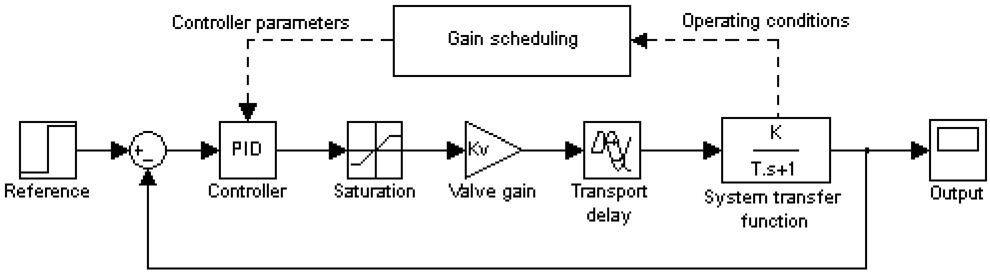

Figure 4 – Schematic representation of an adaptive controller with gain scheduling

To evaluate the controller performance using the presented tuning method some simulations were executed 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 witch the system provides the maximum refrigerating capacity. The delay time was considered to be 2s in all simulations. To evaluate the influence of the valve linearization errors in the system behavior, the valve gain was varied from 1.2 to 2.2. In the Fig. 5 is presented the system response to a step input for each value of Kv. It can be seen that the step response performance is acceptable and similar to the nominal condition.

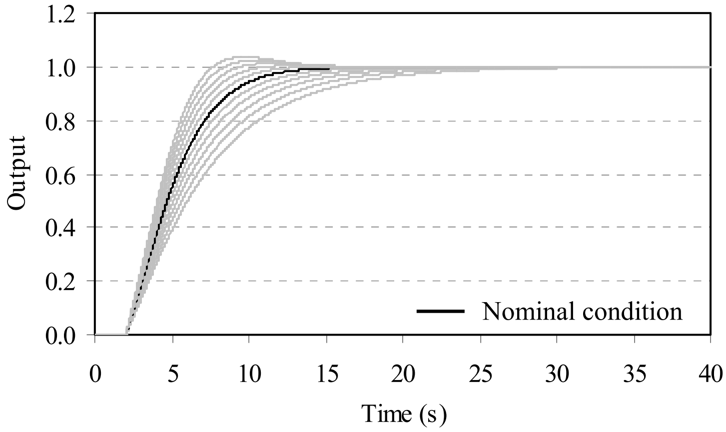

Figure 5. System response to a step input for Kv varying from 1.2 to 2.2

In the Fig.(6) is presented the system response for step input considering the evaporating temperature of -5°C, -2.5°C, 0°C, 2.5 °C, 5°C, 7.5 °C and 10°C. The system responses for all operating points almost match exactly because the tuning technique utilized places the poles in close loop at practically the same position.

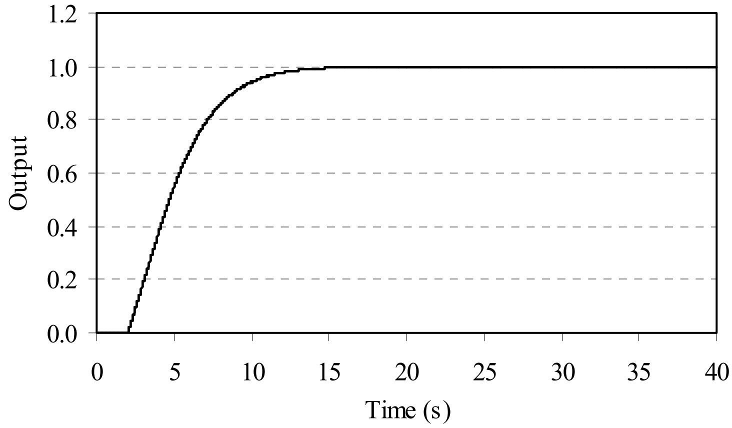

Figure 6. System responses to a step input for evaporating temperature varying from -5°C to 10°C

To evaluate the improvements obtained with the adaptive control, the controller gains where 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 conditions are being presented in Fig.(7).

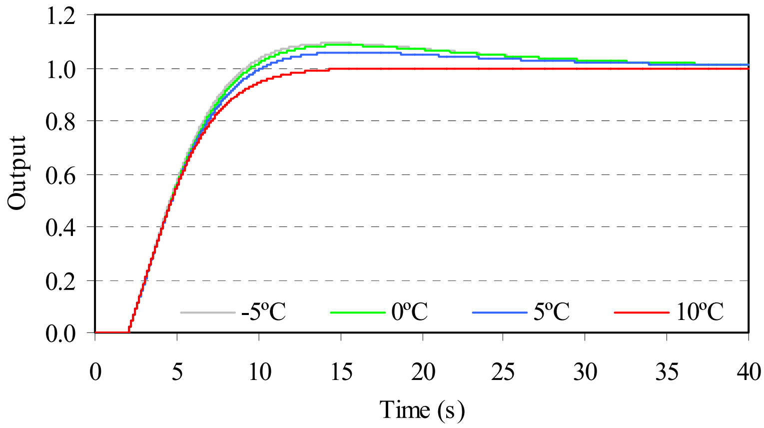

Figure 7. System responses to a step input for evaporating temperature varying from 10°C to -5°C in steps of 5ºC and controller tuned for evaporating temperature of 10ºC.

In spite of being acceptable, for evaporating temperature different from 10ºC the results obtained are not as good as the results presented in Fig. 6. 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 implement the proposed methodology, the whole control algorithm will be programmed in a PIC microcontroller. To meet all the requirements of this project it is necessary a microcontroller with two analog inputs that will be employed to read the signals from the temperature sensors, 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. 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. This microcontroller has eight channel analog to digital converter with up to 10-bit resolution, three programmable timers and several digital input/output (Microchip, 2003). The remaining pins will be set aside for future use. In Fig. 8 is presented a schematic view of the electronics assemblage.

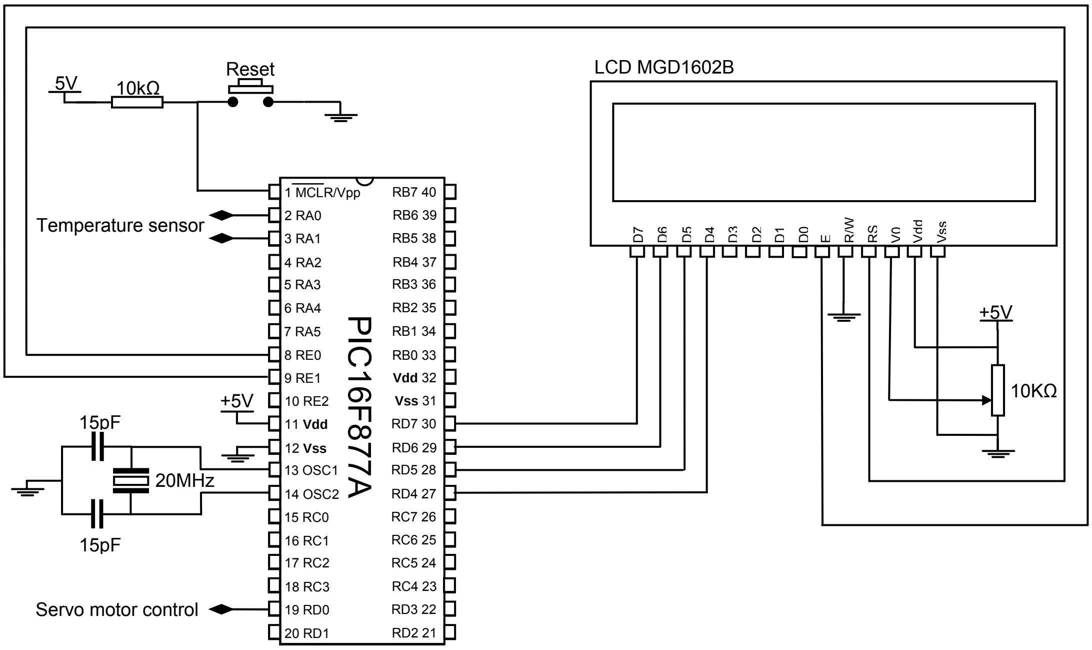

Figure 8. Schematic view of the electronic assemblage.

5.1. Controller algorithm

There are several ways to express the PID control law (National, 2001; Ogata, 2003; Vilanova, 2008). In this work will be considered the typical PID control law.

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

$$e(t) = SP - PV \tag{9}$$

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. By using this strategy, the controller output can be calculated by means of Eq.(10) (National, 2001).

$$u(t) = K_p \left\{ e(k) + \frac{1}{T_i} \sum_{i=1}^{k} \left[\frac{e(i) - e(i-1)}{2} \right] \Delta t + \frac{T_d}{\Delta t} \left[e(k) - e(k-1) \right] \right\} \tag{10}$$

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 will be based in the in the process variable rather than the error. In this way, the derivative part (u_d) presented in Eq.(8) can be written as:

$$u_d(t) = K_p\left(Td\,\frac{d(SP(t)-PV(t))}{dt}\right) = K_p\left(-Td\,\frac{d(PV(t))}{dt}\right) \tag{11}$$

Based on the Eq.(11), Eq.(10) can be written as:

$$u(t) = K_p\left\{e(k) + \frac{1}{T_i}\sum_{i=1}^{k}\left[\frac{e(i)-e(i-1)}{2}\right]\Delta t - \frac{T_d}{\Delta t}\left[PV(k)-PV(k-1)\right]\right\} \tag{12}$$

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 Eq. (12).

There are some additional aspects that must be considered during the controller implementation. The first of them is related with actuator saturation. The control signal u(t) must be restricted to a range that meets the actuator work range. The second is related with the integral part in the PID algorithm. Since it acts summing the process errors in order to define the necessary integral action, the final value obtained can be a large number, either positive or negative. Naturally, the control action as a whole should be restricted to the work range of the actuator. In spite of that, the integral part still is a large number and it can define the control signal direction over the proportional and derivative parts. If there is an error signal change, the direction of the integral control signal will not change until the actual error signal compensates the previous accumulated errors. This phenomena known as integral windup is undesirable because it can cause long periods of overshoot in the controlled variable. One way to prevent wind up from occurring is employing an extra restriction by software that interrupts the integrator work when the control signal reaches a maximum or a minimum value. In Fig. 9 is presented the controller program flow considering the aspects above mentioned.

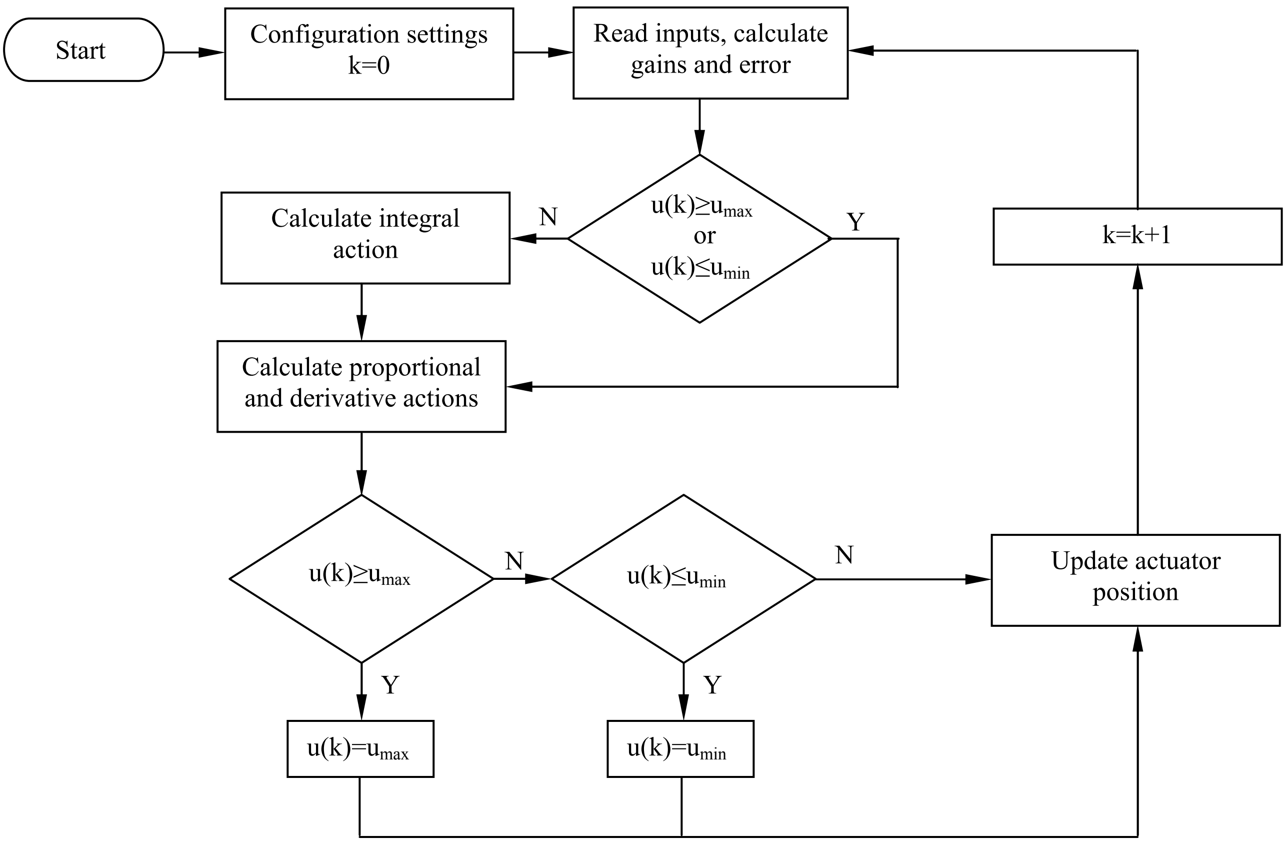

Figure 9. Controller program flow

6. CONCLUSION

In this study was presented an adaptive PID controller to regulating the superheating degree at the outlet of the evaporator. The controller tuning was based on the robust tune rule proposed by Vilanova (2008). The controller effectiveness was evaluated only through computer simulations. The simulations results showed that the self tuning control algorithm can control the superheating at the outlet of the evaporator with good performance, in different evaporating temperatures. The controller is robust and presented acceptable response even when errors of about ±30% were introduced in the expansion valve gain. The adaptive strategy showed to be advantageous providing better response when considering the non adaptive controller for the some operating range. At the end of this work it was presented the strategy and some considerations that will be taken into account when implementing the proposed controller.

7. ACKNOWLEDGEMENTS

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9. RESPONSIBILITY NOTICE

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