Faculdade de Engenharia da Universidade do Porto

Advanced Fuzzy Logic Heat Pump Controller

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Resumo

O processo que está na base do funcionamento de uma bomba de calor é amigo do ambiente e significativamente mais eficiente do que outras soluções convencionais de aquecimento. No caso especifico do aquecimento de água, podem ser conseguidas poupanças elevadas no consumo energético em relação ao uso de esquentadores ou caldeiras devido à utilização contínua durante todo o ano.

Sendo o mercado das bombas de calor ainda emergente, a procura pela máxima eficiência leva a que a operação deste tipo de sistemas seja otimizada em todos os pontos possíveis. A integração de válvulas de expansão termostáticas é ainda uma prática corrente na conceção destes equipamentos, contudo, não fornecem um controlo ótimo da quantidade de refrigerante admitida pelo evaporador. Esta situação promove um sobreaquecimento desadequado do fluido à saída do permutador de calor, levando à diminuição da eficiência e desgaste prematuro dos componentes.

Deste modo, pretende-se alterar a válvula de expansão termostática por uma válvula eletrónica adequada, cujo objetivo principal é o aumento global do rendimento do processo pelo controlo apropriado do sobreaquecimento.

Na presente dissertação é mostrado o procedimento realizado na conceção de uma solução passível de cumprir o objetivo. Foi elaborado um modelo do sistema e, através do mesmo em ambiente de simulação, projetou-se um controlador baseado em lógica difusa. Esse mesmo controlador foi posteriormente implementado na unidade de processamento da bomba de calor, assim como uma metodologia de seguimento do valor ótimo de sobreaquecimento para cada condição de operação.

No final deste documento, são apresentados os resultados atingidos pela solução projetada, assim como a comparação e os ganhos conseguidos em relação a uma bomba de calor idêntica mas com uma válvula de expansão termostática.

Abstract

The process that is the basis of operation of a heat pump is environmentally friendly and significantly more efficient than other conventional heating solutions. In the specific case of water heating, high savings in energy consumption can be achieved in relation to the use of heaters or boilers due to continuous utilization throughout the year.

As the market for heat pumps still emerging, the search for maximum efficiency means that the operation of such systems is optimized at every single point. The integration of thermostatic expansion valves is still a common practice in the design of these systems; however, this type of expansion devices does not provide optimal control of refrigerant mass admitted to the evaporator. This promotes inadequate heating of the fluid at the outlet of the heat exchanger, leading to decreased efficiency and premature wear of components.

Thus, it is intended to change the thermostatic expansion valve to an electronic suitable valve, whose main objective is the increase of the overall process efficiency by proper control of the SuperHeat.

In this thesis is shown the procedure performed in the design of a solution which could meet the goal. A model of the system was conceived which enabled the development, on a simulation environment, of a fuzzy logic based controller.

This was implemented in the processing unit of the heat pump as well as a method of tracking the optimum value of overheating for each operating condition.

At the end of this document the results achieved by the designed solution are presented. Additionally, a comparison between the proposed control and a similar heat pump with a thermostatic expansion valve is performed and the main gains are highlighted.

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Abbreviations and Symbols

Abbreviations (alphabetical order)

AC	Alternating Current
ASHPWH	Air Source Heat Pump Water Heater
COP	Coefficient of Performance
CPU	Central Processor Unit
D	Derivative
DC	Direct Current
DHW	Domestic Hot water
EN	European Standard
EXV	Electronic Expansion Valve
FEUP	Faculty of Engineering of the University of Porto
FS	Fan Speed
FLKB	Fuzzy Logic Knowledge Base
FLPD	Fuzzy Logic Proportional Derivative
FLPI	Fuzzy Logic Proportional Integrative
FLC	Fuzzy Logic Controller
FOTFTD	First Order Transfer Function plus Time Delay
GmbH	*Gesellschaft mit beschränkter Haftung* (Company with limited liability)
HMI	Human Machine Interface
HP	Heat Pump
IRS	Infrared Sensor
I	Integrative
LVDT	Linear Variable Differential Transformer
LUT	Look-Up Table
MSc	Master of Science
MSS	Minimum Stable SuperHeat
NTC	Negative Temperature Coefficient
PTC	Positive Temperature Coefficient
PWM	Pulse Width Modulation

PID	Proportional Integrative Derivative
PD	Proportional Derivative
PI	Proportional Integrative
P	Proportional
PT	Pressure Temperature
RTD	Resistance Temperature Detector
S.A.	*Sociedade Anónima* (Anonymous Society)
SCOP	Seasonal Coefficient of Performance
SH	SuperHeat
SP	Set Point
SM	Stepper Motor
SLHX	Suction Line Heat Exchanger
TC	Thermocouple
TXV	Thermostatic Expansion Valve
WP	Water Pump

Symbols List

e(t)	Error
Kd	Derivative gain
Ki	Integrative gain
Kp	Proportional gain
Ke	Error gain
K	Transfer function gain
k	Parameter of Steinhart-Hart Thermistor Equation to convert their resistance to temperature in Kelvin
Pb	Bulb pressure
Pev	Evaporator pressure
Q	Heat
Sv	State variable
T	Temperature
T	Transfer function time constant
u	Controller output
W	Work
w	rule weight
y	Output
Θ	Transfer function delay
ε	Efficiency
ΔT	Temperature variation
Δe_k	error variation
Δu	controller output variation

Chapter 1

Introduction

In this chapter is made an introduction to this thesis. It is presented a brief explanation of the project and the motivation for it, followed by the description of the internship place and also a summary of the proposed objectives.

1.1. The Project

The Dissertation, final discipline of the MSc in Electrical and Computer Engineering, provides the opportunity to develop a project in a business environment to students, in the form of an internship, in partnership with the Faculty of Engineering of the University of Porto (FEUP), that allows the enrichment of practical and work experience.

The entities involved intend an intelligent controller that will be integrated into a heat pump, similar to a model already on sale, that with appropriate modifications should increase the system performance and user comfort.

Usually heat pumps use mechanical expansion devices to control the SuperHeat value but those have intrinsic problems of instability and are not suitable for tracking variable references according to the operating conditions. When the system works with the optimal SuperHeat value, its performance and efficiency are significantly improved. So, replacing this with an electronic expansion valve and developing the appropriate algorithm of control it is expected to achieve the objective.

This final report incorporates, in first place, the introduction and contextualization of the developed project, then the state of the art technologies for the issues addressed, the system description and problems, followed by the procedure and solutions adopted and finally the results.

1.2. Motivation

The heat pump is a system that uses extremely efficient process compared to other solutions for heating up operations. Over the last years, a strong investment in development of this kind of machine, through improved control or better components, is being made.

The principle of operation is environmentally friendly, does not directly consume fossil fuels, and there is no release of any pollutants into the atmosphere.

The use of heat pumps for water heating can bring several advantages over the conventional systems in domestic or industrial use. So, many manufacturers are betting in this type of technology in order to lead the market of an emerging technology.

Besides the presented facts, this dissertation becomes attractive because it allows the opportunity for development and innovation in a leading company which offers excellent conditions and working resources as well as an environment of professionalism and rigor.

Bosch Termotecnologia, S.A. is the centre of development and innovation of the group Robert Bosch in domestic hot water products. The proposed project aims to use new methods in constant development with known potential which should increase the efficiency and effectiveness of the system and whose domain will certainly be an asset for future projects.

1.3. Company Presentation

Bosch Group

Robert Bosch GmbH (known as Bosch) is a German company founded in Stuttgart in 1886 by Robert Bosch (1861-1942) as a Workshop for Precision Mechanics and Electrical Engineering.

Today, presents itself as one of the largest private industrial corporations in the world: the Bosch Group. The world headquarters is located in Gerlingen, Germany.

The Robert Bosch Foundation owns 92% of Bosch Group and maintains the tradition of promoting several activities to improve the social and human condition, as stipulated by its founder, using their funds to support intercultural activities, social welfare and medical research.

In 2012, Bosch employed 306,000 people worldwide and increased the turnover of the group to 52.3 billion Euros, of which 4.5 billion were invested in research and development and applied for over 4700 patents.

Bosch Portugal is one of the subsidiaries of the Group and actuates in several areas, notably in automotive technology, industrial technology (automation and packaging equipment), construction technologies (power tools) and the production of consumer equipments (thermo technology, several appliances and security systems). It is situated in four distinct locations - Aveiro, Braga, Lisbon and Ovar. In 2011 the turnover was of 1,047 million Euros, employing 3,845 collaborators.

Bosch Termotecnologia, S.A.

Under the heading of Vulcano Termodomésticos S.A., the company started the activity in 1977, in Cacia, Aveiro, operating based on a licensing agreement with Robert Bosch GmbH.

Due to the quality of products, quickly became the market leader in the gas water heaters in Portugal.

In 1983, the Bosch Group acquired Vulcano, to which transferred skills and equipment and began a process of specialization within the Group.

Assigning now as Bosch Termotecnologia, S.A., gives the design and development of new products as well as their production and marketing. Not only for gas water heaters, as well as wall mounted boilers, solar heating panels and heat pumps.

Through brands like Bosch, Junkers, Vulcano, Buderus and Leblanc, is already present in 55 countries and diverse markets from Europe to Australia.

Due to the processes and policies adopted, the company has several important certifications in compliance with international standards.

1.4. Objectives

The challenge launched by Bosch refers to the development of an intelligent controller for a heat pump, which when integrated with the whole system should be responsible for an increased Coefficient of Performance (COP), according to the European Standard 16147.

For every refrigeration cycle, there is a minimum value for SuperHeat (SH) control which assures a stable operation of the system. This Minimum Stable SuperHeat (MSS) depends on several system characteristics and boundary conditions, such as air flow, air temperature, air relative humidity, water temperature, water flow, etc. Since boundary conditions change throughout the operation of the heat pump (water temperature changes while heating) and throughout the year (summer/winter), the standard procedure is to define a large enough SH to assure a stable operation for every condition. This situation is not optimal since the system performance decreases as the SuperHeat increases. So the set-point must be a function of the system real-time operating conditions.

The controller should capable of maintain a stabilized level of SuperHeat in the working cycle, near to the set-point, for every operating conditions. It should also guarantee the easy adaptation to new configurations of heat pumps, and the compatibility with the existing hardware or at least with only some minimal modifications.

The values provided by temperature sensors are the inputs to the controller while the output is the actual expansion valve opening.

The following image represents a high level of the system architecture, according to the description above.

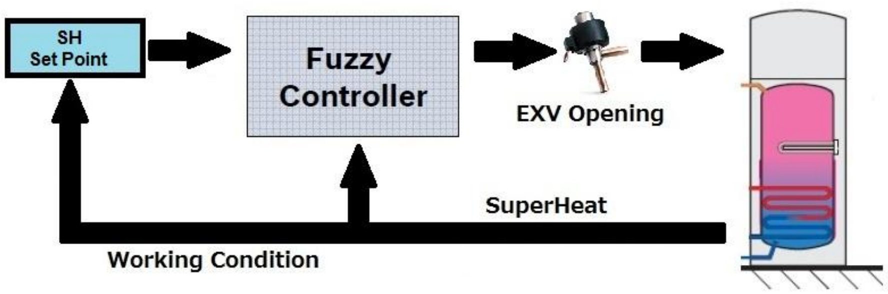

Figure 1.1 High level control architecture

In order to achieve the main goal of controlling the heat pump with a fuzzy controller the following objectives were defined:

- Modeling the SuperHeat behavior;
- Validation of the model with the system;
- Design the control architecture;
- Simulation of the model plus the controller;
- Implementation of the controller in the system;
- Testing and Tuning;
- COP Analysis.

1.5. Document Structure

This document is divided in 5 chapters. Each contains several subsections according to the subjects mentioned.

Chapter 2 introduces and shows an overview of the issues addressed and used technologies.

Chapter 3 details the proposed solution design, simulation procedure and achieved results.

Chapter 4 refers procedures of the controller implementation. Each step is explained and testing results are shown.

Chapter 5 presents an overview of the project; final conclusions are made as some suggestions about future possible developments.

Table 1.1 Document Structure

Chapter	*Title*
1	Introduction
2	State of the art
3	Simulation
4	Implementation and Results
5	Conclusions

Chapter 2

State of the Art

This chapter introduces the theoretical framework that underpins the matters dealt within the dissertation and state of development of the technologies used. It is then, a report of the preliminary study done at the time preceding the work on the solution for the proposed problem.

It is divided in three sections. The first one concerns heat pumps in general but focused on the Air Source Heat Pump Water Heater. A brief introduction is made to the efficiency and performance for this kind of system.

In the second one, the SuperHeat is explained, the methods and devices to measure it are presented, as well as the expansion devices and how do they control this phenomena, giving special attention to the electronic expansion valves and some of the control algorithms used.

The third and last refers the Fuzzy Logic that, as shown in the introduction, will be the base on the controller algorithm to the electronic expansion valve. It is explained how it works, its variants and presented some interesting adopted structures.

2.1. Heat Pump

The idea at the basis of the appearance of the Heat Pump is the energy transfer in form of heat, from one location to another. It is known that this process is not possible, in a spontaneous way, when you want to change heat from an external cold source to a warm receiver [1]. Thus, the system function is to allow, by carrying out work, the exchange.

As in almost any system technology, the development and emergence of the heat pump is due to the necessity.

The 1973 oil crisis lead to high fuel prices, and at that time it was concluded that in certain situations it is more convenient to capture heat from a cold source than the direct production of the same. But only in the year 2000 these systems began to spread faster. Besides the high price of fuel, also the pollution started to be a problem.

The emission of pollutants from the burning of fossil fuels is harmful to the planet and to human health. This time, the rulers of several countries began to adopt policies to encourage the installation of systems able to acclimate and warm water without the need to consume fuel [1].

2.1.1. Operational basics

Heat pumps have the working principle based on a modification on the Carnot cycle [2]: the reverse Rankine cycle, also called as Vapor Compression Cycle, where the turbine is replaced by an expansion valve. Ideally, it is defined by a cyclic process that uses a perfect gas and that includes four different phases:

- Isentropic compression;
- Isobaric condensation;
- Isenthalpic expansion;
- Isobaric evaporation.

Through these, it is possible to exchange heat between different locations. The same is illustrated in figure 2.1 adapted from [3].

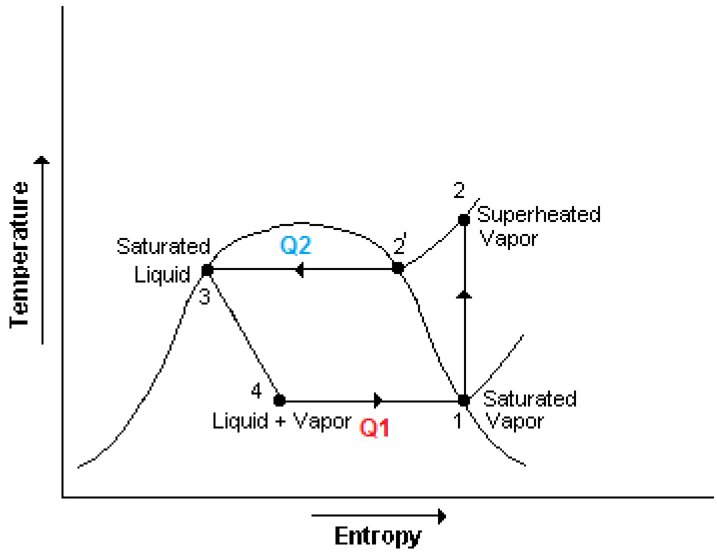

Figure 2.1 Inverse Rankine Cycle

In this example there is a heat exchanger in the source at temperature T_1 and another in the sink a temperature T_2, where Q_1 is the absorbed heat and Q_2 is the transferred heat. The transformations are represented in the paths between 1, 2, 2', 3 and 4, where:

- 1 to 2 (isentropic compression) the system performs work so that the internal energy increases with increased pressure and therefore temperature is increased;

- 2 to 3 (isobaric condensation) Superheated vapor condenses and at this time, the gas is in contact with the steady temperature T_2 system, rejects an amount of heat Q_2;

- 3 to 4 (adiabatic expansion) during this process there is no heat exchange with the environment. Pressure decreases and hence so does the temperature;

- 4 to 1 (isobaric evaporation) the gas is in contact with the system constant temperature T_1, receives the amount of heat Q_1.

The four processes listed occur through different elements. In the case of the heat pump, 2 to 3 and 4 to 1, take place through elements that maximize heat transfer between the different surfaces in contact namely condenser and evaporator. The two remaining are achieved using an electric compressor (1 to 2) and an expansion device (3 to 4).

2.1.2. Components

- Refrigerant: Fluid with low boiling point;
- Compressor: Responsible for the adiabatic compression of the refrigerant;
- Expansion device: Responsible for the isentropic expansion of the refrigerant;
- Heat exchangers (Evaporator / Condenser): Responsible for heat exchange between the refrigerant and the heat sink or source;
- Fans: Allow forced air circulation in heat exchangers;
- Sensors and Control Defrost: Elements responsible for the activation of the defrost cycle when it becomes necessary;
- Accumulator: A small container that can be used as a buffer and accumulator prevents liquid admission into compressor;
- Crankcase Heater: Used primarily in pumps in which the heat source is air, in order to increase the temperature of the oil, which facilitates the vaporization of the coolant and thus prevents the mixture between the two fluids, as well as improves lubrication [4].

2.1.3. Operation Cycles

Figure 2.2, adapted from [5], shows the two main heat pump operation cycles. Both are described below, the third one (defrost cycle) is characteristic of only air source heat pumps and is also described.

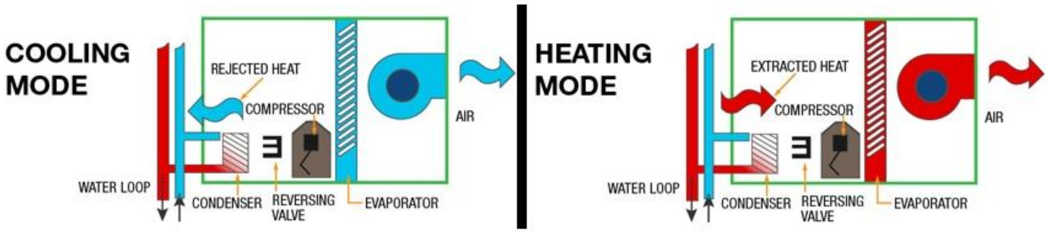

Figure 2.2 Operation cycles

Heating Cycle

Compressor and Expansion only ensure proper function when at the input the refrigerant is in one phase: dry gas or only liquid respectively.

Accordingly, superheated gas at low temperature and pressure is compressed by the compressor to a high pressure and, consequently, to a high temperature. The heated gas is then cooled and condensed in the heat exchanger until only liquid, preferably subcooled, is available. This liquid suffers an isenthalpic expansion in the expansion valve and gets back into a low pressure and low temperature liquid/gas mixture. The liquid portion is then boiled in the evaporator a further heated into superheated gas, returning to the compressor.

Cooling Cycle

This is similar to heating cycle but, in this case, heat exchanger functions are switched because it is intended to remove energy from the recipient. The operation is usually achieved through a four way valve that changes the order of both heat exchangers in the refrigerant circuit.

Defrost Cycle

This cycle only applies to heat pumps whose source is the air in the heating cycle. Sometimes, the evaporator operates at temperatures below 0°C. The humidity allows the formation of ice layers surrounding this element, which reduces heat transfer and hence the efficiency of the heat pump.

In order to remove accumulated ice, the system revert its operation, for a short period of time, to the cooling cycle; the evaporator takes over as condenser, which causes the ice to melt by heating the element [4]. Also, the introduction of another valve that bypasses the condenser and expansion valve can be used. With this, gas only passes through the evaporator and the compressor which highly increases evaporator temperature.

2.1.4. Classification

Heat Pumps are usually grouped according to the used heat provenance. The choice of source depends on several factors of which the most relevant are:

•Characteristics of the external environment;

•Limitations of legislative order;

•Income Required;

•Cost of installation;

•Time of return on investment.

There are three sources used by heat pumps as illustrated in Figure 2.3 [1].

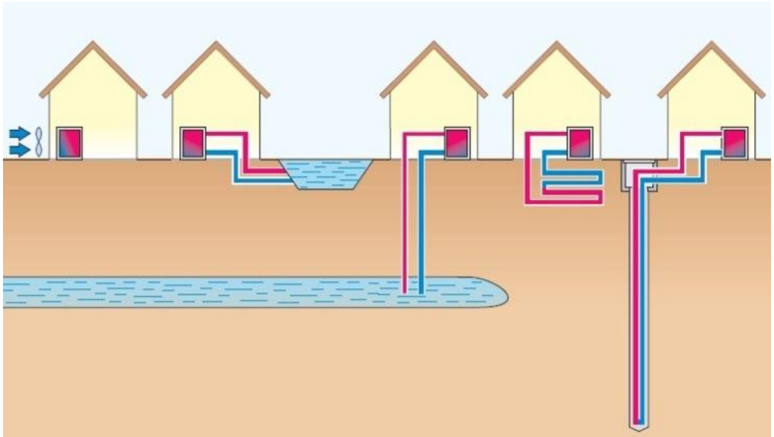

Figure 2.3 Heat sources

In the classification of heat pumps the fluids involved in the exchange can be used. The first referenced is the source, and then the receiver. According to this, there are six distinct types:

- Water – Water
- Water - Air
- Air – Air
- Air – Water
- Soil – Air
- Soil - water

Air

Is always available and can be used without requiring expensive equipment or special licensing. However, there is a high oscillation of temperature over time and when it passes a minimum threshold; the pump efficiency may drop dramatically. In this context, in most installations it is necessary to couple the heat pump with auxiliary heating sources.

Surface Water

Sea water, river or lake may also be used as a heat source. While there are no oscillations as fast as the air temperature throughout the year, in the colder months, low temperatures can also induce the same need of auxiliary heating elements.

Underground

Large amount of stored energy can be found, the source is solar near the surface and geothermal in deeper areas [1]. It can be used by water hole or pit, horizontal collectors, vertical probes or energy poles. These processes are also illustrated in Figure 2.3.

2.1.5. Air Source Heat Pump Domestic Water Heater

The Air Source Heat Pump Water Heater (ASHPWH) is a Air – Water heat pump whose purpose is to heat water for household or similar proposes. The main characteristic in detriment of others is the addition of a water tank and auxiliary heating element.

Accumulator of Inertia

In this type of function, in order to reduce the number of times the compressor starts and the use of energy on expensive tariff time, as well as the regulation of water temperature, leads to the incorporation of a well insulated water tank.

These tanks are also named as accumulators of inertia and include two main functions: Hydraulic separation and thermal flywheel [2].

The separation allows for the independence of the hydraulic pump flow to the flow of heat utilization, this is because hydraulic and thermal requirements are quite different, especially when used with variable consumption flow rate. Also, the cost of hp is directly related to its actual power, so an energy reservoir properly sized for the energy requirement reduces the energy consumption. The numbers 1 and 2 on the figure 2.4, adapted from [1], represent the two flow regulators for the independent water circuits.

The function of thermal flywheel reduces the number of times the heat pump is connected, which improves performance and reduces wear on the components. This function is possible due to the accumulator represented in the picture by the number 3.

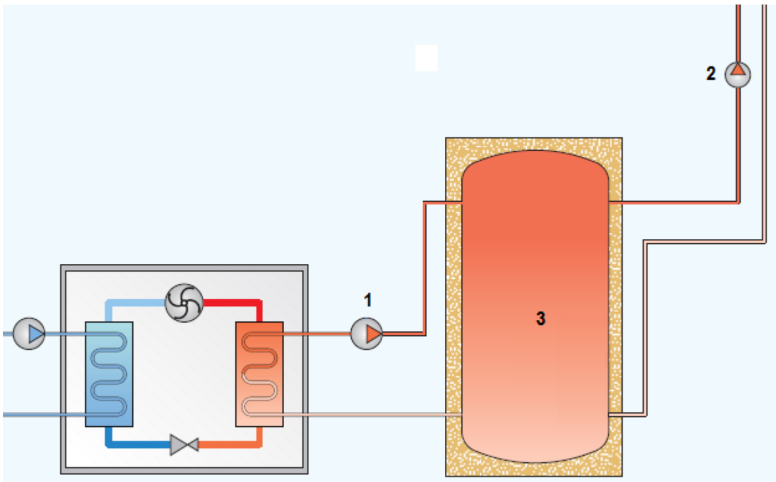

Figure 2.4 Heat pump water heater

The Process

In air source heat pump water heater, the evaporator collects energy from the outside air, which has been pre-heated by the sun and being drawn into the unit by the fan.

According to the cycle explained in 2.1.1, the hot vapor created by the compressor now enters the condenser where it is surrounded by water from the tank that also flows through the condenser system with the help of a water pump, causing the heat to be given up to the cooler water. The cooled refrigerant now returns to its liquid state, although it remains under high pressure from the compressor.

For better understanding, figure 2.5 [6] is a schematic representation of the steps described above.

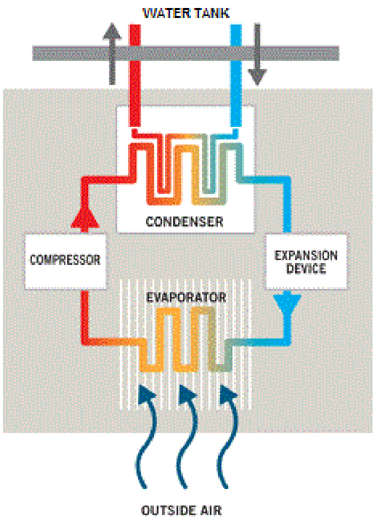

Figure 2.5 Refrigerant circuit

Auxiliary Heating

Compressor decreases dramatically the efficiency with low suction pressures, which leads to the lowering of COP; so, a maximum ΔT, or maximum difference between inside and outside temperature should be established. According to that, in the heating mode, there is a coldest temperature admitted for the heat source, below this point additional heat is needed. Most early heat pump systems augmented the reversed-refrigerant cycle heating with resistance electrical heating coils in the air handler. Electrical Resistance heating is not particularly efficient, however, so as energy became more costly, other auxiliary heat options were sought. Instead of resistance heating on the evaporator, the coils can be inside the tank, maximizing the heat transfer, being in direct contact with the heat receiver, and minimizing losses.

External Auxiliary heating is also used. When the environment conditions are not proper for the heat pump operation or it is desired a faster heating up, other systems like water boiler, gas water heater or solar panel can be coupled to the tank providing the necessary additional heat.

2.1.6. The Coefficient of performance

The instantaneous efficiency of the heat pump refers to well established and controlled testing conditions. It is given by the coefficient ε and relates mainly to the compressor. It represents the ratio between the heat transferred to the hot fluid Q and the energy W the compressor spends plus the auxiliary equipment.

$$\varepsilon = \frac{Qc}{Wcompressor + Waux} \tag{2.1.1}$$

Actually, the value of ε is the thermal power that can be obtained using 1kW of electrical power. It is easy to understand that this value increases with decreasing the temperature difference between the cold source and the fluid. It is easier to transfer energy from 10 ° to 30 ° than from 10 ° to 50 ° [2]. But this is not a good indicator of performance for heat pumps, because only one condition of operation is being measured.

The COP is the coefficient used to compare the performance between heat pumps. It can be measured according to different European Standards in different conditions. For certification in some markets, specific standards have to be followed.

European Standard 16147

For domestic hot water units, in France, the EN 16147 is one of those. This one supersedes the EN 255-3 and determines the COP_{DHW}, instead of the former COP, that is smaller since heat losses on the water storage tank eliminated. Also five different hot water consumption (tapping) profiles can be selected during the test, for different tank sizes [7].

Well defined initial and operation conditions are defined as measuring procedures and minimal requirements for the devices used in.

When the test begins, sequences of operations are made on the heat pump. They consists in six principal stages:

- Heating up period;
- Determination of standby power input;
- Determination of energy consumption and coefficient of performance for heating domestic water by using the reference tapping cycles;
- Determination of a reference hot water temperature and the maximum quantity of usable hot water in a single tapping;
- Determination of temperature operating range;
- Safety tests.

After the third stage it is already possible to know the COP value, which is given by the formula:

$$COP_{DHW} = \frac{Q_{TC}}{W_{EL-TC}} \tag{2.1.2}$$

Where Q_{TC} is the total useful heat during the whole tapping cycle in kWh and W_{EL-TC} is the total electrical energy consumption during a tapping cycle in kWh.

Seasonal Coefficient of Performance

Another performance indicator is the SCOP. This reports the annual efficiency of the installation.

Its value is given by the ratio of the heat transferred to the water for one year and the total energy expended to keep the plant in operation. Thus, its value depends not only on the performance of the heat pump, but on the entire facility which also includes all the systems of regulation and distribution of thermal energy.

The Seasonal COP is not easy to determine as it depends on several unstable variables, such as fluctuation of the cold source temperature, the type of control that manages the installation or regulation type which generates heat pump. One factor that greatly influences this value is the number of activations of the heat pump, since during the transient state the COP is much lower than the referenced [2].

2.2. SuperHeat

The value of SuperHeat is determined by the temperature exceeded by the gas after its boiling threshold, at the evaporator outlet, that significantly influences the performance of the refrigeration cycle in the heat pump.

The power exchange is more efficient if the fluid is in the liquid state while it passes through the evaporator because it is possible to have more fluid mass in contact with the heat source per unit of time than that which could be found in the gaseous state [8]. In the ideal case, the total mass of the refrigerant should only be completely boiled at the exit of the heat exchanger, not just for the reason above but also for the compressor element, to not incur on premature wear by the admission of fluid in liquid state.

Due to the change in the refrigerant's boiling threshold by the imposed pressure, it is possible to control where, in the circuit, the coolant passes totally into the gaseous state. At this point, so that the phenomenon occurs, the difference between fluid temperature and saturation threshold must be greater than zero. The cycle efficiency is increased if it is possible to maintain the SuperHeat value close to 0ºC. However it is imperative to ensure that the fluid is admitted to the compressor only in the gaseous state adopting a margin of safety adopted.

Minimum Stable SuperHeat

The concept is defined as a critical minimal degree of SuperHeat at which a refrigeration system could exhibit unstable operation. It was first proposed by Huelle in 1967 [9].

Huelle observed that the hunting of system parameters often occurred when a low degree of SuperHeat was set in a Thermostatic Expansion Valve (TXV) controlled evaporator refrigeration system. Therefore it was assumed that there existed a certain relationship between Minimum Stable SuperHeat (MSS) and the refrigeration load imposed on the evaporator. In 1972 introduced conceptually a so-called Minimum Stable SuperHeat signal line as shown in figure 2.6 [9]. In addition, he considered that the MSS in a refrigeration system was influenced by the inherited characteristics of the evaporator. In later years it was experimentally confirmed the existence of such a MSS line, as shown by Huelle [9].

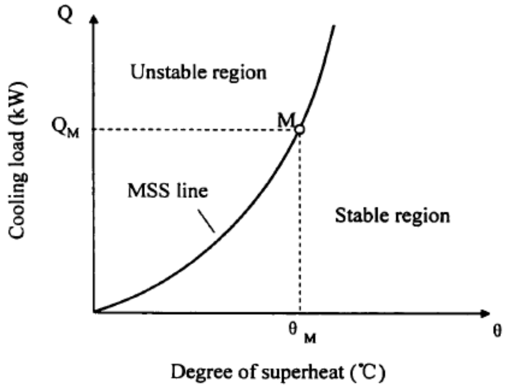

Figure 2.6 MSS line. Left and right side correspond to the unstable and stable regions

Normally exists two regions in an evaporator when in operation, namely a two-phase region and a superheated region, a moving boundary can be assumed separating them. The moving boundary is called mixture-vapor transition point as shown in figure 2.7 [9]. Researchers have experimentally demonstrated that the mixture-vapor transition point in an evaporator oscillated in a random manner even under steady operational conditions [9].

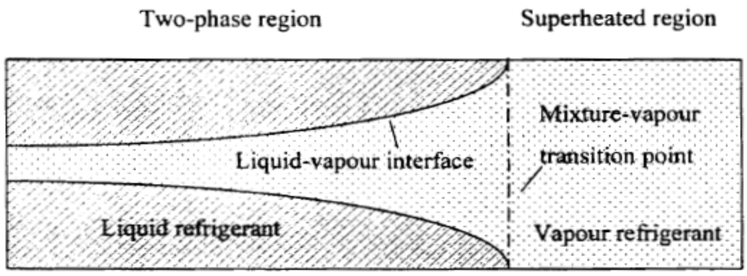

Figure 2.7 Refrigerant state on evaporator. Liquid enters on the left and vapor exit on the right side.

Figure 2.8 [10] illustrates the process of phase change that occurs in the refrigerant in the evaporator. As a result of thermal external action, the evaporation takes place in the evaporator over which the gas quantity is increased continuously relative to the amount of liquid. As soon as the gas occupies 10 to 20 times more volume than the liquid in the evaporator, the speed of the process increases rapidly. Accordingly, the condition at the evaporator becomes increasingly unstable. What remains at the end of the process is dry and overheated gas [10].

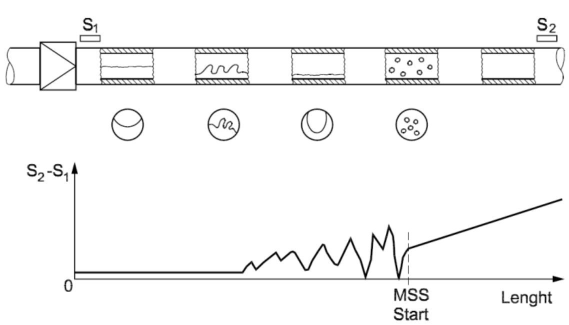

Figure 2.8 Minimum Stable SuperHeat

Adjusting the load of refrigerant in the evaporator is possible to control where the unstable zone occurs in the evaporator, guaranteeing that only refrigerant in gaseous state is admitted by the compressor.

The stable condition (MSS) is generally obtained with a SuperHeat of about 4ºC which can therefore be characterized as the lower limit of SuperHeat in the stable region. In practice, stable operating points are usually obtained around 5ºC to 7ºC [10].

2.2.1. How to measure

The SuperHeat is measured expeditiously through two sensors, the pressure-temperature (PT) table, characteristic of the coolant, and a simple mathematical calculation. An appropriate sensor measures the pressure at the evaporator outlet; using this value and according to the PT table of the refrigerant in use is determined the threshold of boiling each operating condition. Placing a temperature sensor in the same location it is verified the actual temperature of the gas. The difference between the temperature and the threshold determines the SuperHeat.

Alternatively it may be measured by the difference between the coolant temperatures at the outlet of the evaporator and the inlet. This is done assuming two premises. The first relates that the state of the gas at the inlet of the evaporator is biphasic; this means that part of the mass is in the gaseous state and the other in its liquid state, than in the evaporator the rest of the process occurs. The other assumption states that the evaporation takes place at constant pressure (ideal cycle), and therefore are disregarded any loss of load on the evaporator.

Through the control point of view, this introduces a concern. The values of the two temperatures have completely different time constants, as occurs on the TXV which introduces de hunting phenomena.

In Figure 2.9 [11] is represented the ideal cycle (blue line), and the real situation (red line). It is notorious what was described in the previous paragraph. Ideally, the heat exchange in the evaporator is represented by the lower horizontal line, in reality is represented by the dotted line above, so that it can also be seen that due to these approximations, the result obtained by the difference of the values of the sensors introduce a permanent error compared to the real value of SuperHeat. In addition to this fact, substituting a pressure sensor leads to another temperature disturbance through contact with the ambient temperature in which the sensor is located. Should also be considered the reaction time of the sensor to measure variations in the magnitude as well as offset errors that correspond to them.

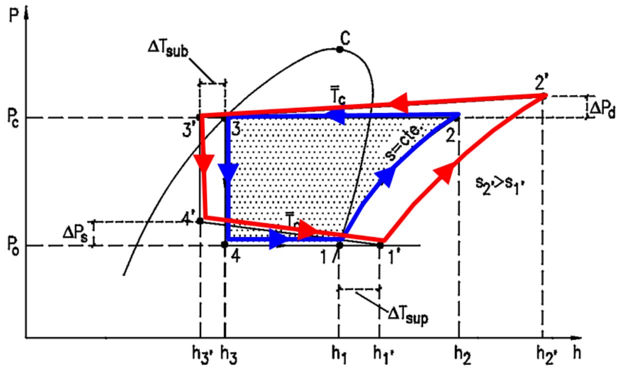

Figure 2.9 Refrigeration ideal cycle (blue) versus real cycle (red)

Temperature sensors

Thermocouples

Thermocouple is a pair of junctions that are formed from two dissimilar metals. One junction represents the reference temperature and the other is the temperature to be measured. They work when a temperature difference causes a voltage, by seebeck effect. That voltage is, in turn, converted into a temperature reading. TCs are used because they are inexpensive, rugged and reliable, do not require external power, and can be used over a wide temperature range.

Thermocouples can achieve good performance and can even be used for short periods of temperatures. They are also faster than resistance thermometers to react to temperature changes.

When using TCs, some considerations must be taken. They measure their own temperature so the temperature of the object must be inferred, and there must be no heat flow between them. They are also prone to temperature reading mistakes after long time of use because of insulation resistance loss to thermal conditions or nuclear radiation or even mechanical interference in the environment. These sensors are electrical conductors so they must not contact with another source of electricity [12].

Thermistors

As thermocouples, these are also inexpensive, readily available, easy to use, and adaptable temperature sensors. They are used, however, to take simple temperature measurements rather than for high temperature applications. Thermistors are made of semiconductor material with a resistivity that is much sensitive to temperature. These are widely used as inrush current limiters, temperature sensors, over current protectors, and self-regulating heating elements.

Thermistors differ from resistance temperature detectors in the material used and the temperature response. Thermistors can be classified into two types; depending on the sign of k. If k is positive, the resistance increases with increasing temperature, and the device is called a PTC. If k is negative, the resistance decreases with increasing temperature, and the device is called a negative temperature coefficient NTC [12].

Resistance temperature detectors

RTDs are temperature sensors with a resistor that changes the resistance value simultaneously with temperature changes. Accurate and known for repeatability and stability, can be used with a wide temperature range.

These sensors have a better accuracy than thermocouples as well as good interchangeability. RTD sensor is also stable over the long term. With such high-temperature capabilities, are used often in industrial settings.

Stability is improved when RTDs are made of platinum, which is not affected by corrosion or oxidation [12].

Infrared sensors

Infrared sensors (IRS) are used to measure surface temperatures ranging from -70 to 1,000°C. IRS convert thermal energy sent from an object in a wavelength range of 0.7 to 20 μm into an electrical signal that converts the signal for display in units of temperature after the compensation for any ambient temperature [12].

These sensors are used when the target is in motion, in vacuum or hazard, if temperatures are too high for contact sensors or a fast response is required.

When selecting an infrared option, critical considerations including field of view, emissivity, spectral response, temperature range, and mounting must be analized.

Pressure sensors

Potentiometric

Potentiometric pressure sensors use a Bourdon tube, capsule, or bellows to drive a wiper arm on a resistive element. For reliable operation the wiper must bear on the element with some force, which leads to repeatability and hysteresis errors. These devices are very low cost and largely used in low-performance applications.

Inductive

Several configurations based on varying inductance or inductive coupling are used in pressure sensors. They all require AC excitation of the coil(s) and, if a DC output is desired, subsequent demodulation and filtering is needed. The linear variable differential transformer (LVDT) types have a fairly low frequency response due to the necessity of driving the moving core of the differential transformer. The LVDT uses the moving core to vary the inductive coupling between the transformer primary and secondary [13].

Capacitive

Typically, these use a thin diaphragm as one plate of a capacitor. Applied pressure causes the diaphragm to deflect and the capacitance to change. This change may not be linear and is typically on the order of several pico Farads out of a total capacitance of 50-100 pF.

The electronics for signal conditioning should be located close to the sensing element to prevent errors due to stray capacitance [13].

Piezoelectric

Piezoelectric elements are bi-directional transducers capable of converting stress into an electric potential and vice versa. They consist of quartz or ceramic materials.

One important factor to remember is that this is a dynamic effect, providing an output only when the input is changing. This means that these sensors can be used only for varying pressures.

The piezoelectric element has a high-impedance output and care must be taken to avoid loading the output by the interface electronics. Some of these pressure sensors include an internal amplifier to provide an easy electrical interface [13].

2.2.2. How to control

SuperHeat control is made regulating the load on the evaporator. This means that if it is possible to control the flow of refrigerant in this heat exchanger, it is possible to vary the pressure and with that the boiling point characteristic temperature thus, the SuperHeat degree according to the definition made above.

Load regulation can be achieved with a variable speed compressor, or through the expansion device regulation, that is the common solution.

Expansion devices

The expansion device, commonly named as throttle is essential to the operation of the heat pump.

The valve's correct sizing and choice of topology play a key role in the improvement of the refrigerant circuit efficiency. If the system has a fixed speed compressor, this element is the only available to control the flow of refrigerant admitted on the evaporator as well as the pressure difference between the high and low side of refrigerant circuit. That is achieved by restricting the volume of refrigerant to a fraction of the capacity of the compressor. Controlling this pressure differential allows the refrigerant to boil in the evaporator at the desired temperature [14].

In this section are presented some of the solutions adopted among time as throttle devices.

<u>Capillary Tube</u>

As its name implies, this is a system that consists merely in a tube with a length usually between 0.5 to 6 meters long and with an inside diameter ranging from 0.5 to 2 millimeters.

The operation is summarized by the pressure drop along its length imposing load losses. As it runs, due to the friction and acceleration the coolant liquid will lose pressure, resulting in the evaporation of some fraction of it.

Several combinations of internal diameter and length of pipe can be adopted to achieve the desired effect according to the load applied and the other components of the pump. However, for a certain combination, is not possible to resize it according variations in the load or pressures involved.

Usually, the designer of a refrigeration circuit that incorporates a capillary tube chooses the diameter and length so that the equilibrium corresponds to the desired evaporating temperature. Then is used a tube longer than the theoretically designed, which results in an evaporation temperature lower than desired. The final length of the capillary tube is obtained cutting it successively until the desired balance condition is obtained.

Capillary tubes (figure 2.10) [14] are generally used in smaller systems, with capacities around of 10 kilowatts and operating conditions practically constant, since any change in the operation will be reflected in decreased efficiency of the installation [14].

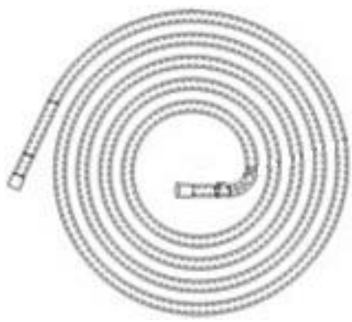

Figure 2.10 Capillary tube

Manually Regulated Expansion Valve

This kind of valve comprises a manually variable area orifice through which the refrigerant flows and feeds the evaporator. To increase the refrigerant flow running through it, it is necessary to operate a screw (figure 2.11 [14]) so the valve is opened. To reduce the valve is closed. The main disadvantage of this expansion valve is the inflexibility to changes in system load automatically and therefore must be adjusted each the system load,changes. Should also be opened or closed every time the compressor is turned on or off.

For these reasons, it follows that is only suitable for use in large systems with regular monitoring and actuation by an operator and where the system load remains nearly constant.

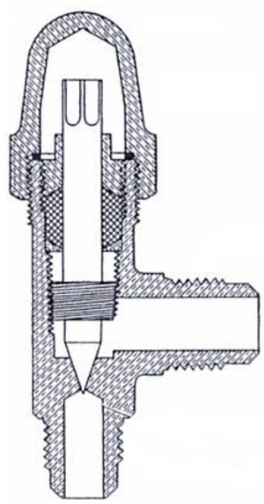

Figure 2.11 Manually regulated expansion valve

Float valve (High Pressure)

A float is placed in the high pressure side of the circuit and immersed in the liquid, which is the input variable to the valve control.

This type of device operates in a critical load of refrigerant in the system. When the hot gas is condensed, it flows into an accumulator chamber. Once the liquid level rises in the chamber the float rises too and the valve opens and allows the liquid refrigerant to pass to the evaporator. This control allows liquid to pass into the evaporator in equal amount to that which is condensed. In this way, an overload of refrigerant result in flooding of the compressor and insufficient refrigerant charge will result in underfeeding the evaporator. Often additional accumulators are used with the aim of making the refrigerant charge less critical. Figure 2.12 [14] illustrates clearly the principle of operation explained.

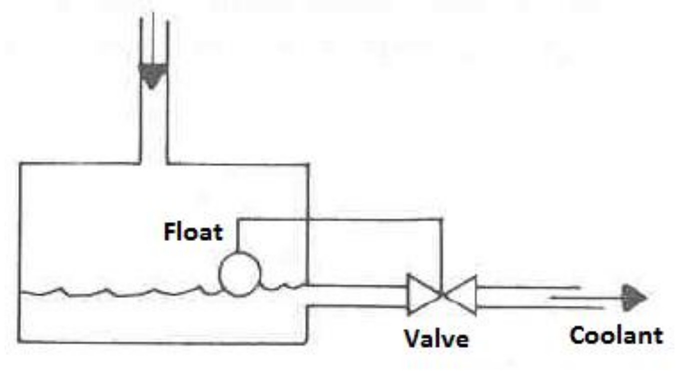

Figure 2.12 Float valve (High pressure)

Float valve (low pressure)

The principle of operation of this system is similar to that presented for the high pressure float valve. The name is due to the placement of the displacer in the low pressure side of the system as represented on figure 2.13 [14].

This actuator can be used only on systems with flooded type evaporator. The floater is placed directly on the evaporator or in a suitable adjacent chamber. However, they must be physically connected to the liquid level in both remains the same under any circumstances.

By increasing the load on the evaporator much liquid is evaporated and the level inside the float chamber and the evaporator lowers. When this happens, the float level is also lower, and is allowed more liquid from the high pressure side. When the load is decreased, the same happens in reverse way.

This type of valve is considered one of the best regulator devices that exist for "Flooded Systems", due to the excellent control that provides and constructive simplicity that makes it robust and practically free from damage. It can be applied to any system with flooded evaporator, whether small or large, and still used with any type of refrigerant.

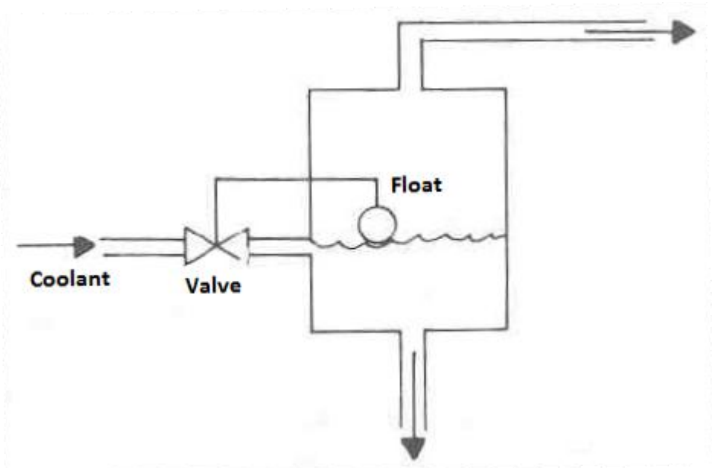

Figure 2.13 Float valve (low pressure)

Automatic expansion valve

This kind of device is used only in systems with single evaporator, especially for small capacities and reduced thermal load variations and applicable in almost all refrigerants.

The automatic expansion valve on figure 2.14 [14] operate on the principle of pressure reducing valve which during operation of the heat pump, retain the evaporator pressure to the set value regardless of temperature conditions, blocking the flow of refrigerant to the evaporator.

On the underside is subject to the force of a closing spring, whose adjustment is fixed, together with the evaporator pressure. At its top the atmospheric pressure acts and the force of a spring whose tension can be changed by previously adjusting screw situated in the upper valve.

The greater the spring tension adjustment, the greater the evaporation pressure and vice versa. Thus, each spring position adjustment, will correspond to a given evaporation pressure, which will automatically remain constant.

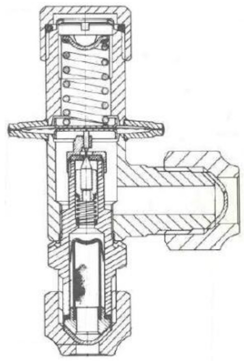

Figure 2.14 Automatic expansion valve

Electric expansion valve

It uses a thermistor to detect the presence of liquid refrigerant at the outlet of the evaporator (figure 2.15 [14]). When there is the presence of fluid in this state, the temperature of the thermistor is lower, which reduces resistance, allowing higher current intensity through the valve to open the fluid channel. This incurs on higher flow, regulating on this way the outlet temperature of the evaporator.

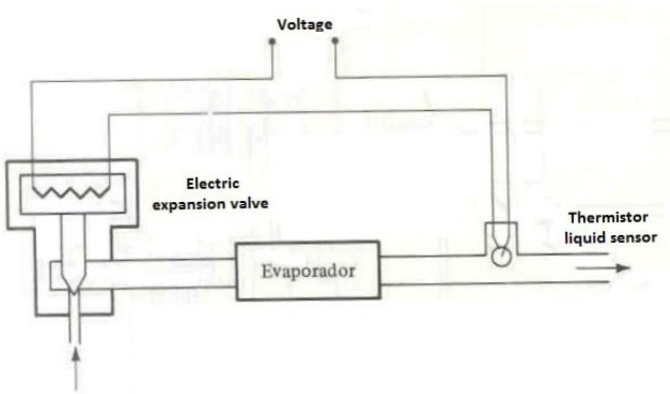

Figure 2.15 Electric expansion valve

Thermostatic expansion valve

The thermostatic expansion valve is the most used regulating device in residential and commercial applications. Include high efficiency and quick adaptation to various types of application.

Due to the automatic adjustment and performance, are meant to refrigeration and air conditioning with one or more evaporators.

Characterized by maintaining relatively constant range of overheating regardless of the conditions of the system, these promote proper feeding of the liquid in a wide range of conditions and thermal load, preventing liquid return to the compressor.

The thermostatic expansion valves may be of internal or external equalization depending on pressure loss in the evaporator.

The biggest drawback of the TXV is the “hunting” phenomena. Hunting is a cyclic fluctuation due to variation in the SuperHeat of the mass of fluid admitted to the evaporator. It results from excessive opening and closing of the expansion valve in an attempt to regulate the SuperHeat. This causes reduced system capacity and efficiency, waste of energy, decreased comfort, as well as premature degradation of other components of the heat pump, such as the compressor. The poor dimensioning of the expansion valve, the load with which the system works, the mass of refrigerant in the circuit or the contact of the bulb temperature sensor, are some of the factors that can result in the appearance of the hunting.

Due to its construction and principle of operation in closed loop, it is clear that the two variables implied in the control (pressure and temperature) are acquired with time constants quite disparate, so if the operating conditions of the system vary relatively quickly is the appearance of natural fluctuations over a period of stabilization.

Thermostatic expansion valve (internal equalization)

The operation of the thermostatic expansion valve with internal equalization depends on the pressure of the evaporator and the pressure created by the thermostatic control bulb.

The thermostatic bulb is installed at the outlet of the evaporator in thermal contact with the suction pipe continuously acquiring the temperature of the refrigerant at the outlet of the evaporator.

According to the figure 2.16 [14], the pressure Pb acts over the diaphragm (1), which depends on the temperature of the bulb. On the down side, in contrast, operate the evaporator pressure Pev and the spring pressure, Pm, through the transmission pin (2).

The movement of the diaphragm, on the downward, moves away the needle from the orifice, opening up a certain area, allowing the passage of coolant.

The opposite movement, upwards, due to the pressure of the spring (4) reduce the flow of liquid, which may reach up to complete closure.

It is found that the temperature rise of the bulb and therefore the pressure Pb, or also lowering the evaporator pressure, Pev, the needle (3) opens, providing increase of liquid refrigerant flow to the evaporator. However, with pressure drop of the bulb, and an increase in evaporator pressure, closes the needle passage, throttling the liquid refrigerant to the evaporator.

Within the chamber beneath the diaphragm and the outlet of the expansion valve (evaporator inlet) there is a communication path, so that the inlet pressure of the evaporator is transmitted to the diaphragm Pev. This path is called as the lower valve internal equalization.

Therefore, on the thermostatic expansion valve with internal equalizer, the evaporation pressure, Pev is equal to the inlet pressure of the evaporator, than is not considered a loss of load or resistance on the evaporator.

The thermostatic expansion valve with internal equalization is employed in systems with small losses (less than 20 kPa). Higher tubing length on the evaporator, leads to greater losses. Increasing the pressure loss, also the pressure difference between the inlet and outlet of the evaporator is bigger. This higher pressure will produce at the evaporator inlet, greater strength below the diaphragm. To remain the valve open, it is necessary that the pressure Pev increases increasing SuperHeat. To achieve this, is necessary that a part of the evaporator's area is used in a less efficient way.

The loss of this area, which is required for evaporation of the coolant, involves several possible consequences: evaporator partially covered with ice, poor heat transfer, reduced performance of the whole installation and a malfunction of the expansion valve.

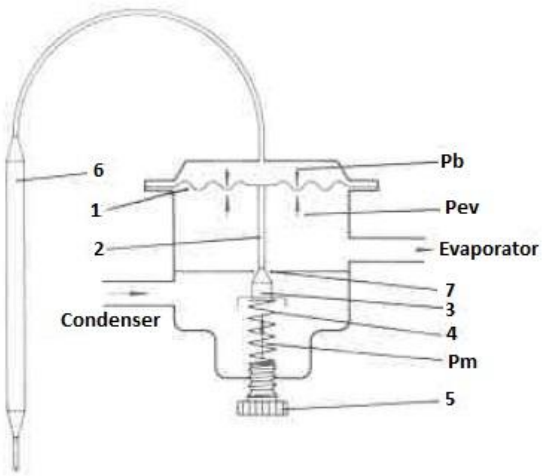

Figure 2.16 Thermostatic expansion valve (internal equalization)

Thermostatic expansion valve (external equalization)

In thermostatic expansion valve with external equalization (figure 2.17 [14]), the inlet pressure of the evaporator, Pev located in the lower chamber (A) has no contact with the diaphragm, due to separation by partition wall (9). The pressure of the evaporator Pev located in the upper chamber (B) is transmitted through the outer pressure equalizer, which is connected to the outlet of the evaporator. Therefore, the pressure Pev acting beneath the diaphragm valves equals the pressure of evaporator outlet.

The external equalization eliminates the influence of the flow resistance of the refrigerant (pressure loss) on the control processes of the thermostatic valves.

In cooling equipment for high load loss (pressure difference between inlet and outlet of the evaporator greater than 20 kPa), is recommended the use of thermostatic expansion valves with external equalization of pressure.

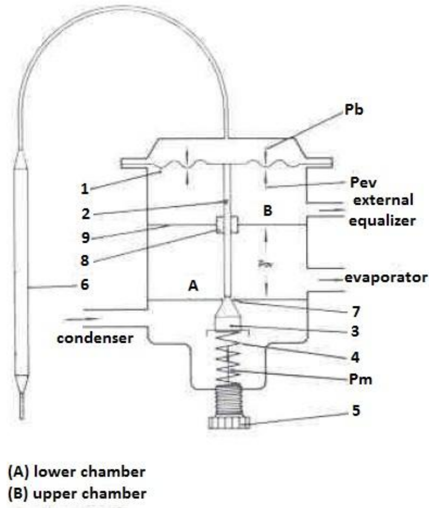

Figure 2.17 Thermostatic expansion valve (external equalization)

<u>Electronic expansion valve</u>

The principle of operation is similar to the manually controlled valves, but in this case, the valve is actuated by an electric device. The opening degree is given by an appropriate algorithm implemented in an electronic controller, usually designed to maintain a determined level of SuperHeat.

Thermostatic expansion valves have been commonly used to control refrigerant flow to the evaporator to match cooling capacity to the load controlling the SuperHeat. In the 1970's, electronic expansion valves were introduced, providing higher efficiency and better control, albeit at a higher cost. In the last years, electronic valves have been further developed and production scale economies have increased, improving performance and driving down costs. Nowadays it is common to see EXV in many applications [15].

Stepper Motor Technology

Stepper motors (SM) operate by rotating a set distance and stopping, based on electrical sequence of pulses from a controller. As the pulses are sent to the coils of the motor, a screw is turned pushing the needle valve for opening or closing an orifice as represented on figure 2.18 [15]. The resulting action is similar to a manual expansion valve. SM technology is commonly applied in a variety of systems, and generally handles single circuit evaporators more smoothly than PWM valves.

SM valve opening range can have a large number of steps, which increase the resolution allowing very precise control of refrigerant flow and smooth adaptation. The SM valves are comparatively more effective than PWM at very low load conditions by holding steady at just a few percent of capacity and without power applied.

These valves are designed to make precise step changes to adapt to system needs, but it is not possible to ensure that all the controller's commands are received and executed by the valve, sometimes resulting in a missed steps, which over time can aggregate to a significant error between the position of the controller and the actual step of the valve [15].

Another issue is that the more resolution the valve has, the more time the valve takes to go from fully open to fully close.

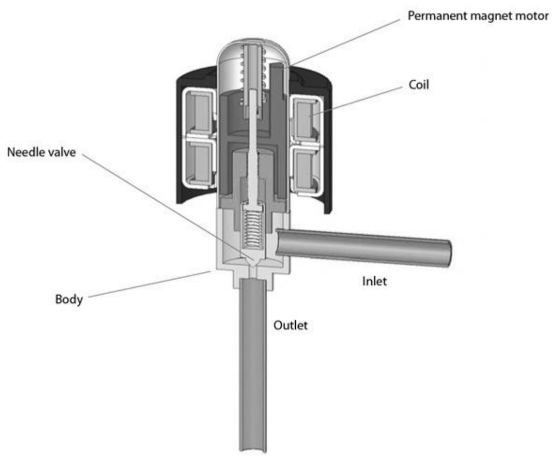

Figure 2.18 Stepper motor electronic expansion valve

Pulse Width Modulation Technology

PWM technology is based on a principle of fully opening and closing a solenoid valve imposed by a PWM control signal (figure 2.19 [15]). For example, to achieve 50% flow with a period of six seconds, the valve will open once, remain open for three seconds, close for three seconds, then the times are continually adapted based on refrigerant flow needs.

As SM, the PWM valves offer unique benefits. The turbulence created by opening and closing improves evaporator circuit distribution, increasing the usable area of the heat exchanger. The flow characteristics also improve oil return, which can be a particular challenge for SM valves in low load conditions. Electrical connections are simpler, with less wires and simpler controller algorithms. Since these valves are normally closed, they eliminate the need for an additional solenoid valve. PWM valves are better to adapt to faster capacity changes, moving from fully closed to fully open and vice versa in milliseconds.

These valves are particularly applied with multi-circuit evaporators. However, the pulsing current created in the system is a drawback especially in residential applications. Another negative point of PWM valves is power consumption compared to SM. Stepper Motor valves consume energy only when changes are required, PWM valves apply current to the coil holding power for the open portion of the cycle [15].

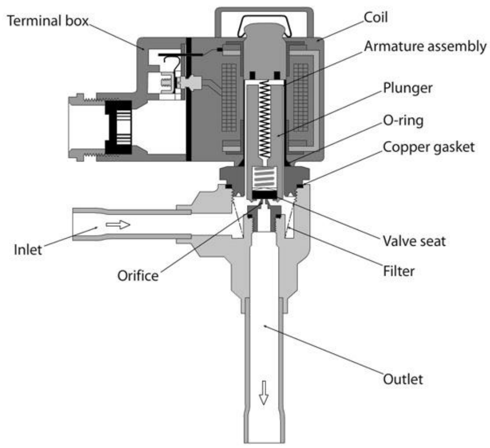

Figure 2.19 Pulse width modulation expansion valve

Electronic controllers

EXV control methods

Not much information is available about the used controller methodologies for the EXV regulation when tracking the SuperHeat. As expected, the majority of companies don't give much information about commercialized controllers. Market giants as Danfoss or Honeywell only announce "advanced adaptive controllers" [16][17]. Some others, like Sanhua, Dotech or Carel reveal that controllers are based on linear PID algorithms [18-20].

A constant in these controllers is the inputs used. As seen above, SuperHeat can be determined by one pressure sensor plus one temperature sensor, or by two temperature sensors, but every seen market solutions use the first option.

Some studies are published with some other experimented topologies. Liu, Wang and Chen (2008) [21] proposed an optimal PD controller based on genetic algorithm, in 2010 five Brazilian engineers developed an adaptive PID controller that according to the operational condition adjusts the controller gains in real-time [22]. Fuzzy logic based structures like Fuzzy PD are also presented by some documents [23] [24].

MSS tracking Methods

There are still in market solutions with fixed SuperHeat set point, Dotech Sporlan, Sanhua or Emerson Climate Technologies are some examples of that [19][20][25][26].

Danfoss and Honeywell are companies that already provide solutions of constant MSS line tracking. The used algorithm still secret, although Danfoss reveals that the controller is always downgrading the SuperHeat set-point, until a condition of instability is sensed at the input (figure 2.20 [17]). At this time the controller increases the set-point until the stability condition is reached [17].

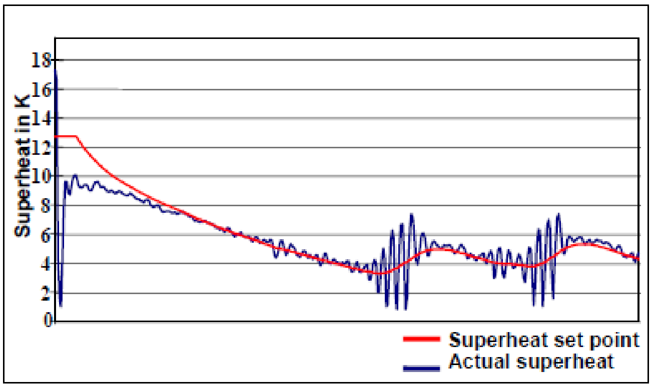

Figure 2.20 MSS tracking algorithm by Danfoss

Jolly, Tso, Chia and Wong (2000) developed a controller with the same principle of operation that is described by Danfoss, but the decision of increasing or decreasing the set point is made based on a fuzzy inference system. The given input is the integral of the error, sampled in a 60 seconds window, with that is possible to quantify the instability of the SuperHeat [23].

2.2.3. Conclusions

Thermostatic expansion valve is still largely used by manufacturers, is the cheapest and most proved solution to achieve satisfactory results when controlling the load in the refrigerant circuit. But when compared with electronic expansion valves there are several drawbacks. To improve efficiency by minimizing SuperHeat, to better control humidity by dynamically managing SH, and to respond more quickly to changing capacity requirements electronic expansion valves take place.

Figure 2.21 from [15] shows the impact of applying MSS theory that can only be achieved with EXV: chart shows a simplified TXV SH a fixed setting and the idealization of MSS theory. Regardless of the choice of PWM or SM valves, applying MSS theory can make systems operate with less degree of SuperHeat in most applications than those with a fixed SH setting, which do not achieve optimal efficiency, and can even result in lower efficiency than a TXV. As such, once a PWM or SM valve is chosen based on the application, it is imperative to use the adequate control algorithms to ensure the desired results.

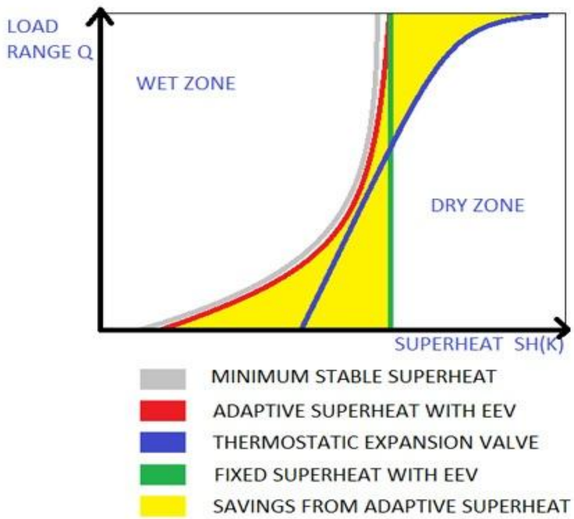

Figure 2.21 Minimum Stable SuperHeat theory savings

Overall, PWM has better performance in multiple circuit units where more turbulence improves refrigerant distribution. On the other hand, on systems with one circuit including most air-conditioning or heat pump applications, SM valves are the best choice to avoid excessive pulsation. PWM valves can improve oil return and can adapt more quickly to fast changes in capacity needs, but SM valves can be more effective in low load conditions and consume less energy in operation [15].

2.3. Fuzzy Logic

Fuzzy logic is a methodology for control or modeling systems that can be used from small systems to large and complex networks. It is possible to implement in hardware, software or both. It is a simple way to be able to set an output based on inaccurate, vague or ambiguous input information. The method resembles the decision making of a human being, but faster and more accurate.

The concept was conceived by Lofti Zadeh, professor at the University of Berkeley in California. It was introduced in 1965, not as a method of control, but as a way of processing information through inaccurate and incomplete sets.

Zadeh's theory is based on the fact that people don't need precise or numerical information to make decisions, and still be highly capable of providing adaptive control to systems. If closed loop controllers can be programmed to accept noisy and inaccurate input, this way it would be even more effective and easy to implement. In 1974, Professor Ebrahim Mamdani implemented the first controller based on fuzzy logic in the United Kingdom.

Only in the late 70s, fuzzy logic began to be used in control systems due to insufficient processing capacity regarding the computers of the time. In the early days, the theory was not widely accepted by the system manufacturers. Later, European and Japanese began to give him confidence and application. These days, there are numerous applications of the Lofti Zadeh's concept. It can be found in areas such as control, supervision and monitoring, decision support systems, classification of information, computer vision, pattern recognition, and knowledge-based systems [27].

When using conventional logic, well-defined boundaries are used to characterize sets. In many situations, this is not the correct approach. For example, to classify the age group of a person based on their age certain limits are created. If you set the age of 20 years as the threshold between a young and an adult means a person aged 20 years and one day is no longer a young man. However, this is not an assignment with meaning in reality.

Fuzzy logic allows defining the degree of membership of a variable to a set using fuzzy sets for it.

Fuzzy set

Fuzzy set is the expansion of the notion of set, in order to make possible the representation of concepts defined by fuzzy boundaries, such as those that arise in natural language or in qualitative concepts. The name of each set is defined by a linguistic variable.

Membership function

The membership function indicates the degree to which a particular concept is a member of a set. The degree is given by a value between 0 and 1, and when it is 0 indicates that the variable has no relation with the used set. The degree of membership with value 1 means that the variable is completely represented by the set.

In figure 2.22 [28] it can be seen that the temperature value represented by the vertical line intersects two sets and has a high degree of belonging to the set cold (blue) of approximately 0.8 and a low degree of belonging to the set warm (orange), near the 0,2.

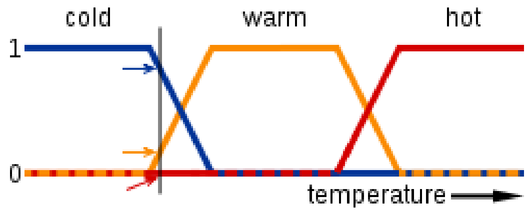

Figure 2.22 Membership function for fuzzy set

Rules

The rule-base is the main part of the FLC. It is formed by a set of logical rules. Rules determine the relationship between the set of input variables and output fuzzy region. Those can be either conditional or non-conditional. An example of a proposition, considering u1 and u2 system inputs and y1 is the output:

If u1 is COLD AND u2 is WARM THEN y1 is MEDIUM.

In this case, the intersection method should be defined to the operator "AND", usually is used the minimum or the product of the degrees of membership of each variable in the respective set. Other logical operators may be used like NOT and OR.

Each rule works in the following way: If the conditions of the antecedent are satisfied, then the conclusions of the consequent are applied.

The importance of the consequent in the controller's output is calculated by the antecedent. In the previous example the antecedent corresponds to the underlined part of the statement. Admitting that the method chosen for the AND operator was the minimum, the importance of the rule would be the lower value of the degree of belonging of the variable u1 in the set HOT and the variable u2 in the set COLD. The part that is not underlined in the proposition is called the consequent.

Usually, in a conventional or fuzzy controller, the inputs are error and error deviation for set-point tracking. The rules presented in Table 2.1 operate with linguistic labels of the inputs and output of the Controller. Their purpose is to describe in a qualitative way the control law of the FLC.

Table 2.1 Example of a Rule-base of a simple FLC

IF error is positive **AND** error change is approximately zero	**THEN** the output is positive
IF error is negative **AND** error change is approximately zero	**THEN** the output is negative
IF error is approximately zero **AND** error change is approximately zero	**THEN** the output is approximately zero
IF error is approximately zero **AND** error change is positive	**THEN** the output is positive
IF error is approximately zero **AND** error change is negative	**THEN** the output is negative

2.3.1. Control architecture

The following scheme (figure 2.23) represents the steps of the fuzzy logic controller in a block diagram. The function of each block is now presented.

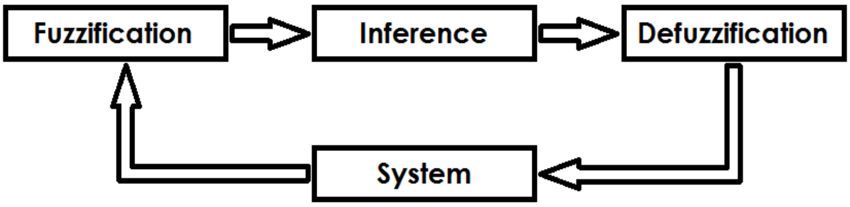

Figure 2.23 Block diagram for fuzzy controller architecture

Fuzzification

The fuzzification is the first stage of the controller. At this time, it is analyzed the system's state and acquired the value of the variables of interest. By the intersection with the fuzzy sets in variables value, related linguistic variables are defined and created regions according to the degree of membership in each set.

The membership functions can take many forms (figure 2.24 [29]) and should be appropriate to the type of application and desired result. In the next are examples of functions that are typically used to describe fuzzy sets.

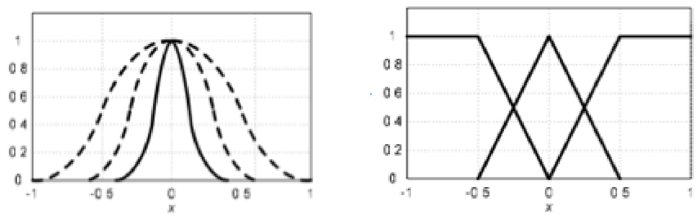

Figure 2.24 Typical membership functions. Gaussian on the left, triangular on the right side

Inference

At this stage, the rules are analyzed and the ones suitable for the situation through the inference method will create the resulting region.

Two methods can be considered: Mamdani and Takagi-Sugeno; for both it is necessary to calculate the weight of each antecedent proposition through as described above.

Mamdani

A region is created through an implication method for each rule. It is generated, for each consequent linguistic variable; a set whose region occupied is a function of the weight of the rule.

Next, another method is applied to aggregate all regions generated by each of the active rules. The resulting region is the entry to the last stage. Figure 2.25 [29] is the illustration of the process.

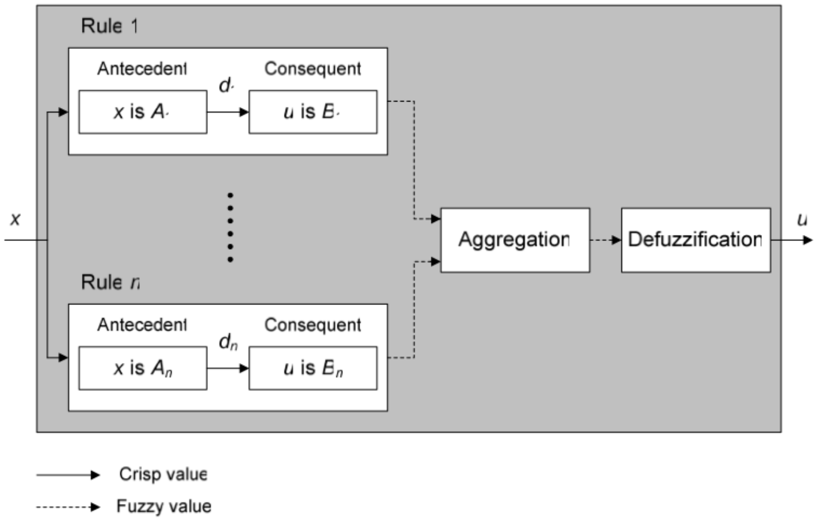

Figure 2.25 Mamdani inference block diagram

Takagi-Sugeno

When the chosen inference method is the proposed by Tomohiro Takagi and Michio Sugeno, the consequent of the proposition is a function (usually linear) of the inputs. After the determination of each rule's output, the value is multiplied by the weight of the corresponding proposition [30]. The controller output corresponds to the average or sum on this set of results.

Being the outlet a direct function of the system input, the Defuzzification process is intrinsic to this method as seen in figure 2.26 [29].

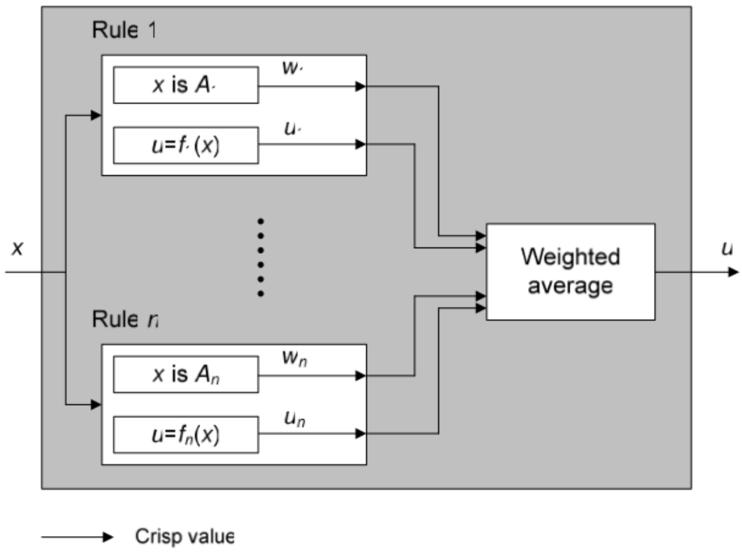

Figure 2.26 Takagi-Sugeno inference block diagram

Defuzzification

In this final stage the resulting region is converted into the output value of the controller. It corresponds to the functional link between the regions and the fuzzy expected value.

The objective can be achieved by different methods. Among them the most common is the "centroid" in which the center of mass of the resulting region will be the output value for the controller. Other methods such as First-of-maxima or Mean of maxima may be used [30]. These are illustrated in figure 2.27 adapted from [29].

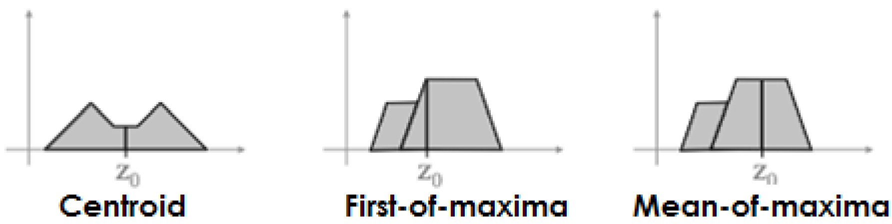

Figure 2.27 Defuzzification methods

2.3.2. Application Examples

Fuzzy logic is already implemented in several areas. Below are some illustrative examples of them with respective application.

- Defense: Systems to support decision making;
- Electronics: Automatic exposure control in video cameras;
- Aerospace: Altitude control for satellites;
- Car: Automatic transmission algorithms;
- Signal Processing and Telecommunications: adaptive Filter for nonlinear control of channel equalization for large spectrum noise;
- Industry: Optimization of production;
- Navy: Control of autonomous underwater vehicles;
- Medical: Diagnosis of diabetes prostate cancer;
- Financial: Stock market forecasts.

2.3.3. Why Fuzzy Logic controllers

Since the appearance of control architectures based on fuzzy logic, several reports of successful implementation of such controllers are published.

A.S. Kamal et al. (1996) applied fuzzy logic to control the refrigerant flow of a refrigeration system. Its performance has been compared with that of a well-known commercial controller. Fuzzy logic achieved better control and improved the performance [31].

Elangeshwaran et al. (2006) explains the advantages of a fuzzy logic based controller over a PID controller from experimental results. Better control performance, robustness and overall stability can be expected from the fuzzy controller even though developing and tuning of the FIS is more intuitive than the PID controller [32].

M. Manjunath et al. (2011) concluded that Fuzzy control combined with conventional PID controller is a way of intelligent control. Fuzzy adjusts the PID control parameters depending upon the error. A two input and three output fuzzy adaptive PID control was simulated in MATLAB environment and the results show that the fuzzy adaptive PID controller have better stability, small overshoot and fast response [33].

Fuzzy logic is not the only way to reason with ambiguous concepts but it is a good alternative to model approximation and control engineering. According to the concept, is easy to enounce some particularities about fuzzy logic:

- It can deal with ambiguousness and uncertain information and still producing the adequate output.
- The intuitive knowledge base design allows easier interpretation and adjustments, even when the user is not a control engineer.
- The algorithm is not heavy from the point of view of computational requirement. In some cases, the controller can be resumed to a Look-up Table with some associated interpolation method.
- Validation, consistency, redundancy and completeness can be checked in rule bases.

The concept is used in many other paradigms, and can also be incorporated with conventional controllers design, i.e. tuning them for certain plant non linearity due to universal approximation capabilities. In the case of nonlinear processes which can be linearized around the operating point, conventional controllers can work successfully, but these are not optimal in the whole working area [34].

Through figure 2.28 [35] is clear to observe the possibilities that arise from the integration described above. On the left side is shown a control surface of a conventional linear controller, on the right the generated surface of a fuzzy logic controller. Adjusting the fixed gains of the conventional controller the only change in the surface occurs on the slope of the represented plan, instead, with fuzzy logic is possible to change the shape of the surface according to the constraints and nonlinearities imposed by the plant.

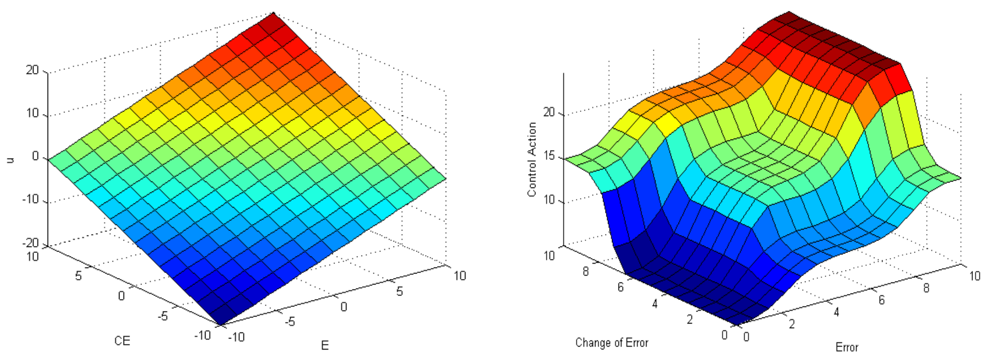

Figure 2.28 linear controller surface (left) versus nonlinear controller surface (right). The lower axes refer the inputs and the vertical one is the controller output

2.3.4. Typical fuzzy controller structures

Some of the typical adopted structures results from the junction of i.e. sliding mode or even conventional Proportional Integrative Derivative (PID) family of controllers. This can be done either by fuzzy adjustment of the gains of the conventional controller, according to the operating condition (fuzzy gain scheduling), or by the emulation of the controller law using knowledge base rule systems (Fuzzy like).

Fuzzy Gain Scheduling

This type of controller utilizes a rule based auxiliary scheme for gain scheduling issues.

It is implemented expecting that operating regions are associated with overlapping membership functions of the fuzzy sets defined in the scheduling variable space and that a fuzzy inference mechanism is used to dynamically interpolate the controller parameters around region boundaries based on known local parameters.

According to state variables of the system, the operating point is determined. In some cases, it can be done through the same variables used in the main conventional controller; others use new state variables so the Fuzzy Logic Knowledge Base (FLKB) is capable of adjusting the gains of the main structure.

Gain scheduling PID with fuzzy knowledge base is exemplified on the block diagram of the figure 2.29. In this example, the two state variables are the inputs to the fuzzy inference mechanism that provides gain adaptation to the three components of the PID.

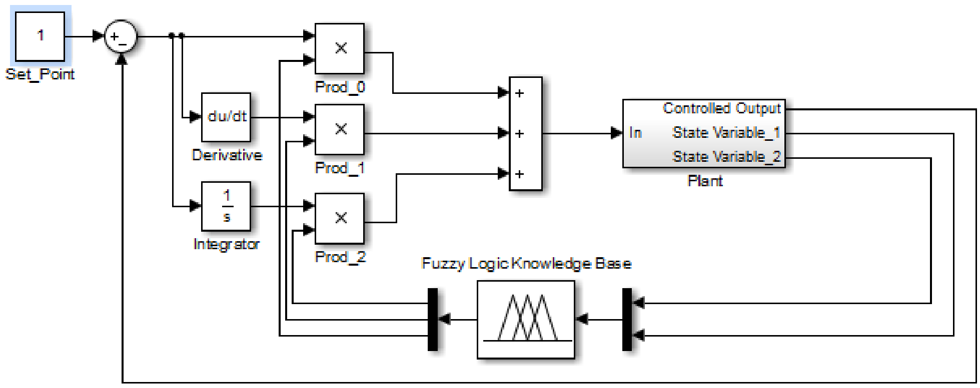

Figure 2.29 Gain scheduling PID controller (fuzzy based)

The control law of the conventional parallel PID is given by the equation (2.3.1), where Kp, Kd, and Ki are the static gains of the controller.

$$u_{PID} = Kp.e(t) + Kd.\frac{de(t)}{dt} + Ki * \int_0^t e(t).dt \tag{2.3.1}$$

When α, β and θ are the three outputs of the Fuzzy Logic Knowledge Base, the output is:

$$u_{FLPID} = \alpha.e(t) + \beta.\frac{de(t)}{dt} + \theta.\int_0^t e(t).dt \tag{2.3.2}$$

Where:

$$\alpha, \beta, \theta = F_{FLKB}(Sv_1, Sv_2) \tag{2.3.3}$$

Fuzzy Like controllers

This variant of controller uses the knowledge rule base to be part of the main structure. The output is provided directly from the fuzzy inference system. In the case of the Fuzzy like PID controller, depending on the adopted structure, there can be a three way input, given by the derivative of the error, the integral of the error, and the error itself. Figure 2.30 represents one of the structures used to achieve this type of controller. It results from the fusion of Fuzzy like PI and Fuzzy like PD.

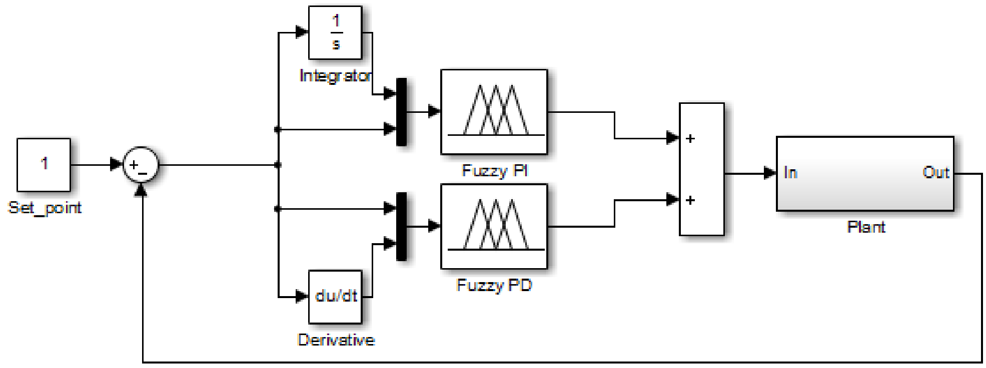

Figure 2.30 Fuzzy like PID controller

The control law is now given by:

$$u_{FLPID} = u_{FLPD} + u_{FLPI} \tag{2.3.4}$$

$$u_{FLPID} = F_{FLPD}(e(t), \frac{de(t)}{dt}) + F_{FLPI}(e(t), \int e(t)dt) \tag{2.3.5}$$

F_{FLPD} and F_{FLPI} are described by the used fuzzy logic inference system.

A fuzzy PID controller can be derived from some configurations. From a practical point of view, the most frequent parallel combinations of fuzzy like PID controllers are: PI+PD, PD+I, PI+D or P+I+D [36].

2.3.5. Adaptive Fuzzy Control

Most of the real-world processes that require automatic control are nonlinear by nature; their parameter values alter as the operating point changes. The necessity of retuning the characteristics when the operating point changes, has driven to the automatic adaptive controllers [34]. This means that the controller by itself changes its parameters according to its actual performance.

Adaptive controllers generally have two extra components plus the standard controller as illustrated on figure 2.31.

The first is a process monitor that detects changes in the process plant. It usually is a performance measurement for the controlling task or a parameter estimator that constantly updates a model of the real process. The second component is the adaptation mechanism itself.

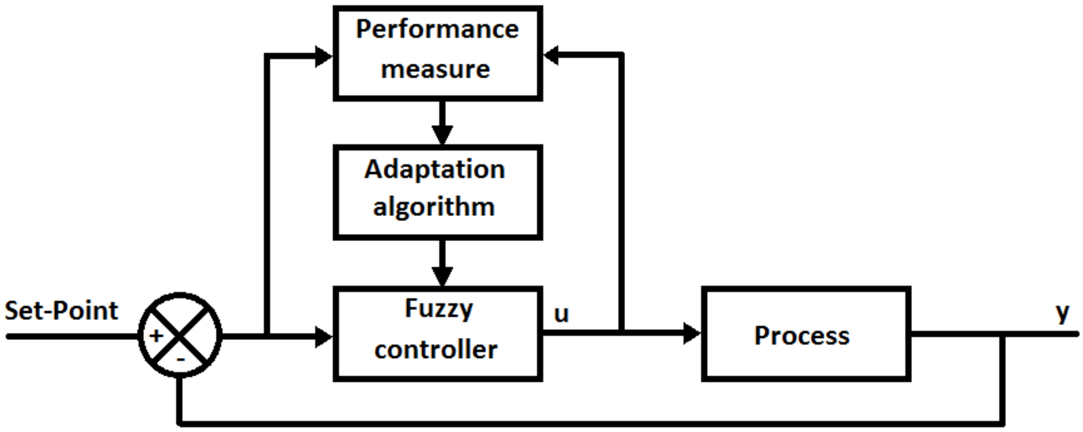

Figure 2.31 Adaptive fuzzy controller

Performance Measures

The choice of one or more performance measures depends on the type of response the control system designer wishes to achieve. The final output of the performance monitor, as seen by the adaptation mechanism, is either the actual values of the measurements, or a performance index derived from the variable readings.

For the regulatory control problem, where the aim is to keep a process variable on its specified set-point, many performance-related variables can be used:

- Overshoot;
- Rise time;
- Settling time;
- Decay ratio;
- Frequency of transient oscillations;
- Integral of the square error;
- Integral of the absolute value of the error;
- Integral of the time-weighted absolute error;
- Gain and phase margins.

Adaptation Mechanism

The adaptation mechanism must modify the controller parameters to improve the controller performance according to the information of the output from the process monitor. There are three ways of adaptation. The first is altering scaling factors; is similar to adjusting the gain in a PID controller but in this case, the input or output variables are multiplied by a scaling factor. With this is possible to change the sensitivity of the controller to the inputs or the actuation gain.

The second is modifying fuzzy set definitions; this one allows small changes in the shape of the fuzzy sets in order to improve the performance of the controller.

The third is the self organizing methodology, which the main purpose is to change the rules from the knowledge base by trying to identify the rules responsible for the poor performance and replacing them for better ones.

Chapter 3

SuperHeat Modeling and Simulation

The third chapter describes the used methodology to achieve a model capable of describing the behavior of the SuperHeat according to the opening degree of the expansion valve in all operating conditions. The controller architecture and its simulation are also presented.

It is divided in four sections. The first one introduces the company's given prototype and describes its key features for the proposed objective.

Then, the adopted solution for SuperHeat modeling is presented and justified and is made a description of how the different variables and parameters were estimated and determined. The validation of the model is also presented in this section.

The third section is related to the controller architecture and the reasons for the adopted solution.

Finally, some considerations are made about the integration of the model with the controller in the simulation and the results are shown.

3.1. System Specification

Bosch provided a prototype for the project which is a complete heat pump similar to the production ones, with the particularity of the expansion device. In this case, instead of the usual thermostatic expansion valve, there is an electronic expansion valve.

According to the single circuit evaporator, the low mass of refrigerant used, the possible residential application of the heat pump and energy consumption issues, a Stepper Motor EXV is used rater than a Pulse Width Modulation EXV which is in agreement with section 2.2 of this document.

There were also added two Negative Temperature Coefficient temperature sensors, the same as used in water or air temperature measures for the heat pump control, but these ones are for sensing the Evaporator inlet and outlet temperatures for the estimation of the SuperHeat.

The choice of two temperature sensors instead of one for pressure and the other for temperature is due to the production savings, and the avoidance of major modifications in the heat pumps that are already in production.

NTC sensors have suitable characteristics for the application. The necessary range of temperatures is not wide and the resolution achieved is enough when few tenths of degrees are not a much relevant change in SuperHeat.

In this prototype, variable water pump flow and ventilator speed are available. With this, is easily and faster to emulate changes on the operating conditions, like water or air temperature without having to change the actual variable physically that consumes a lot of energy and time. The control of these elements is possible through direct action in the Human Machine Interface (HMI).

3.1.1. Main Characteristics

Figure 3.1 schematically identifies the main components of the heat pump. Also identifies the inputs and outputs of the Central Processing Unit (CPU).

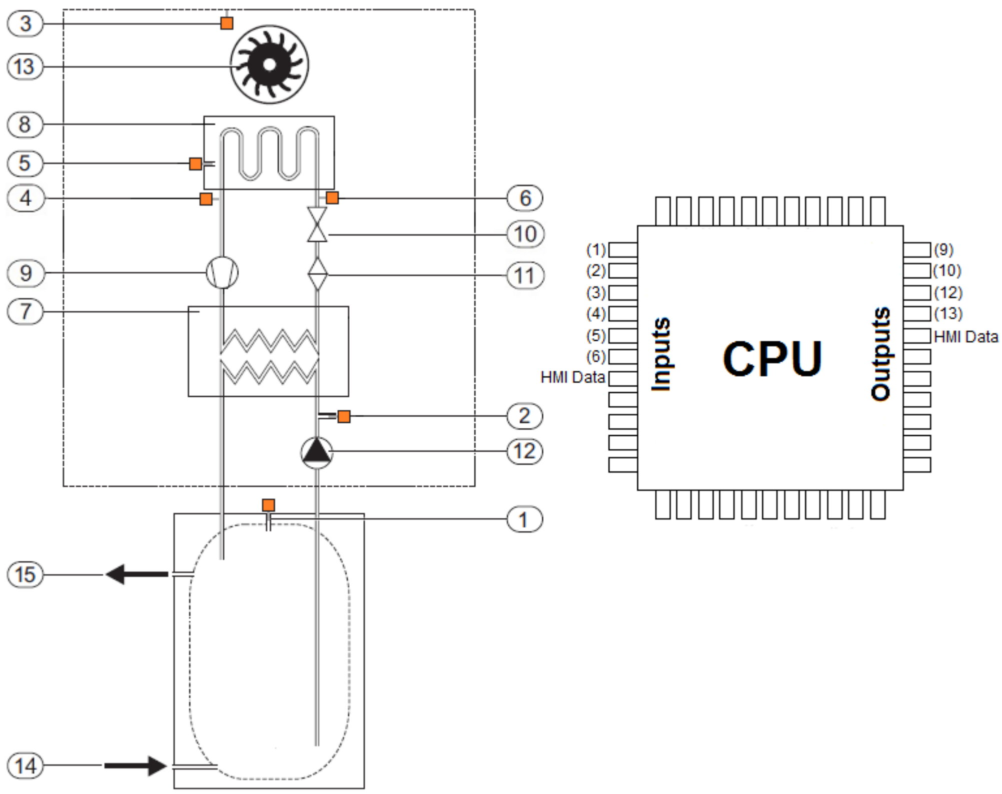

Figure 3.1 System specification

(1) Top Temperature sensor
(2) Bottom Temperature sensor
(3) Air Temperature sensor
(4) Fins Temperature sensor
(5) Evaporator Outlet Temperature sensor
(6) Evaporator Inlet Temperature sensor
(7) Condenser
(8) Evaporator (Single Circuit)
(9) Compressor (Fixed Speed)
(10) Electronic Expansion Valve (SM)
(11) High Pressure Switch
(12) Water Pump (Variable Flow)
(13) Fan (Variable Speed)
(14) Water inlet
(15) Water outlet

3.1.2. Temperature Sensor

- Type: NTC thermistor;
- Maximum temperature: 100ºC;
- Minimum temperature: -30ºC;
- Rated resistance: 10kΩ ± 3% @ 25ºC;
- Maximum deviation: 4.6% @ -30ºC;
- Thermal time constant (in water): 8s.

Acquisition circuit is represented in figure 3.2:

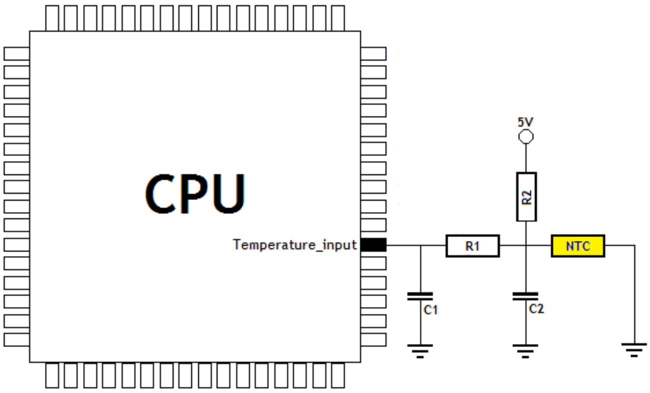

Figure 3.2 NTC acquisition circuit

3.1.3. Expansion Valve

- Type: 4 Phase Stepper Motor;
- Resolution: 500 steps;
- Operating range: 40 to 500 steps (experimentally determined)
- Max flow: 1.8l/m;
- Minimum temperature: -30ºC;
- Maximum temperature: 70ºC;
- Design pressure: 42 bar (600 psi).

Driver circuit is represented in figure 3.3:

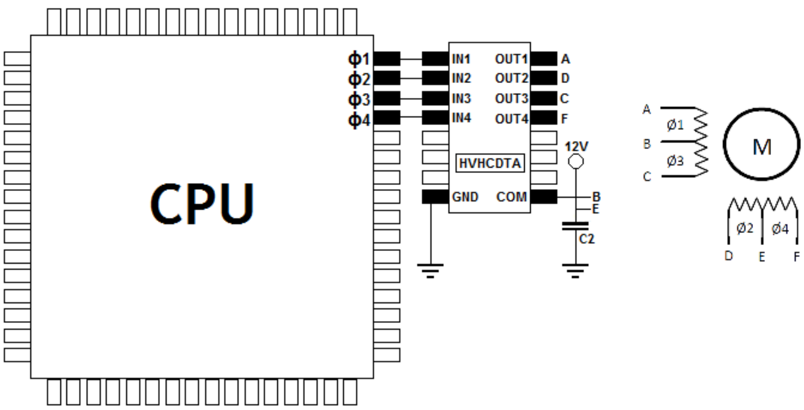

Figure 3.3 Electronic expansion valve driver circuit

3.1.4. FAN

- AC Supply: 230VAC, single-phase;
- Control: PWM;
- Max speed: 2580 rpm;
- Resolution: 0..255 (duty cycle: 0..100%);
- Operating range: 50 to 200 steps.

In order to determine the usable range of the PWM applied to the fan, PWM value was decreased until the fan stops, and then increased until the rpm change is no longer noticeable.

3.1.5. Water pump

- DC Supply: 12V;
- Control: PWM;
- Resolution: 0..255 (duty cycle: 0..100%);
- Operating Range: 40 to 110 steps.

With a flow to frequency transducer, a test was made in order to determine the actual flow of the DC water pump according to the selected PWM and the usable range of values. The results are shown in table 3.1 and in a graphical way from figure 3.4. The water pump stops when PWM step is under 40, and a significant increase in the Power/Flow ratio occurs above 110 steps on the water flow.

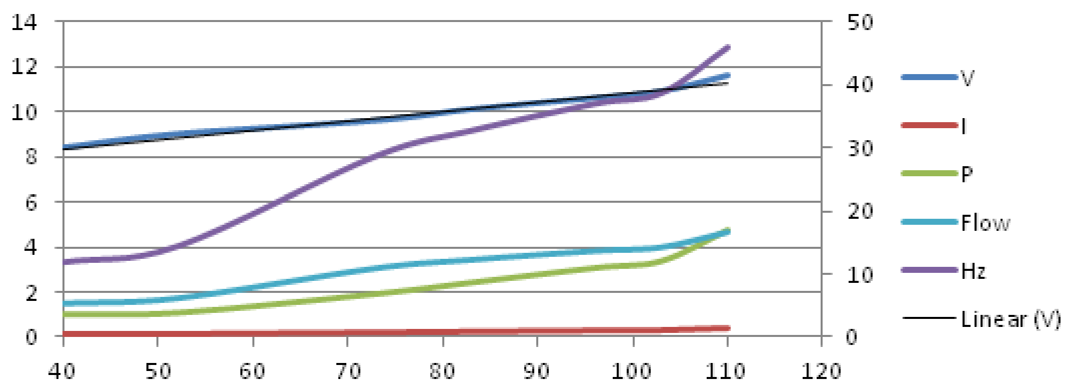

Figure 3.4 Water pump flow rate chart

Table 3.1 Water pump flow rate

PWM Value	*U*	*Current*	*Power*	*Frequency*	*Flow*
49	8.4	0.12	1.01	11.8	1.464
52	9	0.12	1.08	14.3	1.702
73	9.6	0.2	1.92	28.7	3.073
83	10.1	0.24	2.42	32.7	3.453
96	10.6	0.29	3.07	37	3.863
103	10.9	0.31	3.38	38.7	4.024
110	11.6	0.41	4.76	46	4.719

3.1.6. HMI/Central Processor Unit

(Only relevant characteristics for SuperHeat control is presented)

- Temperature acquisition sensibility: 0.1ºC;
- Stepper actuation period: 8ms;
- Expansion valve controller period 250ms.

3.2. System Modeling

The determination of the mathematical model of a complete generic heat pump is a long process that involves highly nonlinear phenomenon and complex thermodynamic equations. Looking at the area of research, this is not within the objective, considering that the model is only necessary to characterize the SuperHeat behavior.

Accordingly, an alternative to the complete model that is capable of describe only the behavior of the necessary variables, with sufficient accuracy, and that enables the design of an appropriate solution has been searched.

The transient response of SuperHeat as a function of mass flow rate passing through the expansion valve can be approximated by a First Order Transfer Function plus Time Delay (FOTFTD) [37-39][22]. Thus, for this purpose it becomes unnecessary the elaboration of a computationally heavy mathematical model for the whole heat pump. Through this model approximation, factors like temperature sensors time, delay or offset errors are already considered in the parameters determination.

Since the characteristic output of the impulse response is similar to the one represented on Figure 3.5, K, τ and θ are extrapolated graphically as illustrated.

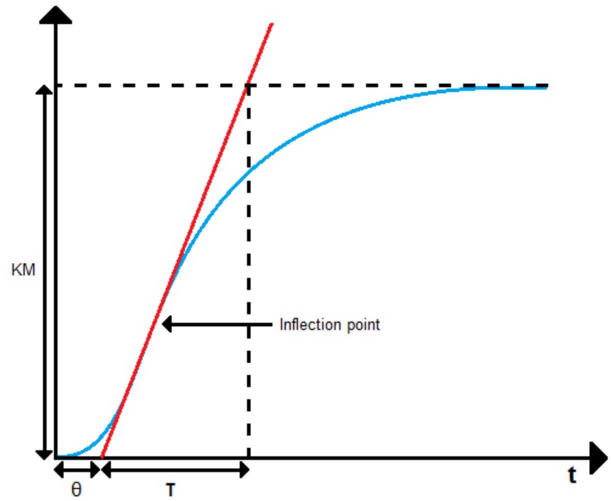

Figure 3.5 Graphical transfer function parameters determination

1. The process gain is determined by the ratio between the total variation of the temperature (KM), and the value of the step in the actuator (M): $K = \frac{KM}{M}$
2. Draw the tangent at the inflection point of the step response. The intersection with the time axis indicates the delay of the system.
3. The intersection of the tangent with the line KM indicates the time instant must be used to determine the time constant, since at this point t = τ + θ.

Analytical determination is also possible:

1. Define T0 as the time when the change in actuator occurs;
2. Locate where the measured process variable first shows a clear initial response to the step change, and name it as T1;
3. Locate where the measured process variable reaches 63.2% of its total final change and call it T2;
4. Time constant is the difference between T2 and T1;
5. Time delay is the difference between T1 and T0;
6. Process gain is the ratio between the total variation of the temperature and the step in the actuator.

Minimum Stable SuperHeat

The Minimum Stable SuperHeat condition, so that the system remains stable, will not be considered in the SuperHeat model. The philosophy that is intended for the controller regards the robustness and applicability of the algorithm in different heat pumps.

Two possible options could be considered to solve the problem of Minimum Stable SuperHeat.

The first is the determination of specific curve of the threshold for stable operation for each heat pump. This would involve intensive and controlled tests to encompass all possible operating ranges. However, it was still imperative a safety margin more or less extensive, depending on the accuracy and the reliability of sensors used, as well as corresponding manufacturing process i.e. the control of load applied to the refrigerant circuit. Factors that change the dynamics of the refrigerant circuit would also result in a change of the MSS line. This adopted margin result in higher SH set-point and consequently the loss of system efficiency.

The second one and adopted, monitors the evolution of SuperHeat along the time and changes the SH set-point accordingly to its stability. Thus there is no need of determining parameters for each pump produced or adoption of a safety margin, so that the SuperHeat is maintained at a level which confers some instability when it crosses the line with MSS, however enables a lower SH value and a more efficient operation for the heat pump. The algorithm will be illustrated in detail in chapter 4, sub-section 4.1.3.

3.2.1. Parameter determination

According to reference [22], the gain and time constant are variable parameters that depend on the operating conditions of the heat pump, particularly in the evaporation temperature and the inlet temperature of the water in the condenser, as shown in figure 3.6.

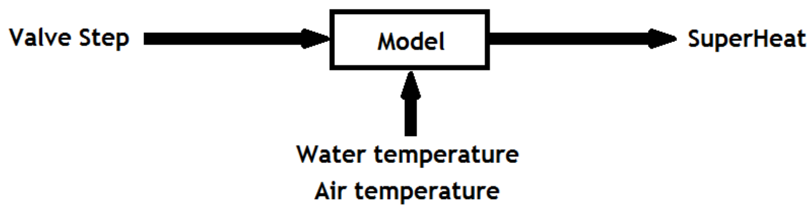

Figure 3.6 High level model diagram

Assuming the above, an assay was made in which the variation in the fan speed and the water pump would produce the same effect that is derived from the variation of air and water temperature. A software version that allows the manual setting of the expansion valve was flashed in the CPU, this means that it was possible to manually choose a value from 0 to 500 in which the EXV should remain. Variations in valve's position were defined and the corresponding change of SuperHeat was analyzed.

During this first test, it was found that whatever the conditions applied, SuperHeat was above the MSS values only when the valve position was lower than 80 steps. This indicates that the valve was oversized, which would cause less refined control by the waste of partial valve's aperture range.

After a new sizing done, it was determined that an EXV with maximum flow of 1.3 L/min should be used in that refrigerant circuit. The expansion valve was replaced for this new one and the tests were repeated. The essay was processed and the delay, time constant and gain for each test were determined.

According to the results, some conclusions were drawn:

- The aperture of the expansion valve has to be reduced to maintain the same SuperHeat value as the evaporation temperature is decreased. This is explained by the reduction of energy exchanged with the air that will slow down the process of evaporation inside the evaporator for the same flow of coolant, so the mass of fluid inside the heat exchanger has to be reduced.
- The same happens when the water temperature increases but, in this case is due to the growth of the difference between the condenser and the evaporator temperatures that also increases the refrigerant fluid pressure which changes the flow. In this case, more mass of refrigerant passes through the evaporator if the aperture of the EXV is maintained.
- If the valve is in a defined step with stable SuperHeat, even if the operating conditions are maintained, a temporarily change in its aperture makes the SuperHeat value slightly different from the last one verified in that step on a random way.
- The delay in SuperHeat response to step changes increases randomly when MSS line is nearby.
- The SuperHeat value suffers from an offset transformation when operation condition changes for the same step in the EXV, this was measured for every tests.
- In the opposite way to what was said in reference [22], that the gain depends on the operating condition, it was verified that it actually changed among tests but it does not follow any pattern.

Looking at valve's datasheet it is verified that the flow characteristic according to the opening is not linear and therefore the determination of parameters through different ranges of steps in the valve opening would invalidate these results for gain parameter determination.

Once again the tests were repeated, but this time with a fixed change form step 60 to 80 and from 80 to 60 which represents a fall and a rise respectively in SuperHeat value.

Table 3.2 represents the calculated gain of the system according to changes in the water flow and air flow. Values are within the scale units and ranges defined of sub-section 3.1.4 and 3.1.5.

Table 3.2 Gain results

rise	*Water Pump Speed*					*fall*	*Water Pump Speed*				
	Gain	**40**	**65**	**85**	**110**		**Gain**	**40**	**65**	**85**	**110**
FAN Speed	**50**	5,5	4,2	4,1	5,7	**FAN Speed**	**50**	5,3	4,7	4,6	5,2
	80	4,6	5,3	4,5	4,4		**80**	4,8	5,1	4,9	5,0
	127	5,3	5,2	4,6	4,8		**127**	5,1	4,8	4,7	5,4
	200	4,8	5,0	5,1	4,5		**200**	5,1	4,9	4,9	5,2

The results indicate that there are no noticeable changes in gain values when operation conditions change, but in the time constant determination a tendency is reflected. The next table represents it in for each test of the essay.

Table 3.3 Time constant results

rise	*Water Pump Speed*					*fall*	*Water Pump Speed*				
	Time	**40**	**65**	**85**	**110**		**Time**	**40**	**65**	**85**	**110**
FAN Speed	**50**	43	41	36	39	**FAN Speed**	**50**	34	38	35	37
	80	42	35	29	31		**80**	30	31	27	28
	127	34	34	29	26		**127**	27	27	24	24
	200	33	29	27	24		**200**	26	23	20	19

A significant reduction of values as the water pump flow and air flow increase is present; in this case, it is also clear the decrease in the time constant for values in the fall tests when compared to the rise ones.

The resulting values for the delay time were practically constant, with an average value of 4 seconds.

The non-linearity of the aperture of the expansion valve was also tested. With fixed speed in fan and water pump, a new essay was made that consists in the evaluation of the SuperHeat value, from the minimum stable value to maximum possible by gradually closing the EXV and registering its position and corresponding SH value. Both fan and water pump were in maximum speed according to the above conclusions in order to achieve the maximum step possible before the SuperHeat value hits the MSS line. Even in this condition the maximum stable step achieved was 240 of 500.

The result is represented in the chart of figure 3.7 by the red dots and is clear the non-linearity of the SH as a function of the EXV aperture. The variation in gain, referred in [22], is probably due to the opening characteristic of the expansion valves and not for de change in the operation conditions. From the four references that support the modeling of SuperHeat, [22] is the only that refer the gain variation.

Due to the introduction of this fact in the simulation model this, and the necessity of approximating the gain for the steps above 240, the characteristic was represented by a potential curve (black line) fitting. In figure 3.7 that process is lustrated. The blue dots represent the estimated values in the same step value of the SuperHeat ones.

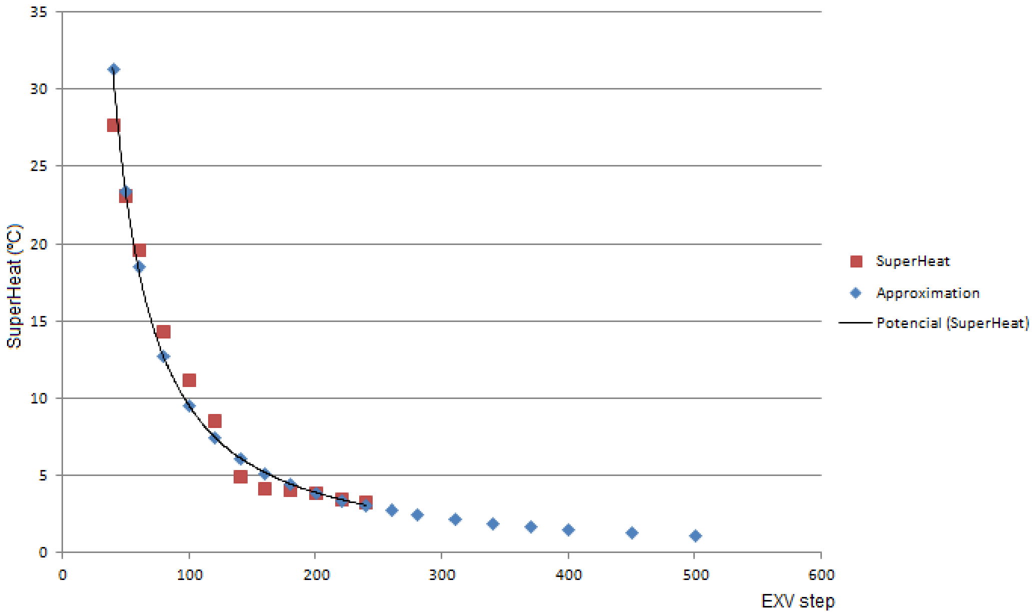

Figure 3.7 SuperHeat according to EXV step

Table 3.4 completes the chart above with the analytic values of the ones represented. In some steps the error of the estimation value is high, but according to the behavior of the existing random offset of the SuperHeat even in same operating conditions, this is not considered relevant.

Table 3.4 SuperHeat for EXV step chart values

EXV step	*SuperHeat (ºC)*	*Approximation (ºC)*	*Error (ºC)*	*Gain*
500		1,2		0,0024
450		1,4		0,0030
400		1,6		0,0039
370		1,7		0,0047
340		1,9		0,0057
310		2,2		0,0071
280		2,5		0,0089
260		2,8		0,0106
240	3,4	3,1	0,3	0,0127
220	3,6	3,4	0,2	0,0156
200	3,9	3,9	0,0	0,0194
180	4,1	4,4	-0,3	0,0247
160	4,2	5,2	-1,0	0,0324
140	5	6,2	-1,2	0,0440
120	8,6	7,5	1,1	0,0627
100	11,3	9,5	1,8	0,0953
80	14,4	12,7	1,7	0,1593
60	19,7	18,5	1,2	0,3086
50	23,2	23,5	-0,3	0,4692
40	27,8	31,3	-3,5	0,7837

On figure 3.8, the determined gain according to actual step is represented. It is given by the quotient of the aperture and the estimated SuperHeat.

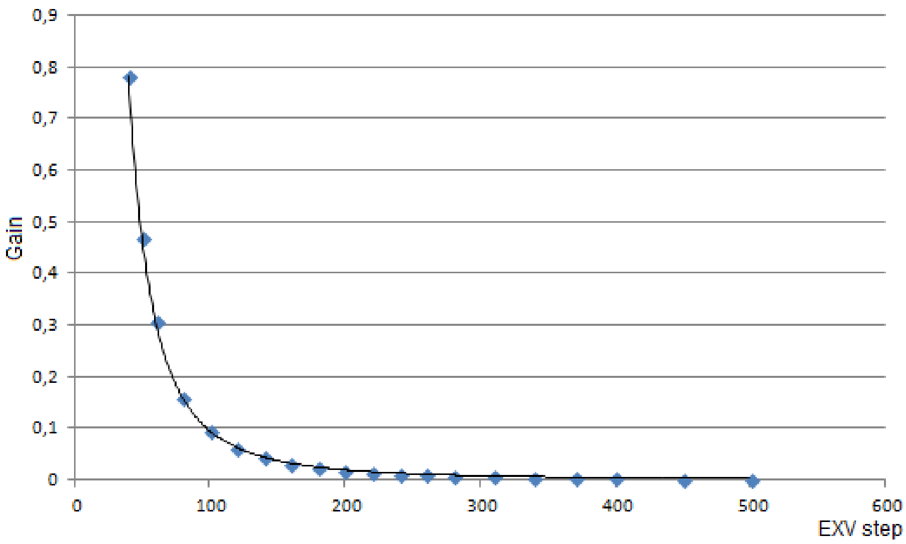

Figure 3.8 Gain for EXV step

3.2.2. The model

While determination of parameters, important system behavior conclusions were made. Accordingly, the model of the system will count on some characteristics:

- The operating conditions are given by the values of the chosen Fan Speed (FS) and for the Water Pump speed (WP);
- Time constant is approximated by the interpolation of two look up tables (LUT), one for the rising SH response and the other to the fall;
- Offset is also given by one LUT according to the two inputs for the operating conditions;
- LUT's contain the exact experimentally determined values from the essays;
- Gain is given according to the position step by the fitted potential function.

The block diagram of the model implemented in *Simulink* is in figure 3.9, the blocks are in numbered groups, and the description of each is below the figure.

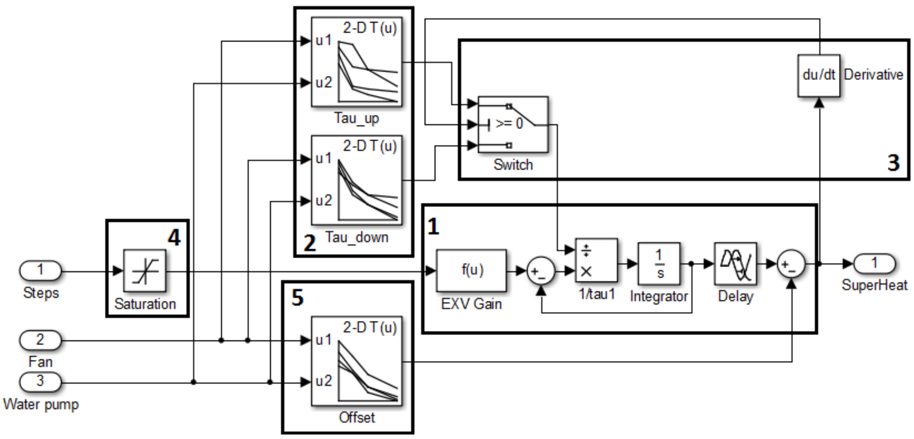

Figure 3.9 SuperHeat model

1- First order transfer function with time delay;
2- Time constant look up tables;
3- Rise or fall behavior of SuperHeat detection for switching LUT;
4- Saturation block for EXV step limitation;
5- Offset LUT.

3.3. Model Validation

Model validation was made through the comparison between the results of essay and simulation of the model using the same operation conditions and step variation. In figure 3.10 one of these tests is shown.

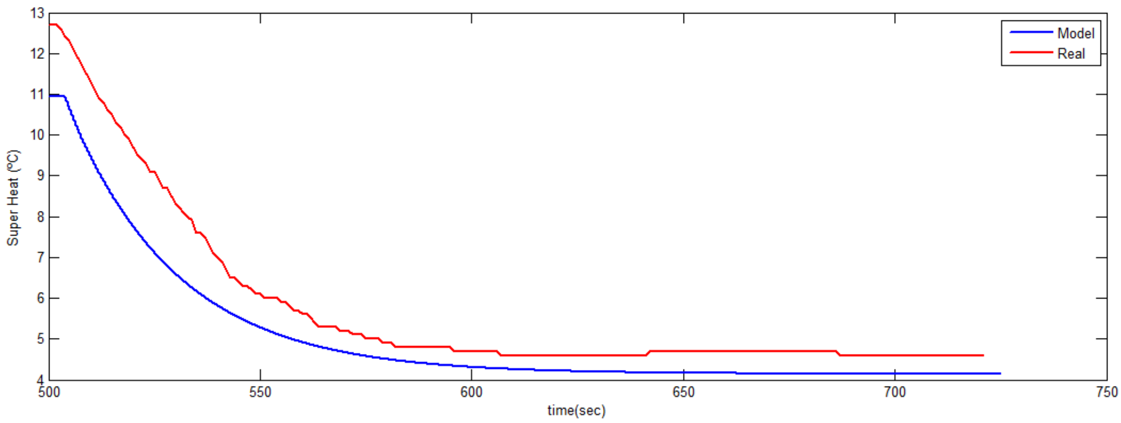

Figure 3.10 Simulation (blue) versus real (red) step response

In the situation above, fan speed was set to 200, and water pump speed to 85. The step on the valve's aperture is from 90 to 190.

From table 3.4, the step 90 (initial condition) is in a zone were theoretical values have the largest error (approximately 1.7ºC). In this essay, the measured delay was of 3 seconds, and time constant is similar from the two plots. In the final condition, the step 190 presents an error of about 0.4ºC.

In other tests, when chosen steps on potential curve approximation steps have less deviation from the real value some results are almost coincident.

From the results, it can be concluded that simulation produce results that are closer to the real conditions attending to all the random behaviors reported and to the errors from the gain approximation function.

3.4. Controller Architecture

Several considerations were made to the choice of controller type. Functional constraints such as limited processing capacity and memory storage are present. Allied to that, the easiness of adaptation for new systems has to be guaranteed.

"Fuzzy like" solutions are proper due to structural simplicity that makes the adaptation for new heat pumps possible only by few adjustments, like the change of gains and the range for inputs if the SuperHeat response changes significantly.

Also, the ECU memory and processor limitations increase the necessity in the simplification of the controller. Fuzzy like controller, according to inference system, can be expressed by a Look up table, whose size can be adjusted according available memory, and an adequate interpolation method, that complexity can be adjusted meeting, the processor speed limitations.

As in conventional controllers, there are variants of 2 terms (PD or PI) and 3 terms (PID) in “Fuzzy like” controllers.

Fuzzy PD control is known to be less used than PI because it is more difficult to remove the steady state error. However, PI type control usually gives poor performance in transient response for higher order process due to the internal integration operation [40].

Fuzzy like PID controller needs three inputs, it makes the rule base larger which is a drawback in memory issues. Also, the performance over Fuzzy PI is not much improved because of the small influence of the acceleration error in general.

With a little change in the structure, the performance and memory usage of the “Fuzzy like PID” can be improved by the avoidance of using the acceleration of error input. This is a hybrid velocity/position Fuzzy PID.

Hybrid Fuzzy Logic PID

This kind of controller uses the combination of a velocity type PI and a position type PD.

Fuzzy PD controller generates control output from error and change in error and PI generates an incremental output from the same inputs. Figure 3.11 represents by block diagram the proposed controller.

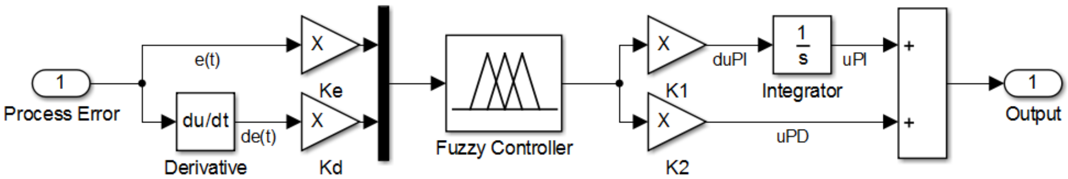

Figure 3.11 Hybrid Fuzzy Logic PID schematic

The output is given by:

$$U_k^{PID} = U_k^{PI} + U_k^{PD} \tag{3.4.1}$$

Where:

$$U_k^{PI} = U_{k-1}^{PI} + \Delta U_k^{PI} \tag{3.4.2}$$

$$\Delta U_k^{PI} = K_1 e_k + K_p \Delta e_k \tag{3.4.3}$$

$$U_k^{PD} = K_p e_k + K_d \Delta e_k \tag{3.4.4}$$

The gains can be expressed as:

$$K_p = K_1 F\{K_d\} + K_2 F\{K_e\} \tag{3.4.5}$$

$$K_1 = K_1 F\{K_e\} \tag{3.4.6}$$

$$K_d = K_2 F\{K_d\} \tag{3.4.7}$$

3.5. Controller Simulation

Two stages of the controller simulation are presented. In the first, controller is considered without characteristic or implementation constraints. The second considers every identified restriction at the implementation for the evaluation of the introduced drawbacks.

In both, the water pump speed is increasing from maximum to minimum speed to introduce the tank heating up condition. Fan speed is fixed in 127 which correspond to actual heat pump working mode.

As in the real operation, initial valve position is defined to half aperture (250 steps).

3.5.1. High Level Simulation

The high level simulation uses only the process model and the controller structure that is achieved by linear block representation and the use of the *Fuzzy Logic Controller* from the *Fuzzy Logic Toolbox*. It has also added a gain scheduling function to the original structure called EXV that actually is the inverse function of the gain to linearize the aperture of the expansion valve. Figure 3.12 represents the model used for simulation.

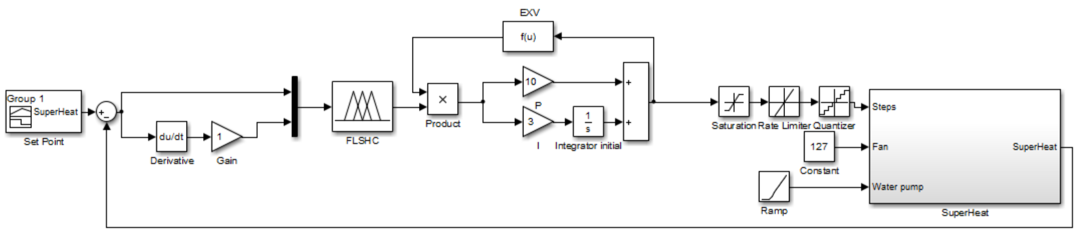

Figure 3.12 High level controller

After the tuning of parameters and adjustments in membership function shape and values, the best compromise between tracking speed and overshooting is represented in figure 3.13. The set point values are changed every 1000 seconds. Initially, tracking point is set to 7ºC, after the 1000 seconds it changes to 3ºC. When simulation reaches 2000 seconds, the set-point is changed to 13ºC. The last step is on 3000 seconds and SP is defined to 4ºC.

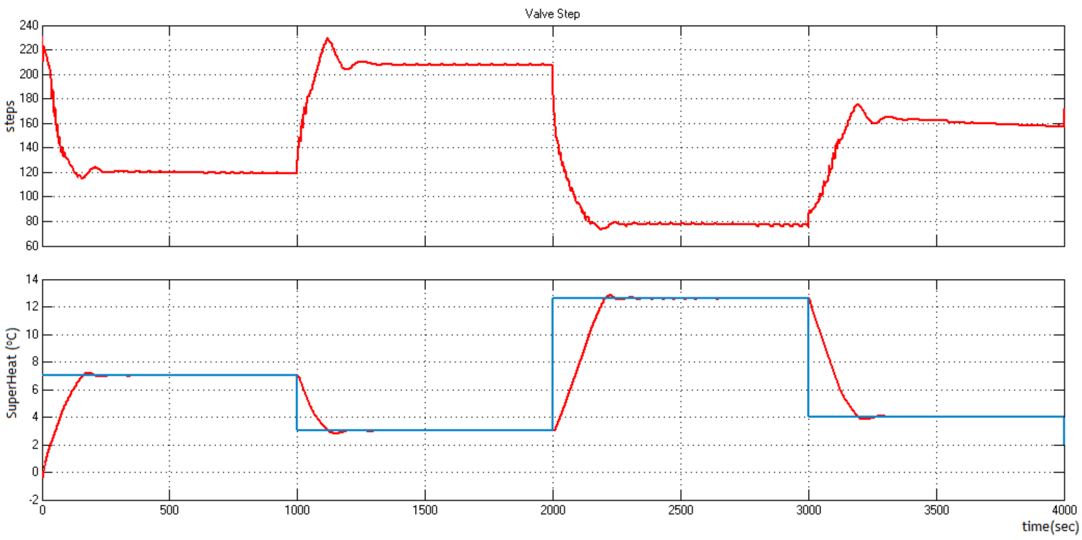

Figure 3.13 High level simulation

The maximum overshoot is of 0.2ºC, and in a step change of 10ºC, the controller takes 250s until stabilization.

3.5.2. Implementation Level Simulation

This level of simulation considers all the constraints derived by the system characteristics. In this, instead of the block of the *Fuzzy Logic Toolbox*, a look up table of two inputs and one output is used. The two look up entries have 20 steps for each. Linear interpolation method is used, as in the real controller.

The EXV gain scheduling is also given by a LUT instead of the potential function.

Sensor resolution in the CPU acquisition, period of the controller cycle, LUT's and derivative method are the same used in implementation. The block diagram of this simulation is represented in the next figure.

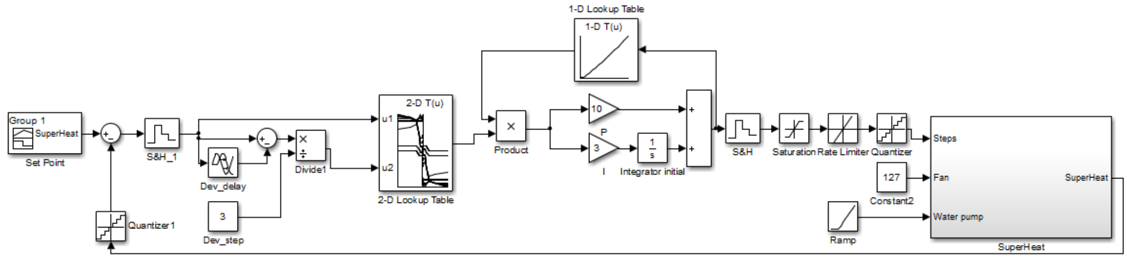

Figure 3.14 Implementation level controller

In the simulation of figure 3.15, the procedure is the same of the above. There were not also gain changes.

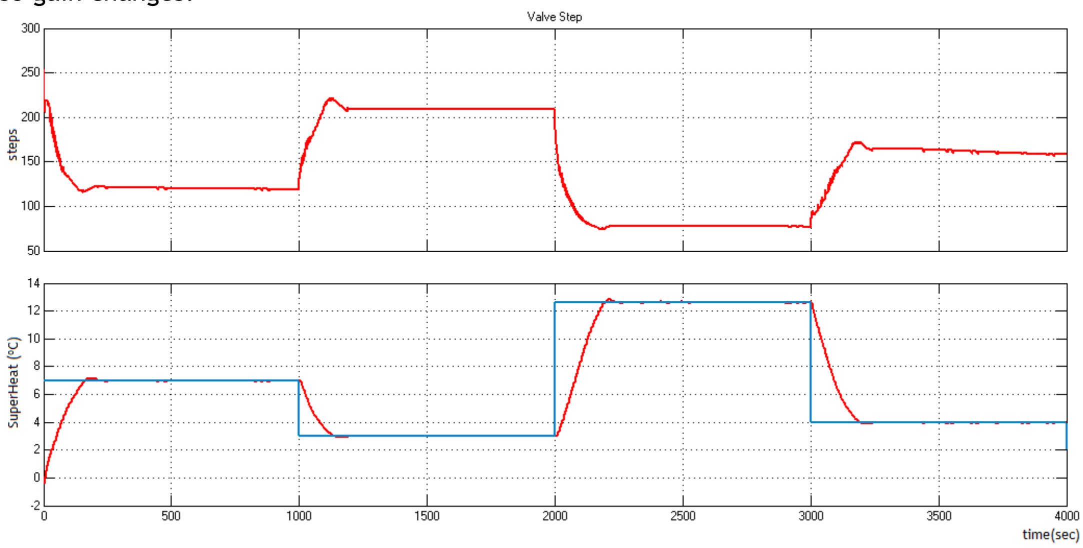

Figure 3.15 Implementation level simulation

3.6. Resume

SuperHeat behavior, above the threshold where the influence of the MSS is not significant, can be approximated by a First Order Transfer Function plus Time Delay with variable time constant. To describe the non-linearity of the expansion valve, a gain scheduling block from the electronic expansion valve has to be introduced.

According to the random behaviors of superheat, there is a sufficient level of correlation between the designed model and real system.

Due to the simplicity of the architecture, easiness of adaptation for new heat pumps with different parameters, and known good performance on systems described by FOTFTD, “fuzzy like PID” topology has been adopted.

Besides the introduction of the system constraints, the controller still able to tracking set-point in accurate way without noticeable performance decrease from the ideal model. The maximum overshoot was of 0.2ºC, and in a step change of 10ºC, the controller taken about 250s until stabilization. These results induce that this solution should be capable of maintain SuperHeat close to set point, with sufficient accuracy, when implemented in the hardware of the heat pump.

Chapter 4

Implementation and Results

This fourth chapter describes how the solution is implemented in the real system for testing and the final results achieved through the developed project are shown.

The fuzzy logic controller response in transient, start up and steady set-point operation are presented. According to the results, elations are made. The behavior of the Minimum Stable SuperHeat tracking is also stated.

A comparison with a similar heat pump which the only difference is in the expansion valve that is a thermostatic one is presented. SuperHeat controller efficiency is compared as well as the heating performance of both heat pumps.

4.1. Implementation

This section is divided in three sub-sections representing the actuation method of the expansion valve, the fuzzy logic controller and lastly the Minimum Stable SuperHeat tracking procedure.

The algorithms are implemented in C++, and the rest of the code already made for the heat pump control suffered only little modifications for the integration of the new expansion valve and the two new sensors. Debugging was made through the presentation of the interest variables directly on the HMI. Initially inputs were given through potentiometers connected instead of the NTC temperature sensors and at a later stage during the actual operation of the heat-pump.

4.1.1. Expansion Valve Actuation

The actuation of the expansion valve is achieved through the activation or deactivation of four outputs in the Electronic Control Unit. As presented on sub-section 3.1.3, the stepper motor has four built-in coils, and through the correct sequence of each one's excitation the valve's orifice is opened or closed. Table 4.1 is given by the manufacturer and shows the correct sequence of steps for movement.

Table 4.1 Coil activation order

Coil	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*
Φ1	ON	ON	OFF	OFF	OFF	OFF	OFF	ON
Φ2	OFF	ON	ON	ON	OFF	OFF	OFF	OFF
Φ3	OFF	OFF	OFF	ON	ON	ON	OFF	OFF
Φ4	OFF	OFF	OFF	OFF	OFF	ON	ON	ON

To open the expansion valve, the activation of steps is made from the number 1 to 8. To close, the inverse sequence must be taken.

In order to know the current step of the valve, an initialization procedure should be adopted. Every time the compressor starts, the valve completely closes through the application of 560 pulses in sequence from the number 8 to 1 and then the used variable for the position is reset. At this point, for opening, the sequence should start from the step 1 and every step applied should increment or decrement in one unit the position according to opening or closing operation.

When the controller orders a new position to the valve, it is calculated the difference between the actual step and the ordered. The Electronic Control Unit starts to give pulses in the correct order until the difference is null.

Valve steps are limited for maximum and minimum values; beyond those the valve is no longer activated.

Due to a mechanical break on the valve, when it doesn't need to be moved, no excitation on the coils are applied, which allows to reduce the power consumption.

The control task has a constant period of 8ms, so the maximum number of step changes for one second is 125.

4.1.2. Fuzzy Logic Controller

Fuzzy logic controller of figure 4.1 is the one responsible for tracking the set point, and it was previously simulated as seen in chapter 3. The algorithm runs every 250ms and is now briefly explained.

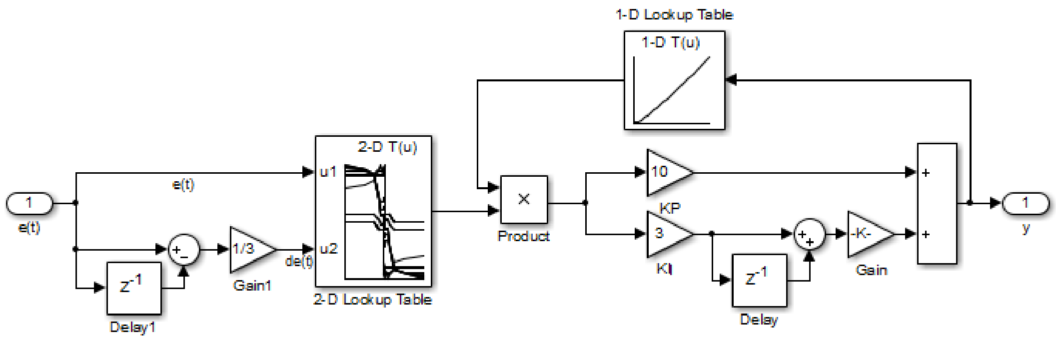

Figure 4.1 Implemented fuzzy logic controller

1- The Temperature of the two NTC sensors is given by pre-made functions that linearly interpolate the value of read voltage with the manufacturer's established constants. In order to use only integer variables, to allocate less memory, the value is already multiplied by 10

2- The difference between the evaporator outlet and inlet temperatures represents the SuperHeat. Signal e(t) is now determined through the deviation of the SuperHeat from the set-point that is also multiplied by 10.
3- The error rate de(t) is calculated by the difference between samples taken every three seconds and divided by 3.
4- The two Look-Up Tables present in simulation have outputs in order of the decimal, so those were copied and all the values were multiplied by 10.
5- The value of the actual EXV position is taken as input for the same function that interpolates the NTC values, but in this case is used on the EXV gain LUT. The output is multiplied by both KI and KP constants.
6- A bilinear interpolation function was created and interpolates the Fuzzy LUT. The values of the two inputs error and error rate are limited to the boundaries of the look up table.
7- The result of step 6 is multiplied by the new KI from step 5 and integrated through the multiplication with 0.25 due to the period of the control task.
8- The result of step 6 is multiplied by the new KP from step 5 and summed with the result from step 7.
9- The last step is the division with the scale factor that results from every multiplication.

4.1.3. Minimum Stable SuperHeat Algorithm

As previously referenced in section 3.2, set-point tracking is made trough Minimum Stable SuperHeat.

The MSS algorithm can be divided into two parts. The first senses the system instability, by measuring the amplitude of the oscillations of SuperHeat around the set point. If the oscillations are above a certain threshold, this means that the set-point is close to the MSS, then a status flag will be activated until the stability condition is achieved.

The second refers a state machine that according to the system condition, changes the set-point having in consideration the system stability and the proximity to the MSS line.

System Instability Detection

Instability condition is defined through the detection of the inflection points in the signal e(t). In the schematic representation of figure 4.2, the procedure is easy to understand.

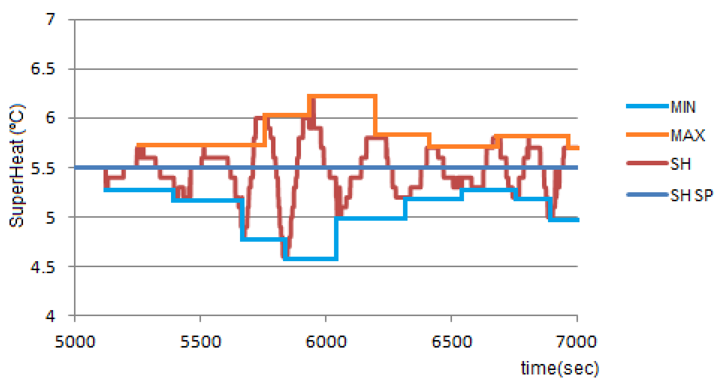

Figure 4.2 Instability determination overview

When e(t) crosses the SH SP (SuperHeat Set-Point) line, it starts the detection to the biggest absolute value until the line is crossed again. The minimum (light blue) is detected when the error is below zero and the maximum (orange) when it is above.

In situations like in the figure 4.3, if only the procedure above was made, the orange line would be like the dashed one, which doesn't make sense. So, a new condition was adopted: If the maximum or minimum value maintain unaltered for more than X time, and if the error change within Y time is too small, then the max value is null.

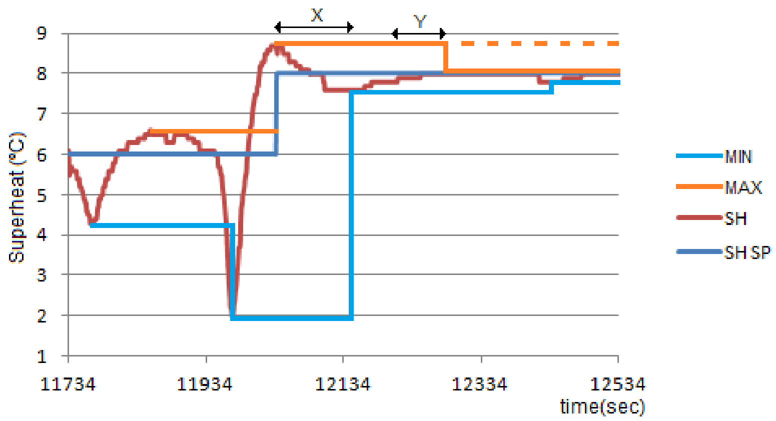

Figure 4.3 Instability detection (particular case zoomed in)

Set Point Control

A state machine works directly on the control of the SP value for the SuperHeat in order to keep it close to the minimum stable.

Whenever the compressor is started, the smaller the flow resistance of the fluid in the refrigerant circuit, the lower the inertia to overcome and therefore the lower the consumed power. To avoid current peaks at startup, expansion valve is set to a large aperture position and remains so for some time in order to make the system reach some stable operation.

Only at this point the SuperHeat control is activated and the state machine starts. When the heat pump starts, SH is set to a value that should be stable.

- State INIT: A fixed value for the last set-point is maintained until the stability condition is reached. At this time, the next state is 1. If the system spend time more than sufficient to stabilize the SH and it remains unstable, the next state will be 2.

- State 1: The value of the set point is gradually reduced until it is determined the condition of instability. At this point, the state set point is overlapped with the line of minimum stable SuperHeat, the next state is 2.

- State 2: If instability is detected, the machine immediately goes to state 3. If it stays a large period in this state, it can mean that the line of MSS can be far, so that a new search should be made, and that the state is changed to 1.

- State 3: Is made a high increase in the set point value, and waits until the SuperHeat returns to stability. As in the INIT state, if spend more time than the sufficient to stabilize the SH and the system remains unstable, the next state will be 2. Otherwise, the set point value will be gradually decremented whose total value of the decrease is only a fraction of the initial increment. In this last process if the instability is reached, the state is changed immediately to 2.

- In any of these states, if the water temperature in the bottom of the tank decreases by more than a certain limit for a given tapping, it can indicate that the value of the MSS changed. Then the state is changed to INIT, in order to the process restarts the tracking of the line MSS again.

The diagram of figure 4.4 with the table 4.2 illustrates expeditiously the above process.

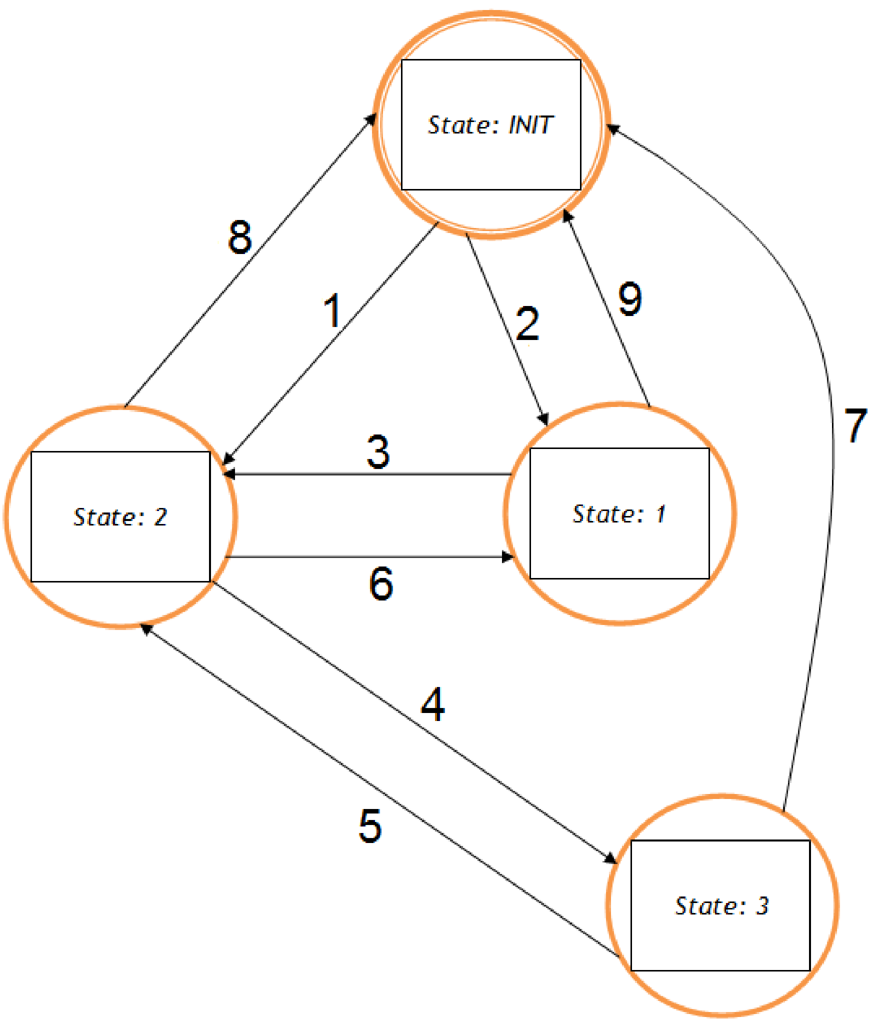

Figure 4.4 State machine diagram

Table 4.2 State and transition description

Number	*Transition Description*
2	SuperHeat is stable
5	SuperHeat is unstable or State ended the task
3,4	SuperHeat is unstable
1,6	Large time on state
7,8,9	Bottom tank temperature decreased
Number	**Action Description**
INIT	Controller starts
1	Gradually decreases SuperHeat set point
2	Do nothing
3	Increases set-point and waits enough time to stabilize, then gradually decreases until a fraction of the initial increment

4.2. Fuzzy Logic Controller Analysis

For testing the controller's start-up, transient and steady operation, a software version without the MSS tracking algorithm was loaded. It allowed changing the set point manually. During the test, the ambient temperature is approximately constant but water pump flow and fan speed were changed as indicator of the robustness of the controller.

The first condition verified is the start up (figure 4.5). Controller's constants, KP and KI, tuning was made also according to this to reach a good compromise between overshoot and settling speed to the set point. To reduce even more the overshoot in this case, the gains should be much lowered what would influence the rest of the operation in the heat pump because this is the only case when super seat value departs from a value near zero, so it will cross the whole unstable zone which changes radically the dynamics of the system.

In figure, the green line represents the position of the EXV, in steps. Even before the controller starts, the compressor is already working and the EXV is steady with large aperture, so the SuperHeat, represented by the red line, is below 0ºC. The controller starts at the moment the variation in the EXV is verified.

In the presented start up, the overshoot was of approximately 6 degrees, and followed by undershoot of only 1 degree. After approximately three minutes since the controller starts, the SuperHeat is on the set-point (blue line).

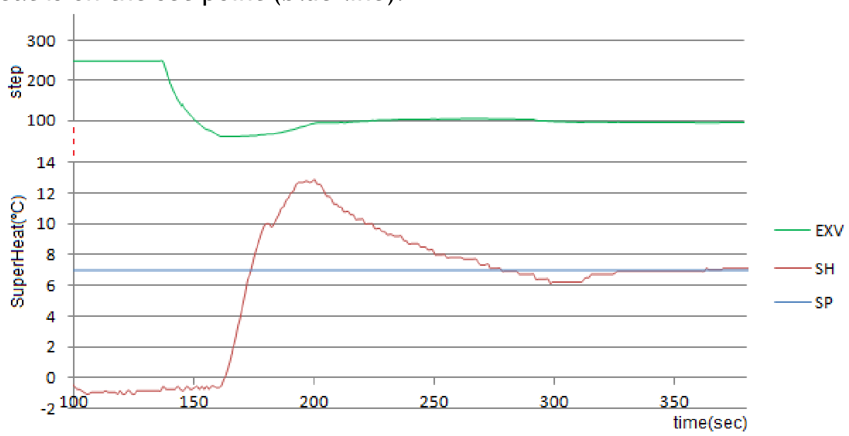

Figure 4.5 Start up condition

Steady operation is shown on figure 4.6. SuperHeat and Set-Point are represented through the colors red and green respectively. Due to the sensibility of the temperature input, only variations of tens of degree are evident. It was not expected to achieve a perfect line for SuperHeat shape because the phase change of a fluid has a randomly unstable behavior [9], so oscillations, even with steady operation conditions actuators, will happen.

It is found low resolution of the EXV when it is with little aperture (blue line). Only one step causes a variation in SuperHeat of more than it should. In the right side of the chart it is seen that the valve tries to settle in a position that makes SH in the desired value but that intermediate step does not exist so the controller is alternating between two positions. This induces that the EXV continues oversized.

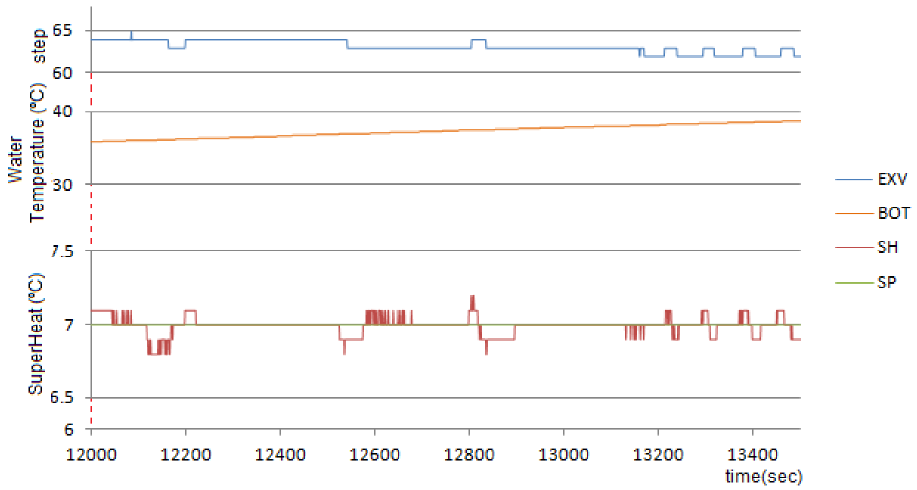

Figure 4.6 Steady state operation

The orange line is the temperature of the water in the bottom of the tank. In the 25 minutes represented in the chart, it increased about three degrees, and is possible to see in EXV position a sudden decrease of the average value. Figure 4.7 has large time range(about three and half hours) and with that, the above statement is more clear to check.

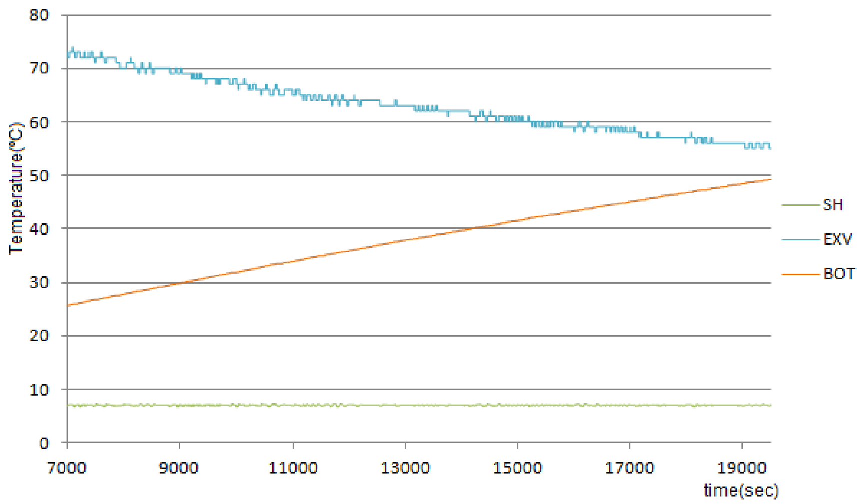

Figure 4.7 Steady state operation (zoomed out)

The assumption of sub section 3.2.1 that the water temperature rise makes the expansion valve to work on more close position in order to maintain the SuperHeat value is once again experimentally verified.

The next two charts are representative of the transient tracking. In the first one (figure 4.8), the set point (blue line) is initially in 6 degrees and no undershooting of SH (red line) is verified, a set point change happens, to five degrees, but this time an undershoot of about 0.3 degrees is verified. As the set-point is getting closer to de minimum stable, the influence of the MSS is more evident and the undershooting is getting higher, as there was an attraction force bellow. Even with this condition, the controller was able to stabilize SuperHeat in five degrees after about 180 seconds.

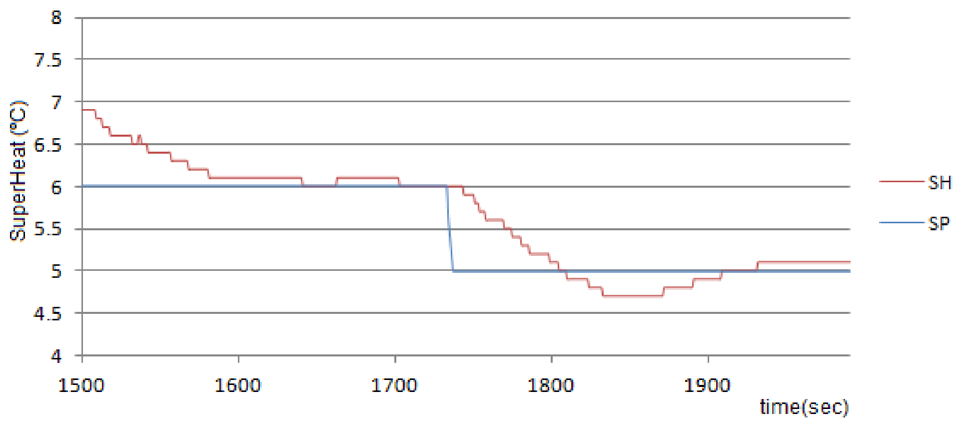

Figure 4.8 Response to set-point fall

The second chart (figure 4.9) is from a set point rise when SuperHeat was unstable. Set-point changed from 4.5ºC to 6.5ºC. In this condition, a bit of the presented effect of the SuperHeat instability zone is felt, and overshoot of about 0.6ºC. The controller managed to stabilize SuperHeat after about 140 seconds of the set-point change.

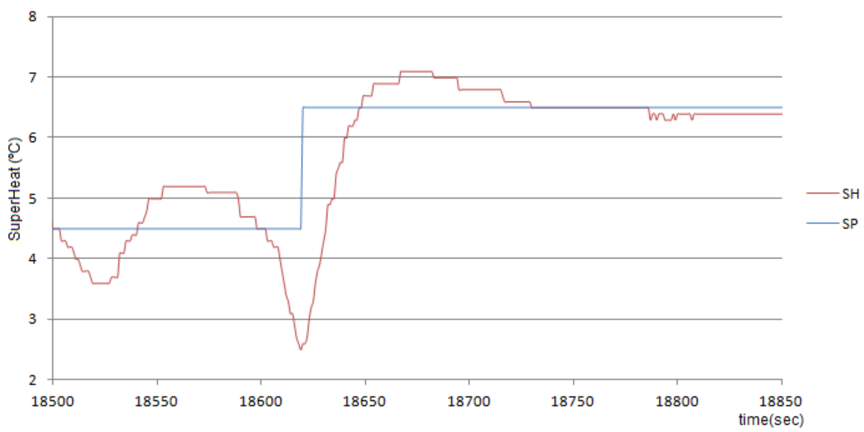

Figure 4.9 Response to se-point rise (while unstable)

4.3. Minimum Stable SuperHeat Algorithm Analysis

The MSS algorithm is based on three states during the heat pump operation, except if the temperature of the water in the bottom of the tank falls more than 5°C. In Figure 4.10, those are represented. The red line represents the current value of SuperHeat and the blue one the set point along the time.

At the beginning of the time scale, it is found that SH is about to enter a state of instability. At this time, state 2 is active. Due to the instability the algorithm immediately passes the active state to 3 and an abrupt increment in set-point happens.

Sufficient time passes until the stabilization, when the system is given as stable set point starts to decrease slowly. When the decrement is completely set next state is 2 however, the system is with a little of instability, but the amplitude in oscillation is not sufficient to consider the system unstable, so the value of the set point is maintained.

After a long period with a constant value of set-point, the algorithm returns to track the MSS switching from state 2 to state 1 and begins to decrement the SP. When in instability set-point is increased again and, after stabilization, returns to decrement a little bit less than the previous increment. This time, in opposite, when set-point stops decreasing, the system is not stable so the next state will be the number 2 but will pass directly to 3 and the procedure is repeated.

Lowering the temperature of water in tank, as well as the restart of the compressor was tested and the algorithm returned to the initial state as desired.

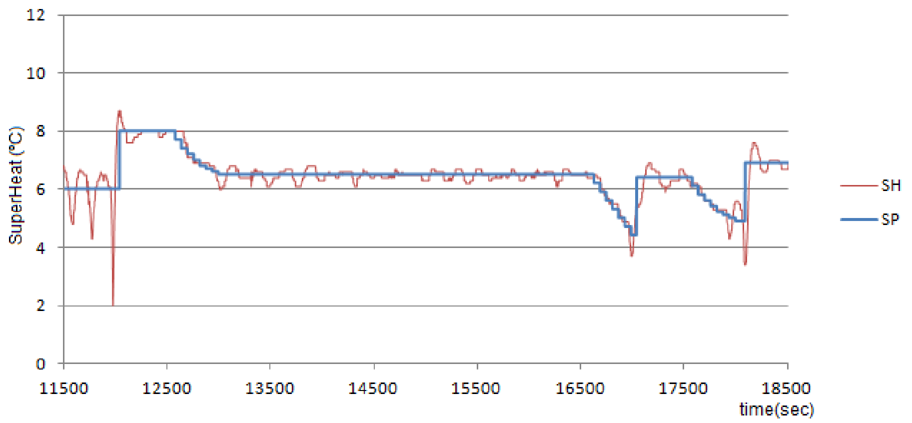

Figure 4.10 Minimum Stable SuperHeat tracking

4.4. TXV versus EXV on SuperHeat Control

The SuperHeat controller ability is now compared between the current solution and the developed one. In the figure 4.11 bellow, it is presented the SuperHeat of the system with a TXV on blue and with EXV with the projected controller in red during the first heating up performed on the COP test.

One of the inputs of TXV is the pressure that has almost negligible time constant, so the actuation is faster to sense fluctuations and that allows the TXV working in less degree of SuperHeat. Although, using this lower set-point, introduces constant oscillations, with high frequency, that are a big drawback in efficiency and promote the wear of the compressor.

The higher set-point operation reduces performance, but in the opposite way, there is an increase due to SuperHeat stability. So the EXV combines system stability and less wear of the remaining elements of the heat pump

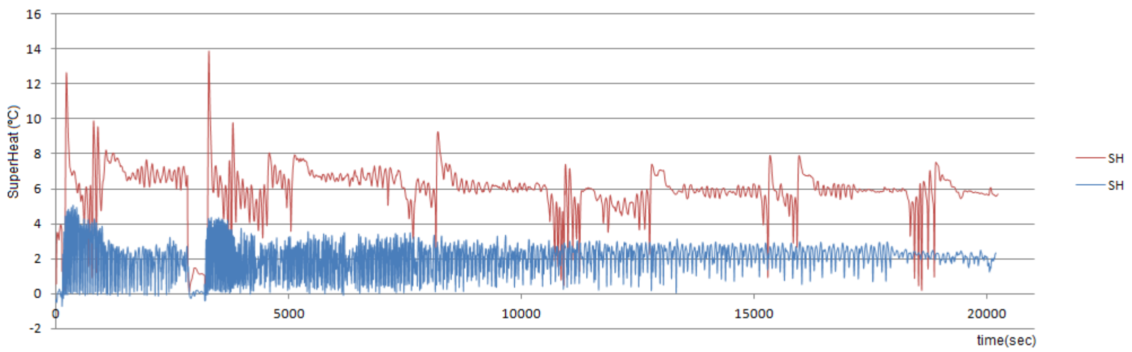

Figure 4.11 SuperHeat behavior during total heating up (TXV on blue, EXV on red)

4.5. Coefficient of Performance

COP test, according to standard EN 16147, on the final version of the project was conducted and relevant results are now presented. With this procedure, not only the performance index but also other parameters such as the total time or energy consumed during the heating are determined. As seen in sub-section 2.1.6 all operating conditions are the same for each test, and maintained with small admissible deviations. With these indicators, it is possible to assess the gains that the changes made to the heat pump allow, for comparison of results from previously performed tests.

Figure 4.12 below represents two curves of a complete heating of the water inside the tank, the red is the result of the project's heat pump, and the blue of a similar heat pump with TXV, the same used for SuperHeat behavior comparing on the earlier section.

In the conditions of this test, when water temperature reaches about 17ºC, the evaporator is frozen, so the heat pump switches to defrost cycle. At this time, the water pump is turned off. Due to the position of the water temperature sensor, readings suddenly decrease until heat pump returns to the normal heating mode.

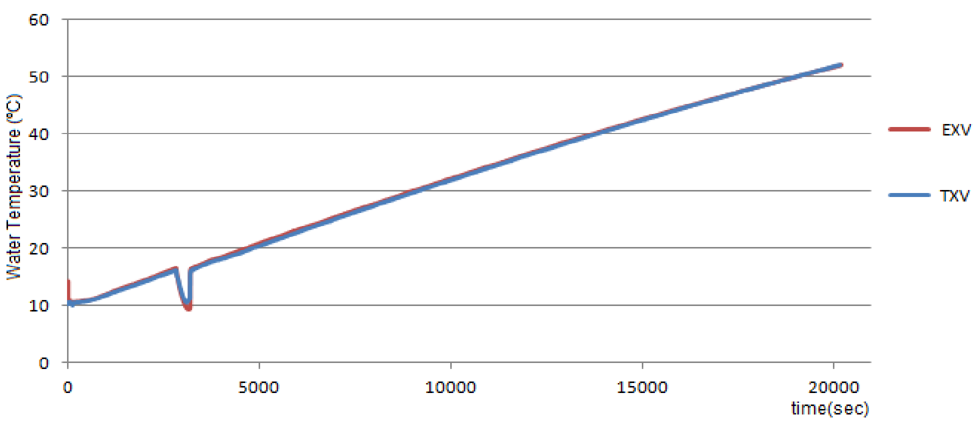

Figure 4.12 Water temperature on heating up (TXV on blue, EXV on red)

The heating up curves are practically superposed, and therefore the spent times for each solution are similar. Only a small difference exists, being this new solution faster in 2:30 min.

The energy consumption, in total duration of test, was also reduced, but this time significantly, compared to the TXV solution. In figure 4.13 both electric power consumption profiles are traced and the overall power consumption decrease is evident. During the test, savings reached 2.31%.

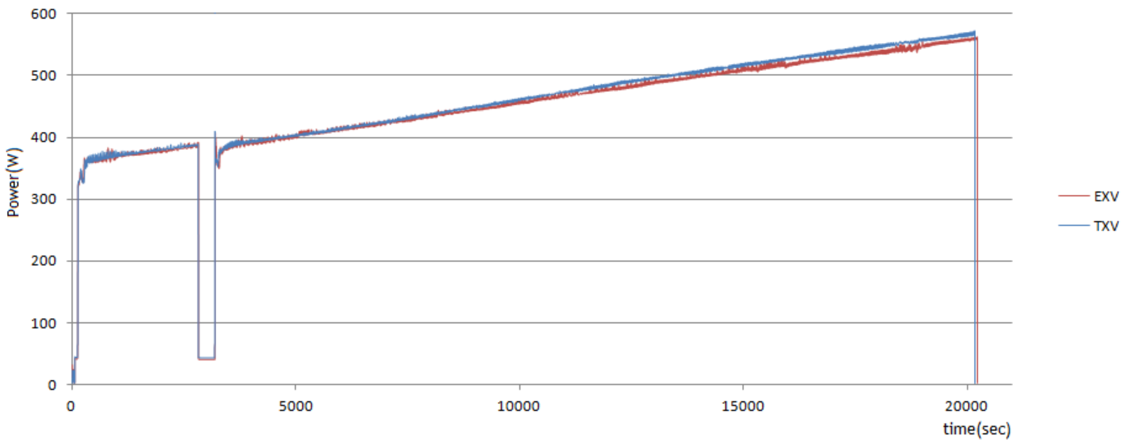

Figure 4.13 Electric power consumption on heating up (TXV on blue, EXV on red)

The maintained performance even with the reduced consumption represents efficient operation which allows an increase in Coefficient of Performance value of 2.14%.

4.6. Resume

Through the results, it is possible to conclude that every stage of the controller is implemented successfully. It was tested in a variety of different conditions in which the start up, transient and steady operation were verified with good results and according with the simulations made.

The application of the MSS algorithm makes the average value of SuperHeat higher, although there is a much significant reduction in the oscillation amplitude and frequency. With this, reduction on compressor wear as well as more stable system operation is achieved.

After realization of the COP test, it is verified that the performance is similar to the thermostatic expansion valve solution, but efficiency is increased by the reduction of power consumption. This leads to the expected rise in the Coefficient of Performance indicator.

Chapter 5

Conclusions

5.1. Epilogue

In this dissertation a solution to the control of an Electronic Expansion Valve integrated on a Heat Pump was designed and implemented. Design process is described and results on simulation and real operation are shown.

Initially a search was done on the main issues addressed throughout the project. The same fell on matters related to the heat pump and control of processes. An introduction to the operation of a heat pump, different types and existing components that constitute it are presented. The focus is oriented ASHPWH due to the type of pump where the controller will be implemented.

Incorporating an EXV is intended for more effective control of the specific phenomenon called SuperHeat, so a study thorough the influence variables and its behavior was made. It was determined that its response is highly nonlinear and there is a minimum threshold which must not be exceeded due to system instability and consequent loss of efficiency. The amount of SuperHeat should be kept as low as possible but ensuring stability.

Solutions already on the market and other studies developed and published articles were evaluated as well as components used for sensing and actuation. It appears that thermostatic expansion valves are still being used on a large scale because it is a cheap solution and operating proven, however there are several negative effects when compared with electronic expansion valves. For a more efficient and dynamic control of the SuperHeat EXV should be used and, in systems with reduced capacity, the stepper motor electronic valve should be chosen because it prevents excessive pulse and promotes efficiency by less energy consumption compared with the PWM.

Two common ways are used for measuring the SuperHeat; through one pressure sensor and one temperature sensor or using two temperature sensors. The great majority of solutions use the first option, however, in attempt to reduce costs, it was given preference to the first and the investment is made in the control algorithm towards gain advantage at this point.

The initial study also passed by the controller logic. Due to the nonlinearity of the process a suitable nonlinear controller should be used. Restrictions in terms of processing speed and memory led to controllers based on fuzzy logic of the “Fuzzy Like” type, by the structural and implementation simplicity, to be adopted. The delay in Superheat’s step response and the necessity for a follow-up set point with a low steady state error were decisive to the choice of a controller with the components proportional, derivative and integrative.

By the complexity that involves a model able to describe the operation of a heat pump, just to collect a state variable, it was used an approximation by a first order transfer function with time delay that describes the behavior of SuperHeat by varying the flow passing through the expansion valve. Experimentally it was found that the gain and delay of the transfer function is nearly constant in different operating conditions of the pump, but the time constant is variable. Due to the nonlinearity of the expansion valve opening when its value is given by the step position and not by the maximum mass flow rate, a variable gain should be considered according to the position of the valve.

Upon experimental determination of the parameters, the model was built with all identified system constraints and nonlinearities. It was verified enough correlation and so, able for being used in the simulation and tuning the controller.

In simulation, the adopted topology proves to be adequate, with a quick response considering the time constants involved and minimum error in steady-state and also low overshoot.

Once deployed, and real tested, the controller continued to verify the effectiveness of set point tracking. However, as expected in situations where the SuperHeat is lower than the minimum stable it is impossible for any set-point stabilization.

The SuperHeat minimum threshold tracking algorithm was effective, keeping a low set point, without compromising system stability.

After realization of the Coefficient of Performance tests, comparing the results with a similar heat pump which the only difference is the use thermostatic expansion device, it was verified that the performance is identical but with more efficient operation, with the reduction in 2.3% in electricity consumption which contributed to an increase of 2.1% in the value of COP in respect to the previous solution.

Looking at logged data of both during the operation, with the EXV there is an increase in the absolute value of SuperHeat, but with a significant reduction in oscillation, which contributes to system stability and durability of the heat pump components.

Strategies presented by some manufacturers for this type of components and solutions announce higher efficiency gains with the use of electronic expansion valves. To heat pumps with similar capacities as the employed one, only the energy spent in the operation of the expansion valve represents an increase of approximately 2% in the system energy consumption. Significant gains should only be observed in solutions which the capacity involved is higher so the valve actuation losses will not be significant.

5.2. Future Work

There are still work that can be done in order to improve the use of the electronic expansion valve. Some suggestions are now presented:

- Reduce the expansion valve dimension for refined control and less power consumption during operation. Even with the change from the 1.8l/m to 1.3l/m the EXV is still oversized.

- Exhaustive Fuzzy PID parameters tuning for better transient set-point tracking which can promote the reduction of stabilization times in the Minimum Stable SuperHeat algorithm.

- Exhaustive testing of MSS algorithm parameters in order to achieve better performance by lowering SuperHeat set-point, but maintaining stability. Also adopt solution that reduces the valve actuation to decrease power consumption.

- Explore the variation in controller performance by changes in the sensors resolution and controller frequency.

- Testing the controller with a pressure transducer instead of two temperature sensors. By the reduced time constant in acquisition, improvements in control can possibly be made.

- Implementation of the SuperHeat controller in the Suction Line Heat Exchanger (SLHX) solution. This is an addition of another Heat Exchanger in the heat pump that allows the evaporator to work with less SuperHeat which increases efficiency and that cannot be controlled with the thermostatic expansion valve.

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