

Faculdade de Engenharia da Universidade do Porto
Bosch Thermotechnik GmbH



BOSCH

**EVALUATION AND DEVELOPMENT OF
LABORATORY COMPONENTS FOR
HYBRID HEAT PUMPS**

William Milner

Supervised by

Professor Carlos Manuel Coutinho Tavares de Pinho

Dr. Christian Glück

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“Sempre chegamos ao sítio aonde nos esperam”
José Saramago

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Abstract

The public's attention frequently underrates the impact of the building sector on energy consumption and CO₂ emissions. However, 38 % of the total CO₂ emissions in the United Kingdom in 2012 resulted from heating (UK Government - Department of Energy & Climate Change, 2012), and according to the German Ministry to Economy and Energy (2018) the heating industry currently represents about 35 % of the total energy consumption in Germany. Thus, in order for countries in the EU to reach the ambitious targets set by the European Commission to cut off emissions by 80 % to 95 % by 2050 in comparison to 1990 levels, the building sector's energetic paradigm must go through an extensive changing process. Heating demand must decrease and primary energy efficiency must increase. Amongst many different solutions proposed for the aforementioned challenges, lies the hybrid heat pump, an appliance equipped with a heat pump and a backup fossil fuel heat generator. This product allegedly overcomes the inefficiency entailed to air sourced heat pumps in cold weather conditions and represents an ideal combination of electricity and fossil fuels for optimal primary energy efficiency. However, its feasibility still lacks proof data. Being a major player in the residential heating market, Bosch Thermotechnik GmbH develops this technology in Lollar, located in the middle of the German state of Hessen. The plant has existed since 1731 with the main focus on development and production of gas and oil boilers, but one of the laboratories has been extended for air-to-water heat pumps and hybrids testing. The market deployment of such appliances requires an energy label established by the European Commission for which the measurement conditions must comply with tight limits imposed by the European standard DIN EN 14511. The goal of this dissertation is to make a thorough analysis and further improvement of the laboratory so that in the end it is possible to assess its suitability to perform such releasement measurements, or rather only for internal testing for research and development purposes.

Resumo

O impacto do consumo energético associado ao aquecimento de espaços interiores e à produção de água quente para consumo sanitário é frequentemente menosprezado durante o debate sobre poluição e sustentabilidade, estando normalmente a atenção pública focada no setor dos transportes. Contudo, 38 % das emissões totais de CO₂ no Reino Unido em 2012 resultaram de geração de energia para aquecimento (UK Government - Department of Energy & Climate Change, 2012) e 35 % da energia consumida na Alemanha (Bundesministerium für Wirtschaft und Energie, 2018) derivam do mesmo setor. Os objetivos propostos pela Comissão Europeia de reduzir em 80 a 90 % as emissões de CO₂ em relação aos níveis medidos em 1990 não serão alcançados caso não se verifique uma mudança radical no paradigma de aquecimento de espaços interiores e de água para uso sanitário. Para que tal aconteça, as necessidades energéticas dos edifícios terão de ser reduzidas e a eficiência de energia primária terá de aumentar. As bombas de calor híbridas são aparelhos destinados ao aquecimento no setor doméstico e consistem na combinação de uma bomba de calor e de uma caldeira cujo funcionamento depende das condições climáticas assim como da exigência energética. Estes aparelhos têm a capacidade de contornar as ineficiências associadas a uma bomba de calor aerotérmica durante baixas temperaturas do ar atmosférico, apresentando teoricamente um equilíbrio ideal entre a utilização de eletricidade e combustíveis fósseis para atingir uma eficiência ótima de utilização de energia primária. Sendo um dos principais intervenientes no mercado do aquecimento, a Bosch Thermotechnik GmbH está a desenvolver e a analisar estes mesmos produtos em Lollar, uma das suas localizações Alemãs, cujo principal foco desde o seu aparecimento em 1731 tem sido a produção e desenvolvimento de caldeiras para uso doméstico e industrial. Contudo, um dos laboratórios existentes foi adaptado para a realização de testes em bombas de calor, nos quais se pretende medir e avaliar os parâmetros de funcionamento de bombas de calor híbridas. Um dos objetivos definidos para o laboratório consiste em torná-lo apto para realizar medições em conformidade com os limites e requisitos impostos pela norma DIN EN 14511. A realização destas medições prende-se à necessidade de um rótulo indicador da eficiência energética do aparelho imposto pela Comissão Europeia, para que este possa ser lançado no mercado. Assim, a finalidade do trabalho realizado durante este estágio e apresentado nesta dissertação é de analisar e melhorar os equipamentos destinados a testar os mencionados aparelhos e de avaliar a sua viabilidade para realizar medições em conformidade com a norma DIN EN 14511.

Zusammenfassung

Die Aufmerksamkeit der Öffentlichkeit unterschätzt häufig die Auswirkungen des Gebäudebereichs in Hinblick auf den Energieverbrauch und die CO₂-Emissionen. Trotzdem ergibt eine Studie der Regierung des Vereinigten Königreichs, dass 38 Prozent der gesamten Emissionen in Großbritannien im Jahre 2012 durch die Heizungssysteme entstanden sind. Laut des Bundesministeriums für Wirtschaft und Energie verbrauchen wir „Rund 35 Prozent unserer gesamten Endenergie in den eigenen vier Wänden, vor allem für Heizung und Warmwasser“. Die EU-Kommission schreibt eine Reduzierung der CO₂-Emissionen um 80 bis 95 Prozent im Vergleich zum Jahre 1990 vor. Um diese Ziele zu erreichen, müssen in dem Gebäudebereich viele Änderungen vorgenommen werden. Der Energiebedarf muss sinken und die Energieeffizienz muss steigen. Eine der vielen Lösungen für diese Herausforderung ist die Hybridwärmepumpe, die sich aus einer Wärmepumpe und einem Heizkessel zusammensetzt. Dieses Gerät kann die Ineffizienz der Luftwärmepumpe bei niedrigen Außentemperaturen überwinden und stellt damit eine Kombination der elektrischen und fossilen Energie dar. Als eins der führenden Unternehmen für Heiztechnik, entwickelt Bosch Thermotechnik Hybridwärmepumpen am Standort Lollar. Dieser Standort existiert seit 1731 mit dem Schwerpunkt auf Entwicklung und Produktion von Gussheizkesseln, wobei einer der Prüfstände auf die Luftwasserwärmepumpe und Hybridwärmepumpe angepasst worden ist. Die Vermarktung solcher Geräte erfordert nach EU-Rechtlinien eine Energieverbrauchskennzeichnung. Die dazu nötigen Messungen müssen der DIN EN 14511 entsprechen. Ziele dieser Arbeit sind eine Klimakammer und ein Prüfstand zu analysieren und weiter zu entwickeln, um die Konformität der Norm zu verifizieren.

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Nomenclature

<i>A</i>	Area [m ²]
<i>B</i>	Induced magnetic field [T]
<i>C</i>	Signal output
<i>CC</i>	Conversion factor
<i>c_w</i>	Specific heat of water [kJ/kg.K]
<i>COP</i>	Coefficient of Performance
<i>D</i>	Diameter [mm]
<i>e</i>	Error signal
<i>H_{CK}</i>	Numbers of hours the unit is considered to work in crankcase heater mode [h]
<i>H_{HE}</i>	Number of equivalent active mode hours for heating [h]
<i>h_j</i>	Number of bin hours occurring for T_j [h]
<i>H_{OFF}</i>	Numbers of hours the unit is considered to work in off mode [h]
<i>H_{SB}</i>	Numbers of hours the unit is considered to work in stand-by mode [h]
<i>H_{TO}</i>	Numbers of hours the unit is considered to work in thermostat-off mode [h]
<i>j</i>	Bin number
<i>k</i>	Factor correcting for geometry
<i>K_d</i>	Derivative Gain
<i>K_i</i>	Integral gain
<i>K_p</i>	Proportional gain
<i>m</i>	Mass flow rate [kg/s]
<i>n</i>	Total number of bins
<i>P_{CK}</i>	Power input during crankcase heater mode [kW]
<i>P_{designh}</i>	Design heating load the unit is suitable for as declared by the manufacturer [kW]
<i>P_{ele}</i>	Electrical power input [kW]
<i>P_h</i>	Heating load of the corresponding T_j [kW]
<i>P_{OFF}</i>	Power input during off mode [kW]
<i>P_{SB}</i>	Power input during stand-by mode [kW]
<i>P_{TO}</i>	Power input during thermostat-off mode [kW]
<i>Q̇</i>	Heat power [kW]
<i>Q_{Fossil}</i>	Fossil fuel power input for the corresponding T_j [kW]
<i>Q_H</i>	Reference annual heating demand [kW]
<i>Q_{HE}</i>	Annual energy consumption for heating [kW]
<i>RH</i>	Relative humidity [%]
<i>SCOP</i>	Seasonal Coefficient of Performance

$SCOP_{ON}$	Average coefficient of performance of the unit in active mode for the designated heating season
t	Time [h]
T_{biv}	Bivalent temperature [°C]
T_{db}	Dry bulb temperature [°C]
T_{in}	Inlet temperature [°C]
T_j	Bin temperature [°C]
T_{out}	Outlet temperature [°C]
T_{real}	Real temperature [°C]
T_{set}	Set temperature [°C]
T_{wb}	Wet bulb temperature [°C]
U	Measured voltage [V]
v	Flow velocity [m/s]
$\dot{V}$	Volume flow [m ³ /s]
ΔT	Temperature difference [°C]
η_s	Seasonal space heating energy efficiency
η_{HE}	Efficiency of the heat exchanger
ρ	Density [kg/m ³]

List of Abbreviations

CC	Climate Control
CCT	Climate Chamber Temperature
DHW	Domestic Hot Water
EC	European Commission
EU	European Union
HT	High Temperature
HVAC	Heating, Ventilation and Air Conditioning
IDU	Indoor Unit
LT	Low Temperature
MT	Medium Temperature
ODU	Outdoor Unit
PID	Proportional Integral Derivative
PLC	Programmable Logic Controller
R&D	Research and Development
TC	Temperature Control
TDS	Tiny Data Server
VHT	Very High Temperature

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Chapter 1

Introduction

According to the German Ministry to Economy and Energy (Bundesministerium für Wirtschaft und Energie), the heating industry currently represents about 35 % of the total energy consumption in Germany, meaning that there is an increasing pressure to tackle this issue by lowering energy consumption and increasing the share of clean energy for central heating and domestic hot water purposes. This can be achieved by decreasing the heat load of buildings or by increasing the primary energy efficiency of the appliances used for this purpose. At a first glance, the answer for such challenges seems quite straightforward: improve dwelling insulation and use solely electrical driven heat generators such as heat pumps for space heating as well as for domestic hot water. But it is not this simple. In a report for the European Commission, Kemna (2014) concluded that the average heat demand in the EU can be considered constant for the upcoming years. Although building insulation improvement must continue to be encouraged, it cannot be considered as an ultimate solution for the decarbonisation of heating, mainly because it is contradicted by phenomenon such as population increase. Air sourced heat pumps present performance issues at colder temperatures which compromise efficiency and durability of such appliances due to the need of oversizing (Klein et al., 2013). Moreover, there is a temperature flow limitation entailed to heat pumps due to the refrigerants condensation limits. There are many possible heating solutions to tackle these problems available on the market, which derive from distinct strategies adopted by appliance developers. But it is not yet clear which one of them carries the best outcome in terms of CO₂ emissions reduction, energy savings, customer satisfaction and profitability. These problems are further analysed and clarified in Chapter 2.

One of the mentioned solutions consists in combining a boiler and an air-to-water heat pump in one appliance denominated as hybrid heat pump. The premise behind these systems is that the secondary heat generator (boiler) can assist the heat pump when the outdoor conditions cause its operation to be inefficient, therefore achieving an optimal primary energy efficiency. Klein et. al (2013) performed a numerical study on such systems, concluding that they have potential for significant energy savings especially in well insulated dwellings, and advised further assessment provided that governments support the deployment of hybrids in existing buildings with subsidies. Di Perna et. al (2014) performed an experimental study on a hybrid generator composed of an air source heat pump coupled with a condensing gas boiler in a residential building, having reached the conclusion that “they are probably the best solution for reducing the primary energy needs of existing buildings”. Hybrid heat pumps seem therefore to be a promising solution for the decarbonisation of the residential heating sector.

At Bosch Thermotechnik GmbH, the development of such systems is being performed for future market deployment, and the present work was done in behalf of this project. Although experimental assessments such as the abovementioned one have already been done and turned out to be considerably conclusive, there is a need for the company to make its own measurements on the these devices.

Firstly, in order to release any kind of heating appliance to the market, the European Commission requires an energy label indicating their energy efficiency, which has to be experimentally assessed (European Commission, 2018). On the other hand, it is important for the company to quantify the Coefficient of Performance (COP), primary energy savings and potential cost savings for costumers of such systems for several reasons: further research and development, profitability analysis, marketing and to have a good general project overview. Measuring capacity is therefore required.

The existing test rigs in the plant where this work was developed have been measuring boilers for several years now but need some adaptation and preparation for being able to measure hybrid generators, which include air-to-water heat pumps in their setup. And this is the point where the practical work developed for this dissertation comes in.

Given that a hybrid heat pump heats up water for central heating and possibly for domestic hot water (DHW) with a heat pump and a boiler (depending on the operating conditions), certain conditions regarding the heat sink and heat source must be simulated inside the laboratory in order to test them. More specifically, the outdoor unit (ODU) of the heat pump must be placed inside a climate chamber which replicates outdoor air conditions, acting as a heat source. On the other side, the hot water flowing out of the appliance must be cooled down in the same way it would in a central heating system for space heating for different operating points (35 °C, 55 °C, etc.). All in all, this is a complex energetic system that transfers heat from a medium to another and can behave in unpredictable ways, as it was verified during testing phase and shown further in this work. Furthermore, in order for measurements to be valid, the achieved temperature and humidity conditions in the heat source, as well as inlet and outlet temperatures of the heat sink must be maintained inside specific boundaries during testing period according to the European standard DIN EN 14511. As such, this standard has been the baseline for the performed work, as the goal is to establish a laboratory capable of achieving suitable conditions to carry out valid tests.

Both the climate chamber and the heat sink already existed before but their capability of achieving and maintaining the aforementioned conditions had not been verified. The climate chamber was characterised at different conditions in order to make sure that temperature and relative humidity were maintained around the heat pump's evaporator air inlet. In the heat sink, all the components as well as the hydraulic scheme were assessed, followed by the implementation of a Simulink program to control the modulation of its pumps and valves in order to achieve and maintain the desired conditions during testing. The program required PID (proportional integral derivative) control for these components given that it is a dynamic and unstable system. The existing data acquisition system consisted in Matlab and Simulink modules which processed and stored the signal provided by the several temperature and flow sensors. Data was further compiled and organized for presentation purposes in Excel.

The initially described context on the challenges faced by the heating industry as well as the heating generators involved in this work were analysed in Chapter 2, consisting of a collection and review on the information provided by the literature. Chapter 3 aims to elucidate the reader about the motivation as well as the nuances associated with testing such devices, and to present in detail every relevant requirement in the standards DIN EN 14511 and DIN EN 14825 for this work. Chapter 4 and 5 contain the methodology, results and discussion of the analysis and work performed in order to correctly set up the heat source and heat sink respectively. Lastly, conclusions about the current status of the laboratory were drawn and required future work was suggested. An overview of the work and time spent at Bosch Thermotechnik GmbH has also been done, including a personal insight about the future of hybrid heat pumps and of the residential heating industry in general, with some strategical reflections concerning new products which could potentially be developed.

Chapter 2

State of the art

2.1 Introduction

The scope of this dissertation encompasses the comprehension and analysis of the current energetic paradigm of the building sector in Europe, the main goals set for the upcoming years in terms of greenhouse gas emissions and energy savings, and the possible solutions and means of achieving these targets without compromising indoor climate comfort. The assessment and discussion of these approaches are a key factor to understand the purpose of the work done in the laboratory.

As such, this chapter presents to the reader an introduction of the aforementioned issues and will hopefully clarify their connection with the main practical work developed during the internship.

2.2 Climate challenges

The main engineering challenges of the 21st century lie on the need to reduce the energetic consumptions and greenhouse gases emissions inherent to economic activities, and turn our growth path towards a sustainable future, where present generations are not spending the following ones' resources. To do so, a thorough investigation of every possible solution must be made in order to achieve an economic viable transition of the energetic paradigm.

In Europe, the political and economic pressure for reducing emissions and subsequent implementation of environmental friendly solutions is increasing (The European Parliament and the Council of the European Union, 2010). Germany aims to reduce greenhouse gases emissions from 80 to 95 % by 2050 relatively to 1990s levels, and to do so, the building sector was targeted to cut off about 70 million tons of CO₂ by 2030, which accounts for a 55 % reduction in comparison to the same period (The Boston Consulting Group, 2018). The United Kingdom has set similar goals (80 % reduction by 2050 compared to 1990s levels) (Howard & Bengherbi, 2016).

In order to achieve such targets, action from governments, scientific and industrial input from companies, and global conscience and willingness to change from people are required. From a political point of view, the strategy adopted by governments, which might be led by many different incentives, plays a key role in the future energetic paradigm of a given country. The two distinct approaches taken by Germany and the United Kingdom make an interesting case. According to Agora Energiewende (2017), Germany is opting for a more electrically driven future, with a general encouragement to implement a high number of heat pumps as well as hybrid systems, whereas the UK is focused in creating a green gas network and to maintain the high usage of gas driven heating systems (Howard & Bengherbi, 2016). Following these political

trends, lie the strategic approaches taken by heating systems developers and providers, such as Bosch Thermotechnik GmbH. Creative “green” solutions that fit different markets are therefore required from these companies. Lastly, a global population awareness on the aforementioned issues is crucial for CO₂ emissions reduction by the building sector due to the impact of good practices (or rather a lack of them) in the energy consumption of a household.

Given that the present work was developed in Germany, the heating market of this country was analysed more thoroughly. Figure 2.1 shows the number of households using different of energy sources for heating.

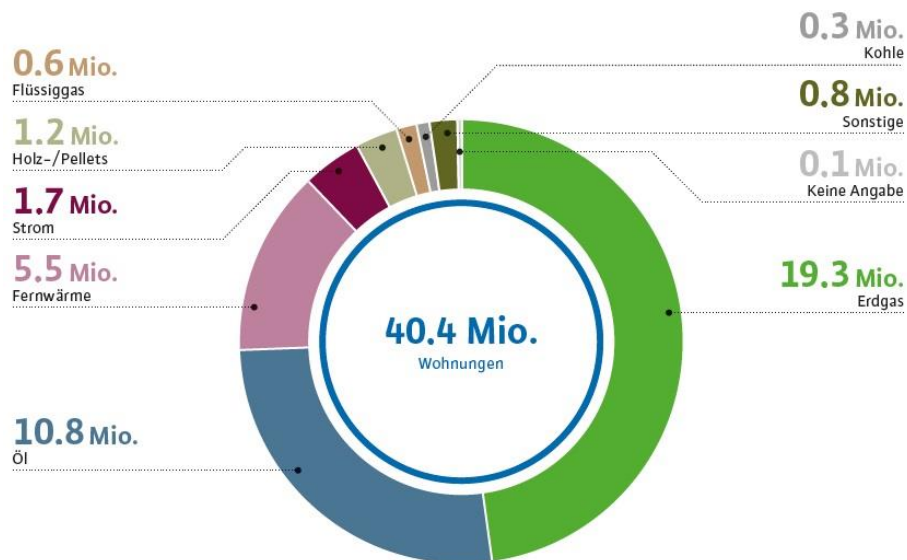


Figure 2.1 – Statistical analysis of heating sources in Germany (BDEW, 2015).

The statistics in Figure 2.1 show that from 40.4 million homes in Germany, 19.3 million are heated by gas driven appliances (60 % of which are non-condensing boilers (Bundesverband der Deutschen Heizungsindustrie, 2017)), 10.8 million with oil, 5.5 million with district-heating and only 1.7 million are heated by electrical driven solutions. In Germany, one of the main goals leading the country to decarbonisation would be to reduce as much as possible the oil-driven heating systems, mainly by replacing them for district-heating solutions, electrically driven heat pumps as well as hybrid setups. On the other hand, gas is used in almost 50 % of the households, meaning that there is a key role to be played by the developers of gas-driven boilers on maximizing their efficiency. As for broader measures, requiring political action, there are three main nuances that should be carefully considered:

- Even if 100 % of the gas appliances used had the maximum efficiency available on the market today, the energy source powering these systems would still be a polluting and limited one. As such, implementing changes to the current gas network in order

to make it “greener” is essential to meet 2050 targets. The number of alternative highly efficient heating solutions using gas as fuel, such as gas driven heat pumps are becoming more and more popular;

- Implementing 5 to 6 million heat pumps (Agora Energiewende, 2017) might be dangerously misleading. One must keep in mind that if the source used to produce electricity to power all of these heat pumps is a traditional thermoelectric plant, these systems will be polluting even more than those they are meant to replace. So, however obvious it may seem, promoting renewable sources to follow the electrically driven heating systems trend is a priority;
- Electricity prices must be discussed, as they represent an important variable in the heating market. Germany, for example, has the highest electricity prices in Europe and one of the highest in the world. This conditions the adoption of electricity powered heating appliances, since it raises the return on investment time. An overview of the electricity and gas prices in Europe is provided on Annex 3.

The mentioned heating solutions will be thoroughly explained in a further section of this chapter.

2.3 The impact of the building sector

ASHRAE Standard 55-2010 (2010) defines thermal comfort as “that condition of mind which expresses satisfaction with the thermal environment and is assessed by subjective evaluation”. It is also stated in the same standard that it is impossible to achieve 100 % of satisfaction regarding indoor thermal conditions, so the ultimate goal of the studied heating systems is, in fact, to maximize the number of thermally satisfied individuals in an indoor space. It is important to keep in mind that this is what Bosch Thermotechnik GmbH and its products thrive for.

Thus, the goal is to decarbonize heating without compromising comfort. It is not an easy target, and the importance of research in this field derives from the statistics regarding energy consumption and CO₂ emissions associated with indoor heating. According to the German Ministry to Economy and Energy (Bundesministerium für Wirtschaft und Energie), the residential heating sector accounts for about 35 % of all the energy consumed in Germany. In the United Kingdom, 32 billion (10⁹) pounds are spent annually in heating, “accounting for nearly half of all energy consumed and one third of total greenhouse gas emissions” (Howard & Bengherbi, 2016). According to a study done for the European Commission by Kemna (2014), HVAC equipment “is the largest contributor to the total EU energy consumption”. Given this situation, it can be concluded that improving efficiency in the power and transportation sectors will not be enough to meet the desired targets.

Reducing energy consumption in building heating can essentially be achieved in two ways: by lowering the heat load demand (which can be done with insulation improvement) or by increasing the efficiency of the heating systems, which consequently also reduces primary energy consumption. In order to correctly analyse and discuss these options, one should keep in mind that, according to Klein, et al. (2013), “in Germany, the annual number of new constructions is as low as 0.5%”, meaning that the newly constructed and highly efficient buildings do not provide a solution for the discussed problem, due to the overwhelming majority of 20th century building stock that will prevail in the upcoming decades. Furthermore, Kemna (2014) concluded the following:

“The future trend in the total EU space heating load can be expected to be more or less stable. Improved insulation, optimised ventilation (with heat recovery), increased urbanisation (heat islands) and global warming will lead to a decrease of the load. Growth of population, dwelling size and comfort level will lead to an increase of the space heating load. A new phenomenon is the diminished contribution of internal heat gains from lighting and appliances due to efficiency improvement, which will contribute to an increase of the heating load for space heating systems.”

Given the presented paradigm, the way of decarbonizing indoor residential heating relies on achieving maximum primary energy efficiency in appliances.

2.4 Heating appliances

This section provides the reader with an overview of the main available residential heating solutions, in which the hybrid heat pumps are included.

2.4.1 Condensing boilers

Gas and oil account for about 75 % of the energy sources used in domestic heating in Germany (BDEW, 2015). As such, improving the efficiency of the heating devices utilizing these energy sources is a key factor to reduce overall CO₂ emission in this sector.

A boiler is an appliance which heats up water by utilizing the energy from the combustion of fossil fuels (gas or oil). Condensing boilers also make use of the latent heat from exhaust gases to heat the cold water in a secondary heat exchanger before entering the primary one (where combustion occurs), as shown in Figure 2.2 (Satyavada & Baldi, 2018). In this way, the water vapour present in the flue gases condenses and the energy demand for achieving a certain water

temperature is not as large as in traditional boilers, since the temperature difference is reduced with the pre-heating (Junkers, 2018).

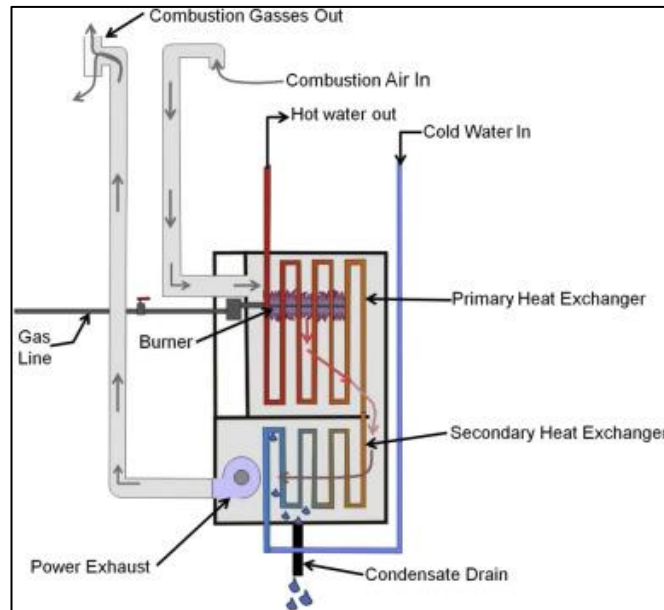


Figure 2.2 – Gas condensing boiler functional representation (Satyavada & Baldi, 2018).

According to Vignali (2016), gas condensing boilers are taking over the heating market by replacing traditional boilers, but only account for about 10 % of the existing gas-fired boilers in Europe. In the same article, the author states that “condensing boilers are in fact considered the best available technology on the market, with only a small margin for improving efficiency”. Further exploring means of obtaining higher efficiency in these devices is therefore not interesting for this dissertation. However, it is important to understand how they work, as they act as a backup heat provider in hybrid heat pump systems.

2.4.2 Heat pumps

Heat pumps are heating appliances which transfer energy from a medium (heat source) to another (heat sink) and take mechanical power as an input, which is normally obtained from an electrically driven compressor. Usually, heat pumps perform this task with an ordinary vapour compression refrigeration cycle and the most widely used refrigerants have been R22, R134a and R407c (Wang, 2000), with a transition to R410a occurring with the most recent products. Transcritical CO₂ cycles have also been adopted in a wide range of heat pump models developed in Japan (ASHRAE, 2010).

COP is the most relevant mathematical parameter characterising these devices, and it is defined in the European standard DIN EN 14511 as the “ratio of the heating capacity to the

effective power input of the unit, expressed in Watt/Watt”. *COP* is given by the following formula:

$$COP = \frac{\dot{Q}}{P_{ele}} \quad (1.1)$$

where $\dot{Q}$ is the heating power output of the heat pump and P_{ele} the electric power input. A more thorough analysis of the mathematical parameters inherent to these devices will be made in a further section regarding the standards associated to heat pump and hybrid system testing and parameter calculation.

Regarding the medium from which the refrigerant absorbs heat from, these devices can essentially be divided into air-source and water-source. In this dissertation, water-source heat pumps will be disregarded as they are mostly a popular choice in Scandinavian countries, where outside air temperature hits very low temperatures during a significant time of the heating season, compromising the overall performance of air-source heat pumps. The most common in commercial and residential use are air-source heat pumps and that these devices are used in the majority of the hybrid setups, including the ones developed at Bosch.

Outside air is therefore the heat source for the considered appliances, and the produced hot water is the heat sink in air-to-water heat pumps. The heat is further dissipated to the house, by heating water which is used for sanitary purposes or for central heating. This process is shown in a simplified way in Figure 2.3.

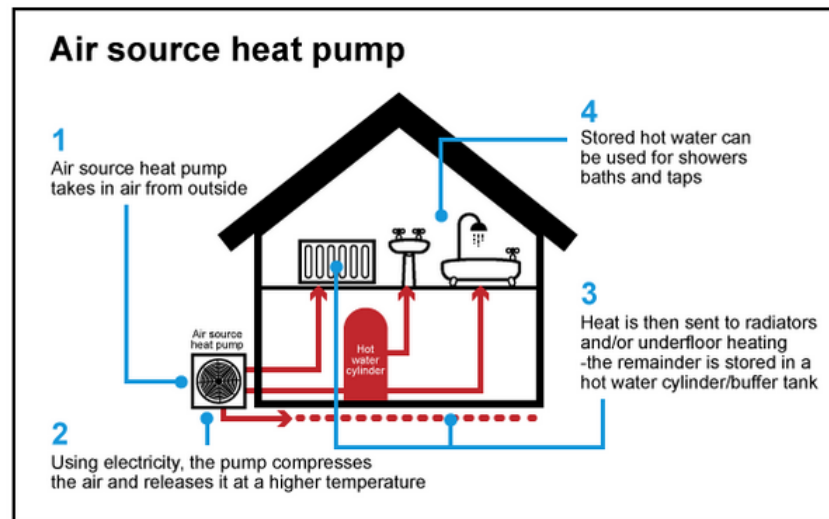


Figure 2.3 – Schematic of a heating system provided by an air-to-water heat pump.

Source: www.clarityheating.com

There are many types of possible configurations for these devices regarding their physical setup. The outdoor unit could only contain the evaporator, compressor and expansion valve, in which case it is called a split unit, with the remaining components such as the condenser being placed inside an indoor unit (IDU). Split units exchange refrigerant with the indoor unit. Given that clients often complain about the ODU oversizing, which is aesthetically undesired, this is a popular choice due to possibility of a reduced size of the ODU. However, it entails more complex piping and controlling systems in IDU, as the hot and high pressure refrigerant leaves the unit and returns after going through the condenser in the IDU. Another common setup is to have a compact unit, where the whole refrigeration cycle is placed inside the ODU, undertaking a larger footprint. These units exchange water with the IDU. Both systems are represented in Figure 2.4.

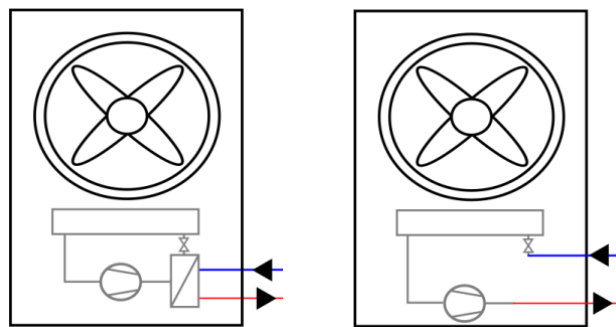


Figure 2.4 – Representation of a compact ODU (on the left) and a split ODU, which does not include a condenser (on the right)

As mentioned before, the large footprint entailed with ODUs can be a source of complaint and therefore another solution provided by some developers is to place the whole system inside the building and connect it to outdoor air via a duct system. These are called ducted heat pumps. However, they carry a nuance – the hydraulic diameter of the ducts must be large enough so that the head losses do not implicate the need of too much electric power for the fan that circulates the air through the ducts. So, the limitation here is that either the ducts are very large, or the efficiency of the whole system is compromised. There is still a big discussion about what kind of devices to bet in within the developers of heat pumps, because it is not yet clear what solution provides better results in terms of product quality, customer satisfaction, efficiency and profitability combined.

Another important topic regarding heat pumps is the type of control system for the compressor (the component that spends about 90 % of the required electrical power), which can be ON/OFF or inverter-driven when a mono-compressor system is used. An ON/OFF control implies that the compressor only works at one constant frequency, and that for an outside temperature over a certain value, the heat pump starts ON/OFF cycles instead of working permanently, given that its full capacity is not required. This is known as bivalent temperature

(T_{biv}), and such an operating mode is highly inefficient due to energy losses (Dongellini, et al., 2015). According to the same author, inverter-driven compressors allow a more efficient operation by modelling the frequency function of the outside temperature, but implies a more expensive and complex system. At last, there are also multi-compressor systems, which are composed of more than one low power compressors. This allows for a big range of outside temperatures where the heat pumps present an optimum efficiency. The lower temperature limit corresponds to every compressor being turned on and the higher limit corresponds for only one of the compressor in operation.

Heat pumps are a promising technology that combined with renewable electricity sources, can deliver the required heat for our comfort with no use of fossil fuels, given that they utilize the energy present in the outdoor environment to heat water for our homes. However, there are some disadvantages inherent to this technology holding it back from further usage, which make heating solutions developers consider other options for a sustainable future.

Firstly, an undesired characteristic of an air-sourced heat pump is the need for defrosting of the outdoor heat exchanger (evaporator). When the outdoor conditions cause the air in the evaporator entrance to be below dew point temperature, frost is formed on the surface, which entails a drop of exchange efficiency due to reduced air flow and added thermal resistance (Qiao, et al., 2018). The device enters therefore a reversed cycle, with heat injection on the surface of the evaporator's coil in order to melt the built frost. This effect causes an efficiency drop due to added power usage with no useful heating output effect. Baxter and Moyers (1985) assessed that 10.2 % of the power consumption of a high efficient air-to-water heat pump is used in defrosting.

However, the main problem is the need for oversizing in order to cover the whole heating season, especially in countries where temperatures hit low values for only a small fraction of the heating season. A drop of the *COP* in air-source heat pumps is observed for low temperatures and, in general, the higher the difference between the heat sink and the heat source temperature, lower the efficiency (Grassi, 2018). This implies that heat pumps work at partial load conditions during most of the heating season, which is energetically and cost-wise inefficient. Most of the developers and manufacturers of heat pumps only disclose the *COP* values for nominal conditions (Dongellini, et al., 2015). However, the appliance only operates at such conditions for a very limited time frame during the heating season. Thus, for a correct evaluation of the overall efficiency, the seasonal coefficient of performance (*SCOP*) is needed. It is defined by the European standard DIN EN 14825 as the “overall coefficient of performance of the unit, representative for the whole designated heating season”. The *SCOP* value for a given heat pump is dependent on the control system used and the degree of oversizing, which is subsequently related to the ratio between the heating power of the heat pump and heating demand conditions throughout the heating season, and it is always smaller than the actual *COP*, which is a consequence of the aforementioned problems.

Different solutions are being proposed by manufacturers. Another type of heat pumps being developed and produced are Gas Engine Heat Pumps (GEHPs), in which the compressor is driven by a small gas engine, turning the power modulation easier and, as such, this solution is claimed to be the overall most efficient in terms of primary energy, due to the combination of renewable nature of the heat source in heat pumps with the flexibility and efficiency provided by natural gas usage with highly efficient engines (AISIN Group, 2018).

However, the proposed solution in which this thesis is focused on is denominated by hybrid heat pump or bivalent heat pump and will be analysed in the following section.

2.4.3 Hybrid heat pumps

Following up the presented challenges for heat pumps, a popular solution among developers is the hybrid heat pump. These appliances include an ordinary air-to-water heat pump and a backup heater (frequently a condensing boiler) which supports the heating system when the energetic demands are too high for the heat pump. This allows a moderately sized air-to-water heat pump for the most of the heating season to be selected without the risk of not fulfilling the required heat load in the peak of the heating season thanks to the support given by a condensing boiler, which is also an efficient appliance.

There are many possible solutions for assembling such devices, as well as for the type of operation they are programmed to perform under the control system. The hydraulic scheme of two existing products in the market are presented in order to highlight the difference between two possible solutions. For simplicity purposes, these will be called Hybrid 1 and Hybrid 2. Both of them consist of packaged solutions of hybrid heat pumps, meaning that there is an outdoor split unit and a packaged indoor unit with the remaining components (including the boiler).

Hybrid 1 is represented in Figure 2.5, and it is the simpler solution and the boiler runs on oil. Oil boilers have a tendency to be more expensive given that the materials used attain more heat stress and that oil must be pre-heated and prepared before burning. The secondary heat exchanger pre-heats the water by condensing the flue gases from the boiler. Usually this is included in the condensing boiler, but in this case it was placed outside and was therefore represented separately. As it can be seen, V1 controls the amount of flow that is heated by the heat pump or by the boiler. The hydraulic separator allows for two hydraulically independent circuits, accounting for an easy and cheap way of controlling the flow temperature to the central heating system as well as how much of it is achieved by the heat pump and the boiler. This product does not provide a DHW tank, so a complementary system would be necessary for that purpose.

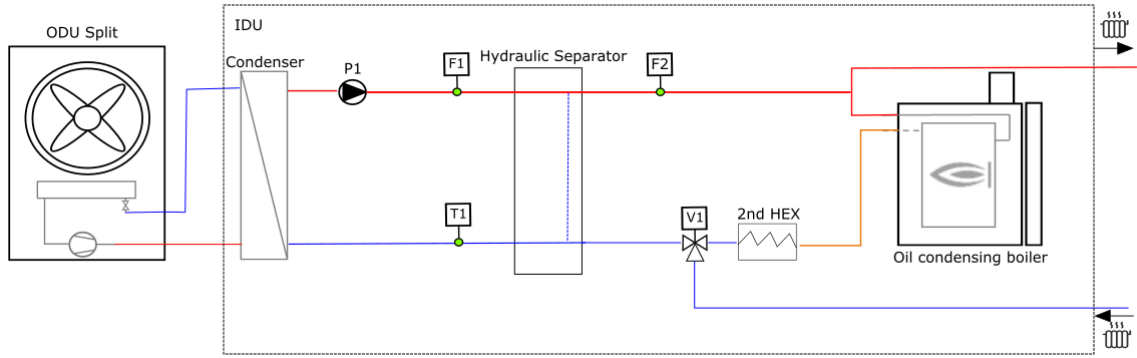


Figure 2.5 – Hybrid 1 hydraulic representation

Hybrid 2 accounts for a more complex system composed by three 3-way valves, seven temperature sensors, two circulation pumps, two pressure gages, one flow meter as well as a complex piping system. V1 controls the amount of flow going to the heat pump, V2 controls the flow temperature and V3 controls how much of the hot flow is used for central heating or for DHW purposes. This device includes a domestic hot water tank, but the water is pumped through a heat exchanger which heats it before it is stored inside the tank, unlike the usual solution, which is a heating coil inside the DHW cylinder.

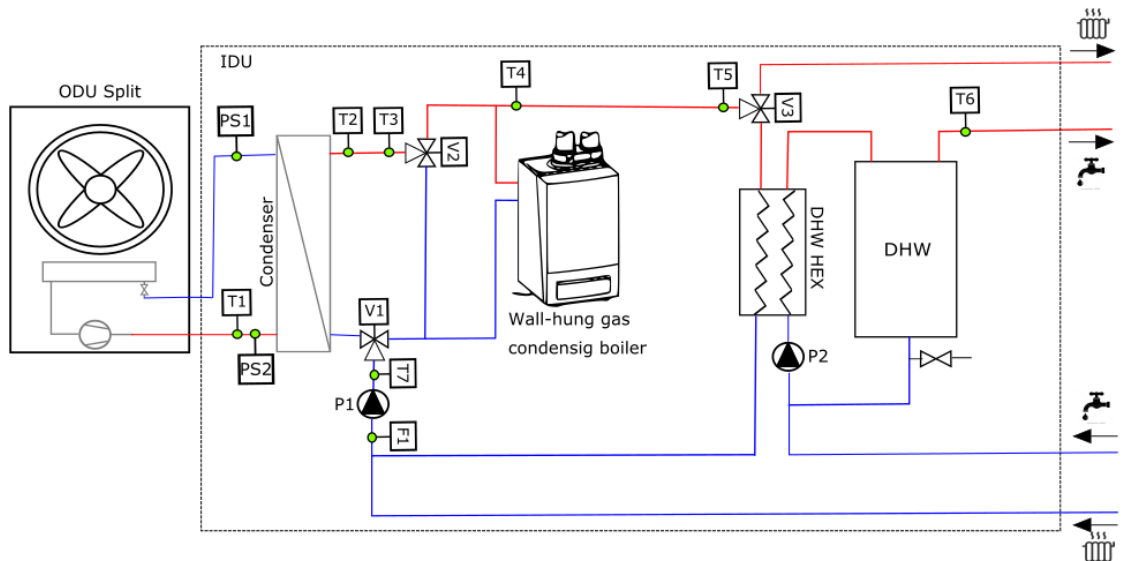


Figure 2.6 – Hybrid 2 hydraulic representation

Hybrid 1 has an ErP label of A ++ while Hybrid 2 has A +. In terms of footprint (physical space), Hybrid 2 accounts for a bigger IDU, but this includes DHW system with its respective tank, so it can be considered more space efficient. Overall, both are promising solutions and the goal is not to assess which product is better (as it is a rather complex task) but to present the

reader with two real life examples of hybrid configurations which are being sold, as well as showing how different they could be.

Klein, et al (2013) conducted a study about the feasibility of retrofitted hybrid setups in existing buildings motivated by the potential of this solution given the energetic paradigm mentioned on the previous sections of this chapter. By means of a numerical simulation, they assessed that these systems show a considerable primary energy efficiency improvement, especially for well insulated buildings and for mid-range capacity heat pumps. As such, further investigation was advised by the authors, provided that financial support is given by governments to lower electricity prices for example, which are particularly high in Germany (Annex 3) acting as a barrier for the proliferation of such systems due to the entailed return on investment time.

Finally, the importance and relevance of testing such products can now be fully understood. From the need of creating products that cut CO₂ emissions off without compromising indoor climate comfort to the analysis of different products and solutions, the hybrid laboratory presents itself as an essential tool for the business development of Bosch Thermotechnik GmbH, and the characterization presented in this dissertation will clarify the current capabilities and potential inadequacies facing the measuring demands presented by the standards DIN EN 14511 and DIN EN 14825.

Chapter 3

Measurement Requirements

3.1 Introduction

As assessed in the previous chapter, there is a need to overcome the challenges posed by the energetic spending in the building sector inherent to indoor climate comfort. This calls for new technology and hybrid heat pumps arise as a suitable candidate considering the current paradigm. For the development of such appliances, a laboratory for testing is necessary given that the European Commission requires efficiency measurement for energy labelling purposes, according to Regulation No. 811/2013. In this chapter, a comprehensive overview of testing procedures and requirements according to DIN EN 14511 and DIN EN 14825 is presented, as well as an overview of the mentioned energy label requirements.

3.2 Motivation for testing

Energy labels containing information about efficiency as well as technical documentation with some more performance indicators are both required for market deployment of hybrid heat pumps (as well as for many other different heat generators and electric appliances in general), thus the need for measurement capacity. Moreover, it is important for Bosch Thermotechnik GmbH, as a developer and manufacturer of such systems to have knowledge of the performance values from its appliances as well as from the competitors' ones in order to improve and further develop better products. To reduce time costs associated to development, it is desirable for the laboratory to be as automated as possible in terms of operation and data acquisition and manipulation. However, for all the measurements to be valid for the mentioned purposes, the conditions must comply with boundaries imposed by the European standard DIN EN 14511.

Many uncertainties arise when a purely theoretical analysis of systems such as hybrid heat pumps is made. These machines are relatively complex and depend strongly on operating conditions which cannot be controlled so numerical simulations are not powerful enough for assessing their behaviour in real life conditions. Hybrids depend on outdoor conditions (temperature and humidity) and on heat load requirements. This last one is especially hard to simulate because it depends on the insulation of the building, on the household, on the number of hours as well as the time of the day where the demand is higher, on the internal gains of the building (from electrical appliances, for instance) and on the characteristics of the heating systems (radiators, radiant floor). As such, the unpredictability of human and nature behaviour influences the operating points of these systems, so much so that a simulation tool powerful enough to predict overall energy usage, *COP* and *SCOP* is not available yet. Furthermore, the task is even harder if the heterogeneity of a country like Germany is considered. The housing stock in the north and the south as well as human behaviour are distinctive, and the climate

conditions depend also in the geographical location as well as demographic density, due to phenomenon such as heat islands in cities (Kemna, 2014). It is believed that only with Artificial Intelligence based tools such as Deep Learning will these kind of assessments be possible without measurement.

All in all, measuring the behaviour of these devices in a laboratory and with field tests is the only solution for making an analysis of their real potential for energy savings and of the different hydraulic and control solutions, so that it is possible to evaluate the feasibility of products in terms of efficiency, market impact and profitability.

3.3 Testing hybrid heat pumps

A hybrid heat pump includes an air-to-water heat pump and a secondary heat generator, usually a boiler. As such, for measuring such systems, the power of the heat pump and the boiler must be measured. The European standard DIN EN 14511 dictates the conditions at which tests must be performed, measuring uncertainties and testing procedures in order to make a valid assessment of heating power produced and electric power consumption for heat pumps. The DIN EN 15502 standard dictates equivalent parameters for boilers testing, but will not be analysed in detail here, as the laboratory has already all the necessary conditions and know-how for testing such appliances, so suitable and compliant test rigs are already commissioned. Standard DIN EN 14825 dictates the calculation methods for various heating appliances including hybrids so, it is the standard in which the actual procedures for assessing performance and efficiency values of hybrid heat pumps are stated, and it depends on the experimental procedures defined in DIN EN 14511 and DIN EN 15502 to get the necessary values for such calculations.

For the air-to-water heat pump, outside air acts as a heat source. The produced hot water is sent to the central heating system or to a DHW tank, so the building acts as a heat sink. For testing hybrids, it is therefore required to simulate these conditions inside a laboratory.

For the heat source side, a climate chamber is necessary to create controlled environment conditions. In order to market and label a product involving a heat pump component according to EC regulation No. 811/2013, the climate chamber must be capable of maintaining temperature and humidity conditions over periods of time long enough for testing in a concrete space according to the specifications in DIN EN 14511, which will be further analysed. This system is fully characterised in Chapter 4.

As for the heat sink side, a device usually denominated as a test rig must take in the hot water produced by the heat pump and dump that heat by cooling the water in a controlled way, so that flow and return temperatures as well as volume flow rate can be controlled regardless the operating condition of the heat pump. This way, the dynamic behaviour of the heating load of a given building can be simulated in a controlled environment.

Heat power is given by the following equation:

$$\dot{Q} = \dot{m} \times c_w \times \Delta T \quad (3.1)$$

and mass flow rate is given by:

$$\dot{m} = \rho \times \dot{V} \quad (3.2)$$

The water density ρ and specific heat c_w can be considered constant for the operating range, meaning that heating power is thereby controlled by volume flow and difference between flow and return temperatures (ΔT). A detailed description of the test rig that provides such conditions is given in Chapter 5 and corresponds to the second part of the practical work performed during this thesis.

As seen in Figures 2.5 and 2.6 of the previous chapter, which show hydraulic schemes of two different hybrid setups, the power from the heat pump cascades with the power from the boiler when both systems are working simultaneously, by joining the flow coming from the heat pump to the hot flow originated in the boiler. Figure 3.1 shows how this systems would be placed for making measurements on the Hybrid 1 setup.

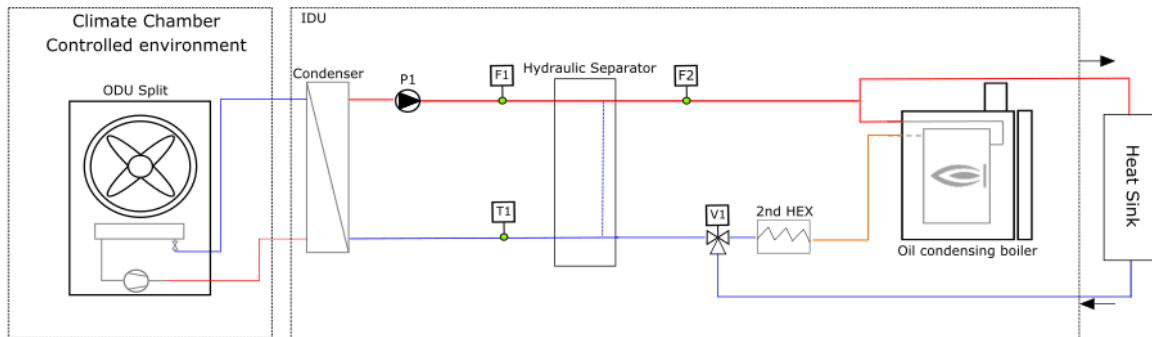


Figure 3.1 – Representation of a testing scheme for a hybrid heat pump

Both testing devices will be thoroughly explained as well as all the measuring equipment associated with them. The next sections of this chapter will describe the normative requirements for such testing systems.

3.4 Energy Label

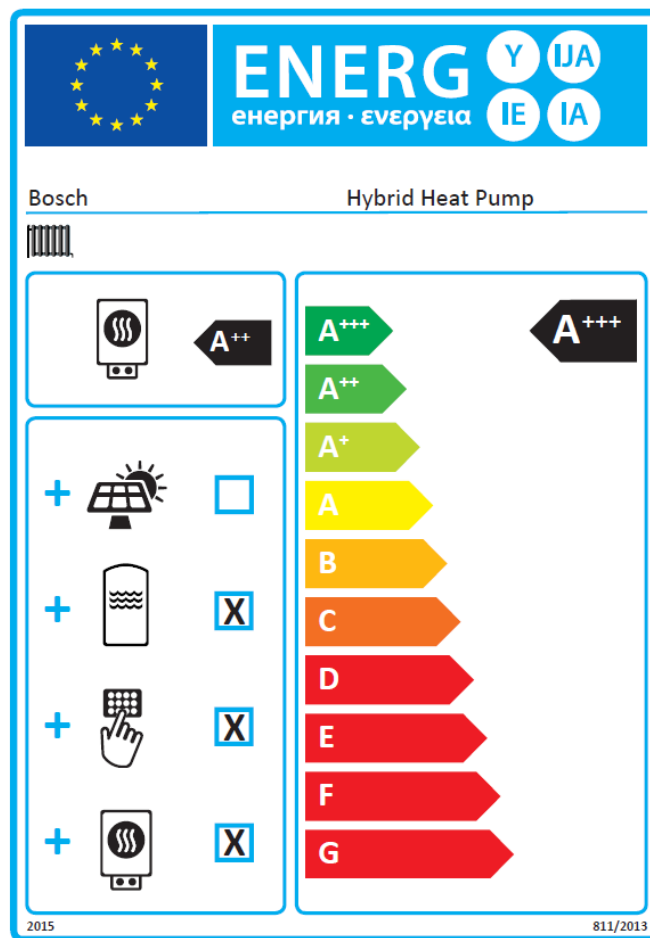
The Regulation (EU) No 811/2013, which is part of the Directive 2010/30/EU appeared as an added measure to mitigate the energetic consumption of appliances providing space and water heating. The goal of the regulation is to incentive manufacturers to develop efficient appliances as well as providing consumers with easily interpretable information about all the available options on the market, so a more educated purchase is facilitated. It “is expected to result in estimated annual energy savings of around 1 900 PJ (about 45 Mt) by 2020, corresponding to about 110 Mt_{CO2} emissions, compared to what would happen if no measures were taken” (European Commission, 2013)

The regulation aims to do so by enforcing the presence of label which classifies the energy efficiency of a given appliance as a condition for its releasement to the market. The scale for such a classification depends on the type of unit and what whether it uses renewable energy sources or not. The top of the scale could be as high as A+++, and the bottom is always G. To asses this parameter, the seasonal space heating energy efficiency (η_s) must be calculated and it is defined as “the ratio between the space heating demand for a designated heating season, supplied by a space heater, a combination heater, a package of space heater, temperature control and solar device or a package of combination heater, temperature control and solar device, and the annual energy consumption required to meet this demand, expressed in %”. This value is the reason why the procedures in DIN EN 14825 and subsequently on DIN EN 14511 are needed, as they provide the required calculation and measurement standards for determining η_s .

After assessing the mentioned value, the classification must be done in accordance to Table 3.1. The presented values do not apply for low-temperature application heat pumps. After making the necessary assessments for the type of appliance and its efficiency, a label can be created on the European Commission website. For demonstration and clarification, a label was created for a hybrid heat pump with A+++ efficiency which includes hot water storage and temperature control and supplementary heating (inherent to a hybrid definition), and it is shown on Figure 3.2.

Table 3.1 – Seasonal space heating energy efficiency (η_s) value for equivalent class

Seasonal space heating energy efficiency class	Seasonal space heating energy efficiency η_s in %
A +++	$\eta_s \geq 150$
A ++	$125 \leq \eta_s < 150$
A +	$98 \leq \eta_s < 125$
A	$90 \leq \eta_s < 98$
B	$82 \leq \eta_s < 90$
C	$75 \leq \eta_s < 82$
D	$36 \leq \eta_s < 75$
E	$34 \leq \eta_s < 36$
F	$30 \leq \eta_s < 34$
G	$\eta_s < 30$

**Figure 3.2** – Energy label example

3.5 DIN EN 14825

Performance values at nominal conditions are not a good enough indicator for a heat pump given that most of the time it does not operate at such conditions. The standard DIN EN 14825 defines SCOP as the “overall coefficient of performance of the unit, representative for the whole designated heating season” and that it is calculated “as the reference annual heating demand divided by the annual energy consumption for heating”. The standard also states that measured values have to obey to conditions imposed by DIN EN 14511 for heat pumps and DIN EN 15502 for boilers. But most importantly, it provides the methodology for calculating the seasonal space heating energy efficiency (η_s), which is required for the energy label defined by the Regulation No 811/2013 by the EU. According to this standard, η_s is given by:

$$\eta_s = \frac{1}{CC} \times SCOP - \sum F(i) \quad (3.3)$$

where CC is the conversion factor and is equal to 2,5 and $\sum F(i) = F(1) + F(2)$, where F(1) is the correction that accounts for a negative contribution to the seasonal space heating energy efficiency of heaters due to adjusted contributions of temperature controls, equal to 3 % and F(2) is the correction that accounts for the negative contribution to the seasonal space heating energy efficiency by energy consumption of brine and water pumps. This factor is only for water (brine)-to-water (brine) and water (brine)-to-air units and is equal to 5 %. The most relevant factor in that equation is SCOP, given by:

$$SCOP = \frac{Q_H}{Q_{HE}} \quad (3.4)$$

where Q_H represents the reference annual heating demand, expressed in kWh and Q_{HE} represents the annual energy consumption for heating, expressed in kWh. To calculate Q_H , the following formula applies:

$$Q_H = P_{designh} \times H_{HE} \quad (3.5)$$

where $P_{designh}$ is the design heating load the unit is suitable for as declared by the manufacturer, expressed in kW and H_{HE} is the number of equivalent active mode hours for heating, expressed in hours. The latest are provided in Annex 4 together with all the relevant information needed for calculations for different climates. As for Q_{HE} , it is given by:

$$Q_{HE} = \frac{Q_H}{SCOP_{ON}} + H_{TO} \times P_{TO} + H_{SB} \times P_{SB} + H_{CK} \times P_{CK} + H_{OFF} \times P_{OFF} \quad (3.6)$$

where H_{TO} , H_{SB} , H_{CK} and H_{OFF} are the numbers of hours the unit is considered to work in thermostat-off mode, standby mode, crankcase heater mode and off mode respectively, expressed in h and P_{TO} , P_{SB} , P_{CK} , P_{OFF} are the power inputs during thermostat-off mode, standby mode, crankcase heater mode and off mode respectively, expressed in kW. $SCOP_{ON}$ is defined as the average coefficient of performance of the unit in active mode for the designated heating season. It is arguably the more complex parameter to calculate in the whole process, for which specific procedures for hybrid heat pumps are provided. For hybrid heat pumps, two methods are provided: the combined method, which requires both heat generator working at the same time, and the separated method, which consists in assessing the relevant values for each generator individually, further combining them into a formula. Let us consider the combined method for the calculation of $SCOP_{ON}$:

$$SCOP_{ON} = \frac{\sum_{j=1}^n h_j \times P_h(T_j)}{\sum_{j=1}^n h_j \times P_{ele}(T_j) \times \sum_{j=1}^n h_j \times \frac{Q_{Fossil}(T_j)}{CC}} \quad (3.7)$$

where j is the bin number, n is the total number of bins, h_j is the number of bin hours occurring at the corresponding temperature T_j , which is the bin temperature, P_h is the heating load for the corresponding temperature T_j , expressed in kW, P_{ele} is the electric power input for the corresponding temperature T_j , expressed in kW, Q_{Fossil} is the fossil fuel power input for the corresponding temperature T_j , expressed in kW and CC is the conversion coefficient, equal to 2,5. P_h is obtained by multiplying the full value ($P_{designh}$) by the part load ratio correspondent to T_j .

As the scope of this dissertation is not to actually assess the efficiency of a specific appliance, but to improve a test rig so that it achieves standard compliance, the presented formulas are not thoroughly explained due to the lack of relevance. Annex 4 contains supplementary information on the bin values for the calculation of $SCOP_{ON}$ which leads to the assessment of the seasonal space heating energy efficiency, allowing the establishment of an energy label accordingly. A selection of relevant definitions was compiled in Table 3.2.

Table 3.2 – Selection of definitions adapted from DIN EN 14825

Term	Abbreviation	Definition
Hybrid Heat Pump	--	Encased assembly or assemblies designed as a unit consisting of an air/water(brine)/DX-to-water(brine) electrically driven heat pump with a second heat generator using fossil fuel, and managed by a common controller providing an optimized operation of the heat generators for space heating.
Seasonal coefficient of performance	$SCOP$	Overall coefficient of performance of the unit, representative for the whole designated heating season.
Active mode seasonal coefficient of performance	$SCOP_{on}$	Average coefficient of performance of the unit in active mode for the designated heating season, determined from the part load, supplementary heating capacity (where required) and bin-specific coefficients of performance ($COP_{bin}(T_j)$) and weighted by the bin hours where the bin condition occurs.
Switch temperature boiler off	$T_{fb,off}$	For hybrid systems, temperature above which the fossil boiler is switched off and heat only provided by the heat pump.
Switch temperature heat pump off	$T_{hp,off}$	For hybrid system, temperature below which the heat pump is switched off and heat only provided by the fossil fuel boiler.
Bin	--	Outdoor temperature interval.
Bin hours	h_j	Hours per season for which an outdoor temperature occurs for each bin j .
Bin limit temperature	--	Temperature in the bin for which no more heating or cooling is required.
Bin-specific coefficient of performance	$COP_{bin}(T_j)$	Coefficient of performance specific for every bin j with outdoor temperature T_j in a season.
Bin temperature outdoor temperature	T_j	Outdoor air dry bulb temperature.
Coefficient of performance at part load	COP_{bin}	Coefficient of performance at the declared capacity, corrected with the degradation coefficient, where applicable.
Cold climate conditions	--	Temperature conditions characteristic for the city of Helsinki for the heating season
Average climate conditions	--	Temperature conditions characteristic for the city of Strasbourg for the heating season
Warmer climate conditions	--	Temperature conditions characteristic for the city of Athens for the heating season

3.6 DIN EN 14511

DIN EN 14511 is an European Standard denominated “Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling prepared by the Technical Committee CEN/TC 113 “Heat pumps and air conditioning units” (Secretariat: AENOR, Spain). The conditions achieved in the test rig must be in accordance to this standard, from the climate chamber to the heat sink, in order to make valid heat pump measurements. It also refers to testing procedures, measurement errors and definitions. It is divided in four parts, which will be here analysed separately, with exception to part 4, so that a thorough comprehension of all the relevant information is achieved

3.6.1 Part 1: Terms, definitions and classification

The first part of this standard defines all the terms related to the rating of performance of heating devices such as heat pumps. The relevant terms for the scope of this thesis were put together in Table 3.3.

Table 3.3 – Selection of definitions adapted from DIN EN 14511

Term	Definition
Heat pump	Encased assembly or assemblies designed as a unit, using a vapour compression cycle driven by an electric compressor, to provide delivery of heat.
Packaged unit	Factory assembly of components of refrigeration system fixed on a common mounting to form a discrete unit.
Single split unit	Factory assembly of components of refrigeration system fixed on two mountings to form a discrete matched functional unit.
Heating capacity (P_h)	Heat given off by the unit to the heat transfer medium per unit of time, expressed in Watt.
Total power input (P_t)	Power input of all components of the unit, expressed in Watt.
Effective power input (P_e)	Average electrical power input of the unit within the defined interval of time, expressed in Watt, obtained from: - power input for operation of the compressor and any power input for defrosting; - power input for all control and safety devices of the unit; - proportional power input of the conveying devices (e.g. fans, pumps) for ensuring the transport of the heat transfer media inside the unit.
Coefficient of performance (COP)	Ratio of the heating capacity to the effective power input of the unit, expressed in Watt/Watt.
Rated coefficient of performance (COP_{rated})	Declared capacity for heating [kW] divided by the rated power input for heating [kW] of a unit when providing heating at standard rating conditions.
Defrost mode	State of the unit in the heating mode where the operation is modified or reversed to defrost the outdoor heat exchanger.

3.6.2 Part 2: Test conditions

The second part of the standard states the rating conditions at which tests must be performed. Table 3.4 shows all the rating conditions imposed by this standard for air-to-water heat pumps, which are related to outdoor air temperature and must be achieved in the climate chamber, as well as to the indoor heat exchanger temperatures, which must be achieved in the heat sink. T_{db} means dry bulb temperature, T_{wb} is wet bulb temperature, T_{in} is the inlet temperature for the indoor heat exchanger of the heat pump, which is the same as the returning flow from the heat sink, and T_{out} is the outlet temperature of the heat exchanger, which is equivalent to the inlet temperature of the heat sink. LT, MT, HT and VHT mean low temperature, medium temperature, high temperature and very high temperature respectively. These classifications regard flow temperature, which must be higher when the heat load conditions are more demanding. For instance, LT relates to a situation where the heat load demand is small, therefore the required flow temperature is qualitatively classified as low and quantitatively equal to 35 °C. The chosen temperatures to test the climate chamber derive from the required outdoor air conditions on this part of the standard.

Table 3.4 – Test conditions relevant for the scope of the dissertation. Adapted from DIN EN 14511

	Outdoor heat exchanger		Indoor heat exchanger	
	Outdoor Air T_{db} [°C]	Outdoor Air T_{wb} [°C]	T_{in} [°C] LT/MT/HT/VHT	T_{out} [°C] LT/MT/HT/VHT
Standard rating conditions	7	6	30 / 40 / 47 / 55	35 / 45 / 55 / 65
Application rating conditions	20	12	30 / 40 / 47 / 55	35 / 45 / 55 / 65
	2	1	*	35 / 45 / 55 / 65
	-7	-8	*	35 / 45 / 55 / 65
	-15	--	*	35 / 45 / 55 / 65
	12	11	*	35 / 45 / 55 / 65

3.6.3 Part 3: Test Methods

This section presents the most important requirements as the core challenges presented for this dissertation are based on them. The climate chamber, heat sink, data acquisition system and all the measuring equipment must perform well enough so that the measurements are valid and comply with this part of the standard and can therefore be used for equipment release.

Firstly, this part points to the method for calculating heating capacity of an air-to-water or air-to-air heat pump, by stating the following:

“The heating capacity of air-to-water, water-to-water heat pumps and liquid chilling packages shall be determined in accordance with the direct method at the water or brine heat exchanger, by determination of the volume flow of the heat transfer medium, and the inlet and outlet temperatures, taking into consideration the specific heat capacity and density of the heat transfer medium.”

Following the above statement, the heat capacity calculation formula is given by:

$$\dot{Q} = \dot{V} \times \rho \times c_w \times \Delta T \quad (3.8)$$

Table 3.5 contains the maximum uncertainty allowed for measurements performed on the liquid side (heat sink) and Table 3.6 for measurements on the air side (climate chamber), meaning that any value read and subsequently analysed must have the presented accuracy.

Table 3.5 – Measurement uncertainty for liquid side (water) measurements. Adapted from DIN EN 14511

Measured quantity	Unit	Uncertainty of measurement
Temperature difference	K	± 0.15 K
Temperature inlet/outlet	K	± 0.15 K
Volume flow	m ³ /s	± 1 %

Table 3.6 – Measurement uncertainty for air side measurements. Adapted from DIN EN 14511

Measured quantity	Unit	Uncertainty of measurement
Dry bulb temperature	K	± 0.2 K
Wet bulb temperature	K	± 0.4 K
Volume flow	m ³ /s	± 5 %

Table 3.7 presents the permissible deviations from the arithmetic mean of the measured values as well as for each single value in liquid measurements (heat sink) and Table 3.8 contains the same information for the air-side measurements. The arithmetic mean applies for cases where more than one sensor is used for measuring any value. In practical terms, this situation only applied to the air inlet dry bulb temperature measurement. This measurement was performed with four sensors placed at the air inlet of the evaporator, in accordance to this standard. These conditions represent a requirement for the climate chamber and the heat sink, which are responsible for controlling simulated outdoor conditions as well as indoor heat dissipation. The standard predicts an exception for cases where transient conditions are considered. In practical terms, this means that the boundaries in Table 3.9 apply during a defrost cycle as well as for the following 10 minutes after its termination.

Table 3.7 – Boundaries for deviations from set values during testing for liquid side. Adapted from DIN EN 14511

Measured quantity	Permissible deviation of the arithmetic mean values from set values	Permissible deviations of each of the individual measured values from set values
Inlet temperature	$\pm 0.2 \text{ K}$	$\pm 0.5 \text{ K}$
Outlet temperature	$\pm 0.3 \text{ K}$	$\pm 0.6 \text{ K}$
Volume flow	$\pm 1 \%$	$\pm 2.5 \%$

Table 3.8 – Boundaries for deviations from set values during testing for air side. Adapted from DIN EN 14511

Measured quantity	Permissible deviation of the arithmetic mean values from set values	Permissible deviations of each of the individual measured values from set values
Inlet temperature (dry bulb)	$\pm 0.3 \text{ K}$	$\pm 1 \text{ K}$
Inlet temperature (wet bulb)	$\pm 0.4 \text{ K}$	$\pm 1 \text{ K}$
Volume flow	$\pm 5 \%$	$\pm 10 \%$

Table 3.9 – Boundaries for deviations from set values during defrost cycles for air side. Adapted from DIN EN 14511

Measured quantity	Permissible deviation of the arithmetic mean values from set values	Permissible deviations of each of the individual measured values from set values
Inlet temperature (dry bulb)	$\pm 1.5 \text{ K}$	$\pm 5 \text{ K}$
Inlet temperature (wet bulb)	$\pm 1.0 \text{ K}$	— —

As it can be seen in the table above, requirements for dry bulb and wet bulb temperatures must be met. However, a wet bulb sensor was not available, so an equivalent relative humidity for lower and upper dry-bulb and wet-bulb temperatures was calculated using a Matlab toolbox. Applying the calculation method in Matlab for $T_{db} = 20 \text{ }^{\circ}\text{C}$ and $T_{wb} = 12 \text{ }^{\circ}\text{C}$, the following limits apply:

Table 3.10 – Calculated boundaries for relative humidity at $T_{db}=20 \text{ }^{\circ}\text{C}$ and $T_{wb} = 12 \text{ }^{\circ}\text{C}$

	$T_{db} \text{ [}^{\circ}\text{C]}$	$T_{wb} \text{ [}^{\circ}\text{C]}$	Equivalent RH [%]
Lower Limit	19.7	11.6	34.3
Upper Limit	20.3	12.4	39.8

Table 3.11 presents every relevant point with the respective relative humidity set point and tolerated range calculated.

Table 3.11 – Relative humidity equivalent values and boundaries for measurement points in DIN EN 14511

T_{db} [°C]	T_{wb} [°C]	RH [%]	RH range [%]
20	12	37.0	34.3 – 39.8
12	11	88.8	84.4 – 93.2
7	6	86.7	81.4 – 91.9
2	1	83.6	77.1 – 90.1
-7	-8	74.0	63.8 – 84.4
-15	--	--	--

Lastly, this part of the standard presents an extensive procedure for measuring predicting any possible scenario during measurement and what should the procedure be. This involves situations such as values going out of boundaries during *COP* measurement or undesired defrost cycle initiation (during pre-condition period, for example). It was assessed that in a worst case scenario, a test could take up to 5 hours, which is yet another motivation for automatizing the whole process as much as possible. The mentioned scheme is presented and explained in Annex 5.

Chapter 4

Climate Chamber Characterisation

4.1 Introduction

This chapter shows the work done as well as results and conclusions taken from the characterisation of the climate chamber available for the hybrid heat pump laboratory. This climate chamber provides an energy source for these appliances given that they combine an air-to-water heat pump and a boiler. Therefore, it must be capable of maintaining the temperature and humidity boundaries imposed by the European standard DIN EN 14511 established for air in the inlet of the evaporator.

Together with everything stated so far in this dissertation, this chapter will clarify the purpose of the work. Firstly, the goals and methodology for achieving such targets are explained, clarifying what was done and how it was done. Secondly, the obtained results are presented, including an uncertainty analysis of the sensors used. Finally, conclusions about the capability of the climate chamber for establishing the desired laboratory are drawn, as well as suggestions for potential improvements.

4.2 Goals

Given the described system, its current capabilities are not clear due to modifications and maintenance procedures done over the years. Considering that the boundaries imposed by DIN EN 14511 are tight, there is no certainty about the ability of a 30 years old hardware controlled system to maintain such conditions. As such, the goal of the work presented in this chapter is to verify whether the climate chamber available in the laboratory is capable of maintaining such conditions under several scenarios such as a defrost cycle as well as for a long period of time. This knowledge is also essential for the implementation of the measurement and data acquisition program, whose importance has been explained in the previous chapter. More specifically, the following topics were assessed:

- Measuring system verification – every sensor as well as the data acquisition had to be verified so that it was possible to assess the uncertainty of measurement and whether it is or not within the limits imposed by DIN EN 14511. In addition, an uncalibrated humidity sensor had to be used for timing reasons. Therefore, although this cannot be considered a calibration, a comparison between the measured values by this sensor and a calibrated one was done. Lastly, there was an offset between the set value for the climate chamber temperature and the actual achieved value, meaning that an offset curve had to be calculated so that it could be implemented into the measurement and data acquisition program;

- Continuous operation – the climate chamber behaviour was observed over a long period of time, so that it was possible to know if good enough conditions for testing are maintained for the required time. The standard procedures for different operating scenarios show that a test for a single operation point could take up to 5 hours. Measurements for different temperature, humidity and heat pump operation conditions were made
- Dynamic characterization – observation and further calculation were performed for the speed at which the chamber achieves stable conditions for testing when the set point is changed. With such knowledge, it is possible to program several tests at different set points, contributing to the automatization of the laboratory.

4.3 Methodology

Firstly, for the verification of the measurement system, technical documentation of the measuring devices was used to assess the uncertainty of measurement. This is presented on the next section. A curve and equation for the mentioned offset of the climate chamber was obtained by measuring the actual temperature for several points (as described below) and comparing it to the set value. The measured values by the uncalibrated humidity sensor were compared to a calibrated one for several temperature points and a subsequent curve relating these values was calculated. This way, for any measured value with the uncalibrated sensor could be converted into the correct value.

As the standard requires specific boundary conditions for the inlet area of the evaporator, it was decided only to analyse that zone instead of the whole space inside the climate chamber, which would require more time as well as more equipment. A physical setup of four temperature probes and a relative humidity probe was established. For a better understanding of the measuring points, two representations were drawn. Figure 4.1 represents the entire climate chamber from a top view, with the current placement of appliances being measured there. Two of these appliances are split units and the other is a compact one. The latest therefore contains a condenser, meaning that it exchanges water with the outside, whereas the split units exchange refrigerant which is sent to a condenser placed outside of the climate chamber. The unit used for the assessments made for this chapter was the one denominated “Split 2” which is standing in the middle. Figure 4.2 shows a rear representation of this unit, where the exact position of each probe can be seen. In both figures, the red dots represent temperature probes whereas the blue one represents the humidity probe.

For the tests where the heat pump was turned on, it was also important to measure data for electric power, heating capacity and *COP* of the heat pump. As such, water temperature at the inlet and outlet of the condenser was measure, as well as its volume flow.

Table 4.1 contains a summary of every test performed. For each test, the temperature and humidity of the above mentioned sensors was taken every second. The average temperature read by the four sensors at the inlet of the evaporator was then calculated for each second, and the maximum and minimum values over a big period of time of that average were determined. This data post processing was done in Excel.

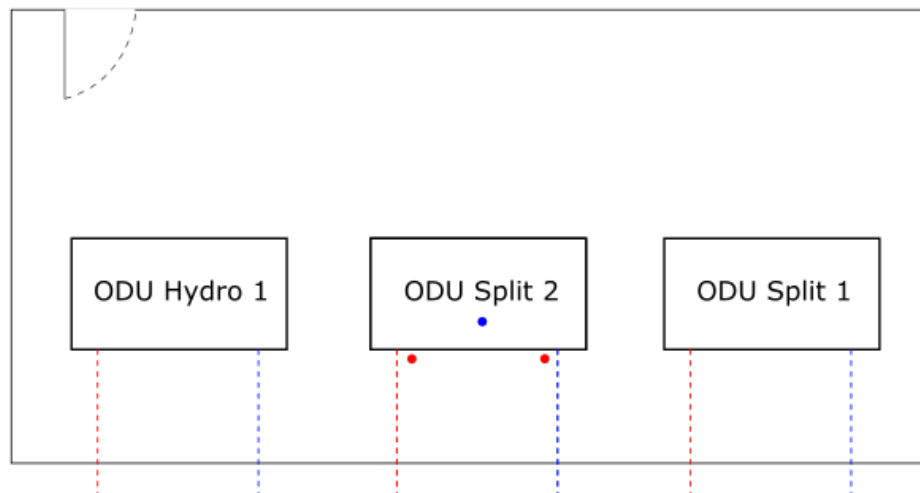


Figure 4.1 – Top view of the climate chamber setup. The blue dot represents the humidity probe and the red dots represent temperature probes.

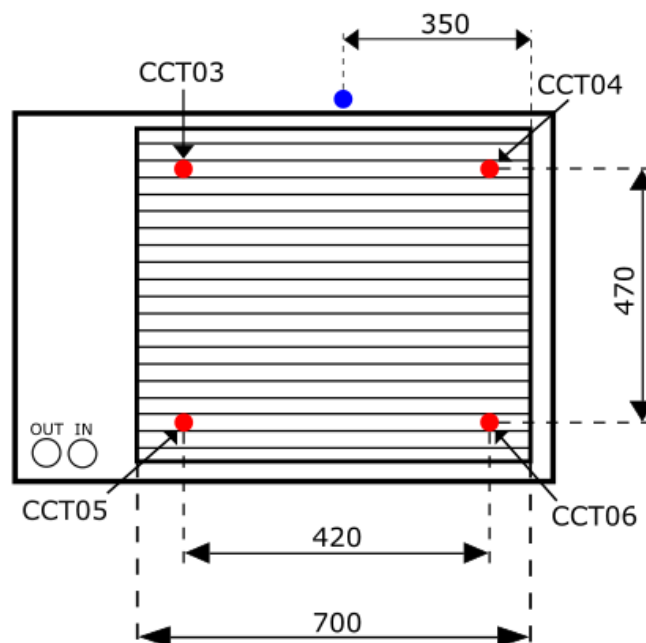


Figure 4.2 – Split 2 rear view, with representation of the sensors measuring air temperature around the evaporator entrance.

Table 4.1 – Performed tests conditions

Test number	Date	CC Mode *	Heat Load	Temperature [°C]	Relative Humidity [%]	Water temperature [°C]
1	2018/03/01	TC	None	12	--	--
2	2018/03/05	CC	None	7	86.7	--
3	2018/03/06	CC	None	2	83.6	--
4	2018/03/07	CC	None	12	88.8	--
5	2018/03/08	CC	None	-7	74.0	--
6	2018/03/09	CC	None	-10	74.0	--
7	2018/03/12	CC	None	16	90.1	--
8	2018/03/12	CC	Split 2 On	2	83.6	35
9	2018/03/13	CC	Split 2 On	2	83.6	65
10	2018/03/13	CC	Split 2 On	7	86.7	65
11	2018/03/14	CC	Split 2 On	-7	74.0	65
12	2018/03/14	CC	Split 2 On	12	88.8	65
13	2018/03/15	CC	Split 2 On	12	88.8	35
14	2018/03/15	CC	Split 2 On	7	86.7	35

*There are two different controlling modes for the climate chamber: TC (Temperature Control) – in this mode the climate chamber controls only the temperature; CC (Climate Control) – in this mode the climate chamber controls both humidity and temperature.

A Simulink program which controlled the desired set points and processed all the required into “.mat” files was already available. The “.mat” file contained a matrix with values from every sensor for each second. A similar program was developed for the heat sink and is described in the following chapter together with the data acquisition system. After every test, the collected data was converted into an Excel file and was there further converted into graphics which revealed the values behaviour over time and under different conditions. Initially, the performed tests had a duration of about sixteen hours because they were left running during the night. However, a feature was added to the Simulink program halfway through the test plan, which allowed the system to change set points after a certain duration. This meant that it was possible to program two tests to run during the night (more than two would not be feasible given that a minimum duration of six hours was required) and also observe the dynamic behaviour of the climate chamber between set points.

4.4 Results

4.4.1 Temperature off set curve and humidity sensor calibration curve

The obtained points and calculated curve for the offset, as mentioned in the previous sector, are represented in Figure 4.3. The chosen temperatures are the measuring points required by standard DIN EN 14511 (see Table 3.4), providing more relevant results for the application. These measured points are represented by the red dots. The curve equation is used latter in this chapter.

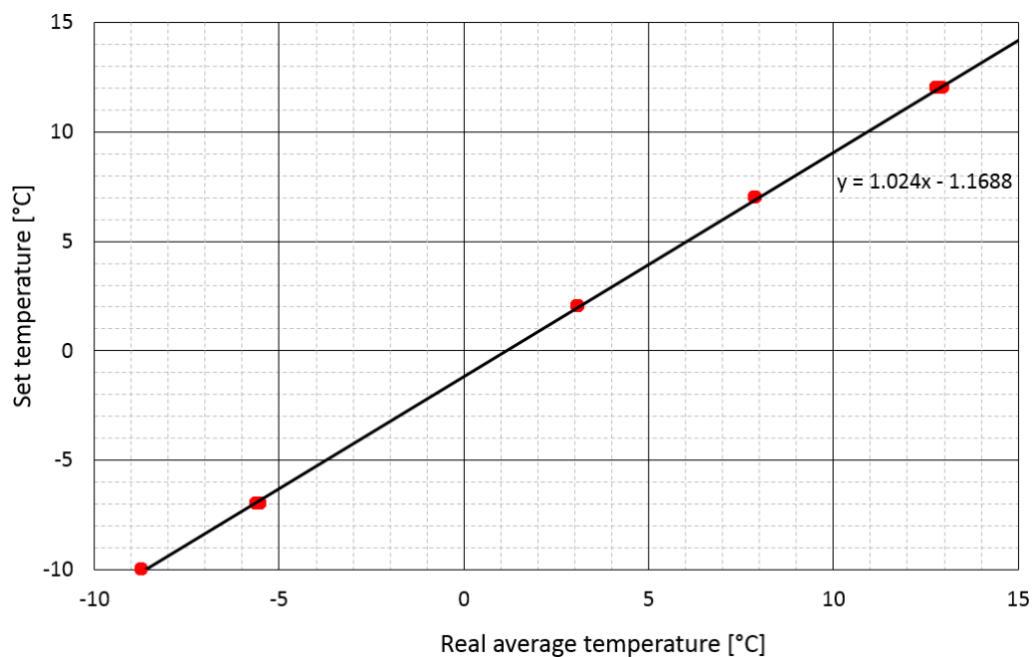


Figure 4.3 – Temperature offset curve obtained with measured values. The red dots represent the measured values and the dark line a linear approximation to those points.

As with the offset assessment, relative humidity points were measured at the set values required by the standard. However, this parameter oscillates more than temperature, providing more spread out results. The obtained graph and calculated curve are represented in Figure 4.4.

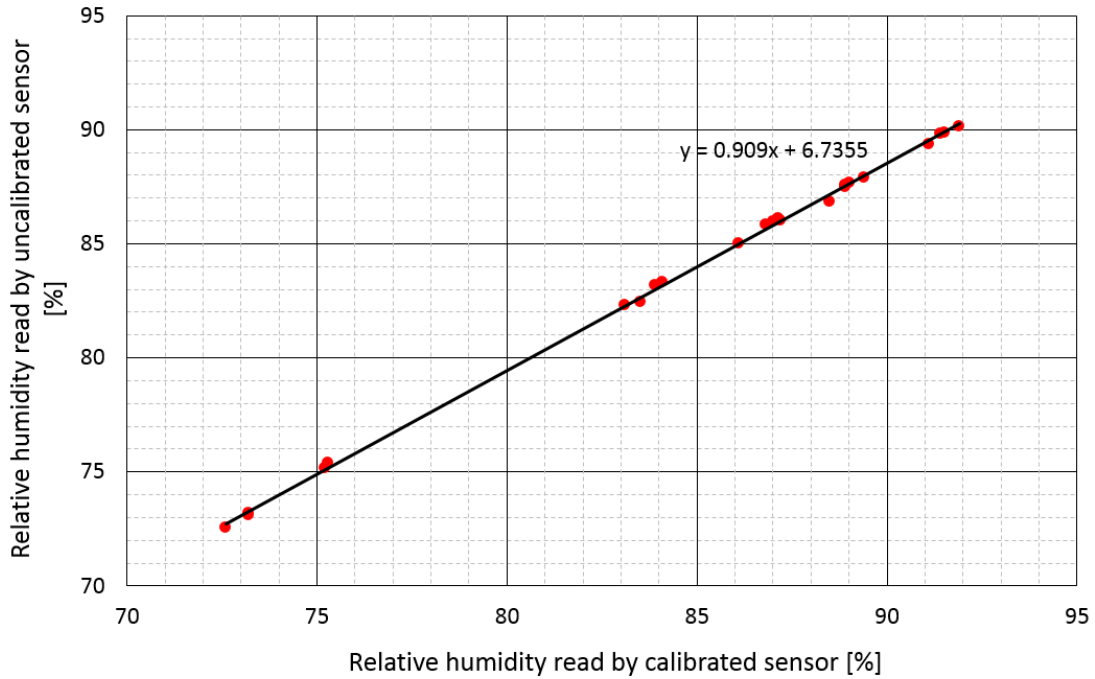


Figure 4.4 – Humidity sensor calibration curve obtained with measured values. The red dots represent the measured values and the dark line a linear approximation to those points.

4.4.2 Temperature analysis

Initially, the uncertainty of the temperature measurement had to be assessed. To do so, the manufacturer manual was consulted, where it stated that the PT100 had a measurement error in accordance to DIN EN 60751. Table 4.2 summarizes the useful information regarding measurement uncertainty contained in the mentioned standard.

Table 4.2 – Uncertainty of measurement provided by different classes of PT100 sensors.

Adapted from DIN EN 60751

PT100 Class	Measurement uncertainty [°C]
AA	$\pm 0.1 + 0.0017 \times T $
A	$\pm 0.15 + 0.002 \times T $
B	$\pm 0.3 + 0.005 \times T $
C	$\pm 0.6 + 0.01 \times T $
$ T $ – Absolute value of measured temperature, in °C	

It was then established that every PT100 used inside the climate chamber is Class B, meaning that the uncertainty is at least of ± 0.3 °C (for a 0 °C measurement and neglecting other factors such as wire length and A/D conversion). Therefore, no further assessment was done, since the permissible limit for releasement testing is ± 0.2 °C according to DIN EN 14511 (see Table 3.6).

Temperature at the air inlet of the Split 2 unit was measured with four probes: CCT03, CCT04, CCT05 and CCT06, as shown in Figures 4.1 and 4.2. The average of every temperature was calculated for each second and every value was further analysed and compared to the limits imposed by the standard DIN EN 14511. For the tests in which the heat pump was working (tests with heat load), the produced heat by the Split 2 unit was dissipated into an already existing heat sink, different from the one developed for this dissertation and presented in the next chapter. Although some tests had a duration of up to sixteen hours, results are always presented for the initial six hours so that the timeframe is consistent and given that a testing period larger than five hours is not required.

The results were firstly observed through graphs containing every temperature point obtained for the considered time frame. In terms of temperature behaviour, every test resulted in very similar outcomes. For this reason, only graphs for tests 5 and 9 (see Table 4.1) are presented. These were chosen because they represent the most demanding conditions for the climate chamber – lowest temperature of all the required operating points with humidity control (see Table 3.4) and highest temperature difference between heat source and heat sink in the case of tests with heat load. The remaining results can be consulted on Annex 6.

Figure 4.5 shows temperature over the entire considered time frame and Figure 4.6 shows a more detailed view of its behaviour during a stable period.

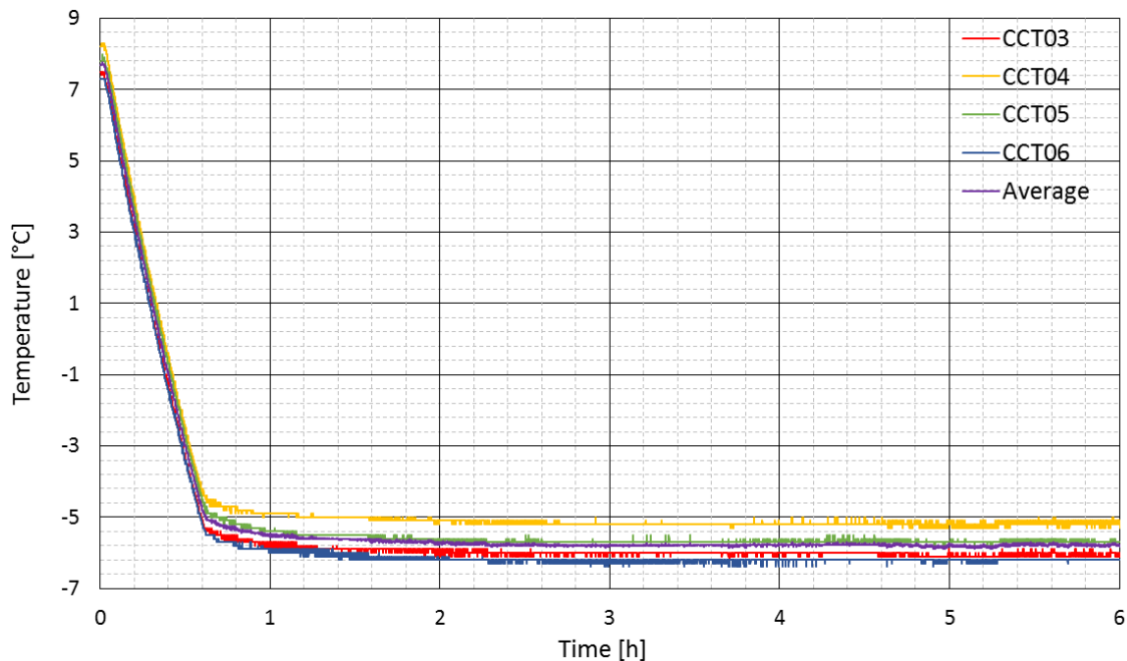


Figure 4.5 – Temperature values over a 6 hour period for 4 sensors and a set point of -7 °C and 74.0 % RH (Test 5).

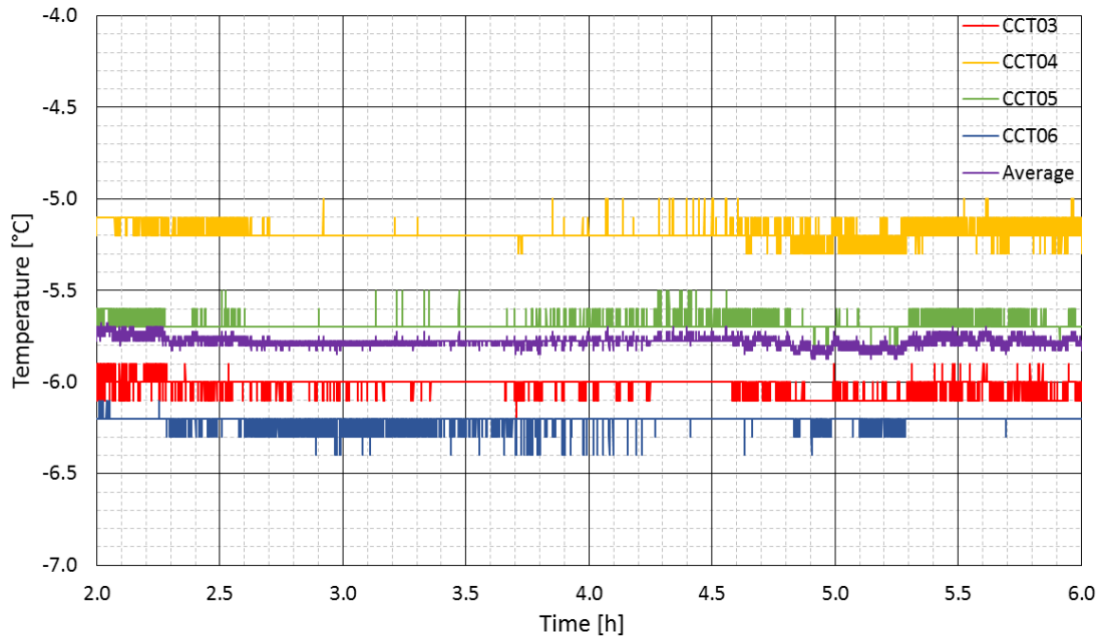


Figure 4.6 – Detailed view of temperature behaviour during stable period.

There are 3 points to be considered through the observation of these graphs:

- A two hour stabilisation period around the desired values must be considered, so only from there on is the obtained oscillation analysed;
- A consistent disparity between the read temperatures on different sensors can be observed. The reason for such difference lies on the position of the probes. The probes located on the right side (see Figure 4.2) read higher temperatures than the ones on the left due to the proximity to the fans of the climate chamber. One can also note that for the same side, the sensor on the top position reads a higher temperature. This is due to convective currents;
- The obtained values actually go off limits considering the set point (-7°C). However, it must be taken into account that the calculated set point curve had not yet been implemented into the control system, meaning that the obtained values have an offset.

Given the last mentioned point, another calculation had to be done to verify the validity of the test. Figure 4.3 shows the following empirically obtained equation:

$$T_{set} = 1.024 \times T_{real} - 1.17 \quad (4.1)$$

where T_{set} corresponds to the set temperature and T_{real} corresponds to the actual average air temperature at the inlet of the evaporator. As such, for $T_{set} = -7^{\circ}\text{C}$, one must consider the following real temperature:

$$T_{real} = \frac{-7.00 + 1.17}{1.024} = -5.69 \text{ }^{\circ}\text{C}$$

The offset is given by:

$$offset = -5.69 - (-7.00) = 1.31 \text{ }^{\circ}\text{C}$$

For an analytical assessment of the stability during the tests, a table containing the maximum, minimum and average value of each sensor as well as of the average temperature was built. The obtained offset was then subtracted from each of these values. They were further compared to the set point temperature. The resulting values are presented in Table 4.3.

Table 4.3 – Maximum and minimum limits and average for read temperature values during test 5 (see Table 4.1)

	CCT03 [°C]	CCT04 [°C]	CCT05 [°C]	CCT06 [°C]	Average [°C]
Maximum	-7.08	-6.18	-6.68	-7.28	-6.87
Minimum	-7.38	-6.48	-7.08	-7.58	-7.12
Average	-7.24	-6.39	-6.88	-7.42	-7.00
Deviation	0.38	0.82	0.32	0.58	0.13

The row denominated “Deviation” considers the maximum difference between the set temperature and the maximum and minimum values. The green colour means that the deviation is within the limits imposed by DIN EN 14511. Every test performed resulted in similar tables, where no deviation went above the allowed value.

Test 11 consisted in setting the air temperature to -7 °C and ask the heat pump for a 65 °C water flow temperature. Figure 4.7 shows temperature and heating power over the entire considered time frame and Figure 4.8 shows a more detailed view of their behaviour during a stable period. Heating power was calculated with the inlet and outlet water temperature at the condenser and with volume flow. These measurements are not relevant for this part of the work given that they were performed by an already established heat sink.

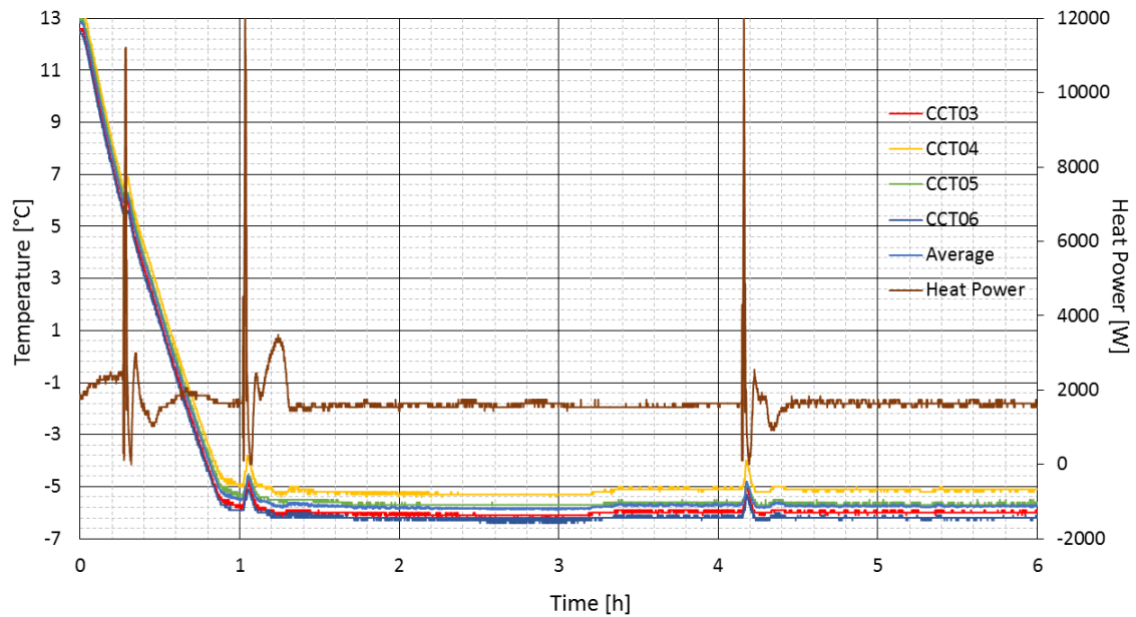


Figure 4.7 – Heat load and temperature values for 4 sensors over a 6 hour period. Set point of -7 °C and 74.0 % RH and water at 65 °C (Test 11).

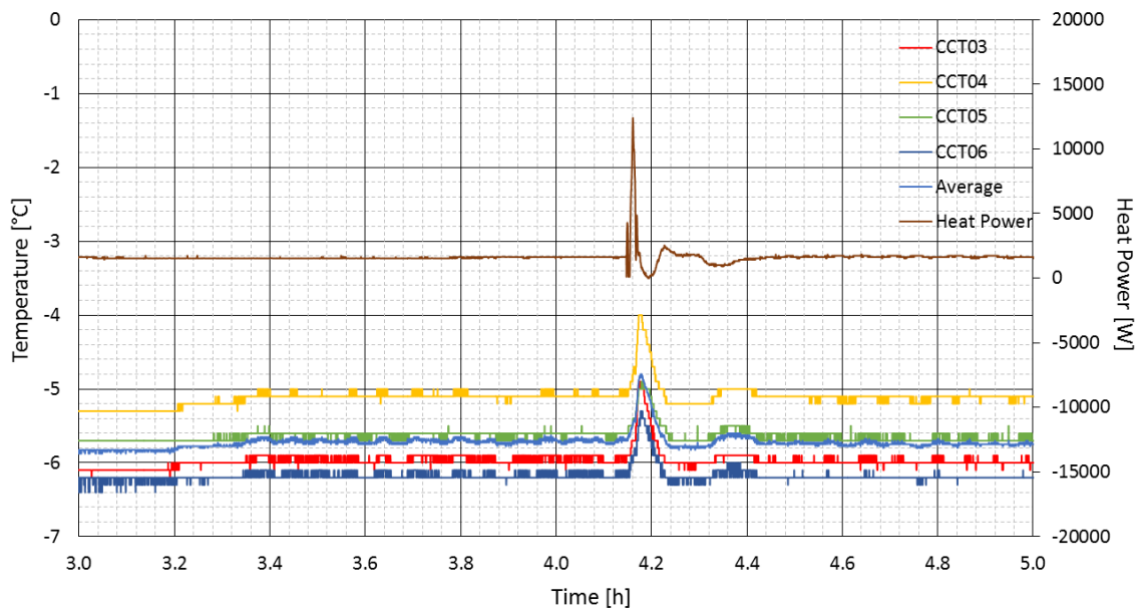


Figure 4.8 – Detailed view of temperature and heat load behaviour during stable period, including a defrost cycle (Test 11).

This test produced interesting results, from which the following points were assessed:

- Even though the conditions were very demanding for the heat pump with a heat source temperature of -7 °C and a heat sink temperature of 65 °C, the heating power output did not correspond to the maximum capacity of the appliance (it was less than 2 kW). This happened because the heat pump and the heat sink were trying to achieve

a flow temperature of 65 °C, which was never reached. As such, little heat was dissipated, so that the return temperature would be almost identical as the flow temperature, which is the behaviour of the control system when flow set point has not been reached yet. Therefore, the temperature difference between inlet and outlet was always small, implicating a low heat power output. The flow and return temperatures as well as their difference are represented in Figure 4.9;

- Due to the low temperatures inside the chamber and to the constant heat production, the heat pump went into defrost mode 3 times during this test. That is observed by the peaks of the heat output curve. These peaks are induced by the value of temperature difference (ΔT) which drops and rises as the cycle is reversed for injecting heat into the evaporator surface;
- The defrost cycles have implications on the temperature conditions inside the climate chamber. In a short period of time, the heat pump goes from removing to injecting energy from the surroundings, leading to a temperature rise, so much so that it seems that CCT04 goes out of boundaries (even considering the offset). However, as it was seen in the previous chapter, the boundaries change for transient conditions, including defrosting periods. As such, in this case the permissible deviation would be ± 5 °C (see Table 3.9).

For analysing the boundaries for this test, the same methodology as the previous one was used. However, the defrosting period was excluded due to the above mentioned reasons. The results can be seen in Table 4.4.

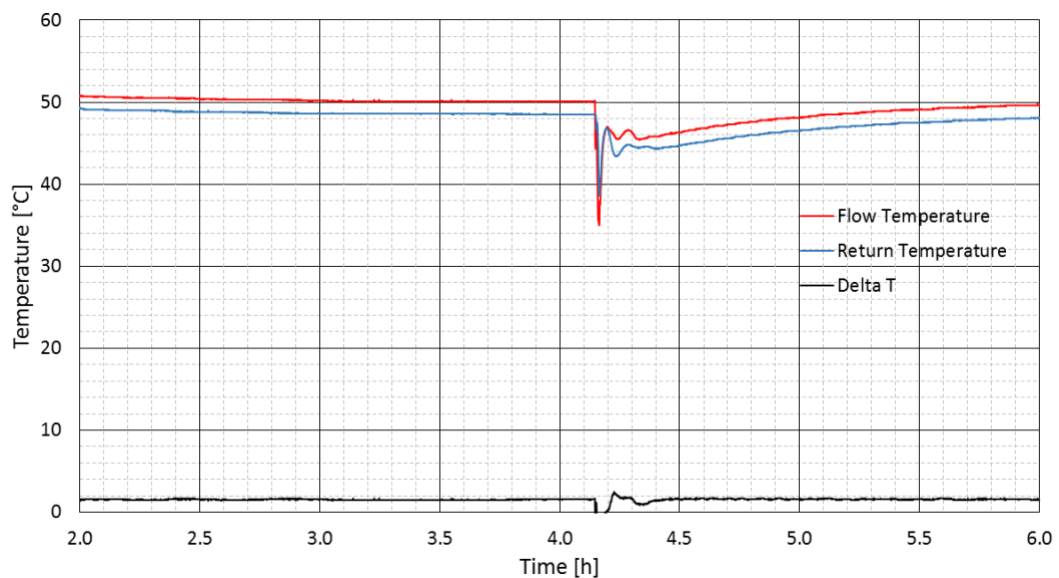


Figure 4.9 – Detailed view of flow temperature, return temperature and delta behaviour for a stable period, including a defrost cycle.

Table 4.4 – Maximum and minimum limits and average for read temperature values during test 11 (see Table 4.1)

	CCT03 [°C]	CCT04 [°C]	CCT05 [°C]	CCT06 [°C]	Average [°C]
Maximum	-7.1	-6.2	-6.7	-7.3	-6.9
Minimum	-7.4	-6.6	-6.9	-7.6	-7.1
Average	-7.3	-6.4	-6.9	-7.4	-7.0
Deviation	0.4	0.8	0.3	0.6	0.1

It is also relevant to analyse more thoroughly the climate chamber behaviour to a defrost cycle while cooling as part of the dynamic behaviour analysis, which was a case provided by this test, as seen in Figure 4.7. Figure 4.9 shows a more detailed representation of the average temperature during this period. The temperature raised 0.3 K and returned to cooling after about 110 s.

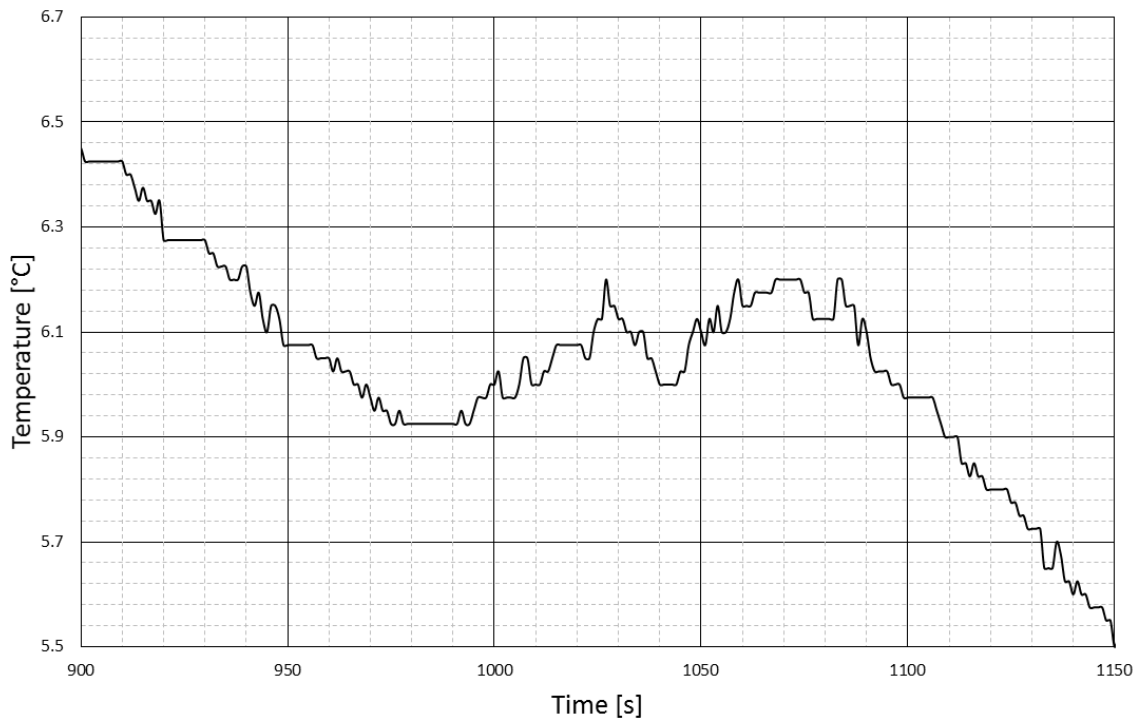


Figure 4.10 – Average temperature over time during a defrost cycle and climate chamber cooling.

4.4.3 Relative Humidity analysis

As with temperature, the uncertainty of measurement must be first assessed. The uncertainty provided by the manufacturer can be considered the absolute uncertainty because the equipment comprises the whole measurement system (including data acquisition), meaning that the relative humidity value is provided. The manufacturer claims the values in Table 4.5 for uncertainty of measurement.

Table 4.5 – Uncertainty of measurement claimed by the manufacturer of the humidity sensor for different ranges

Range [% RH]	Uncertainty of measurement [% RH]
0-90	± 1.0
90-100	± 1.7

The relative humidity measurement over time was done in order to assess whether the boundaries for wet bulb temperature are being maintained. The equivalence calculation was shown in the previous chapter. It can be seen in Table 3.12 that the allowed range gets bigger when the temperature is lower. As with the temperature stability analysis, an initial observation through graphical representation of the values over time has been done and it is shown in Figure 4.10 for the tests without heat load and in Figure 4.11 for the tests with heat load.

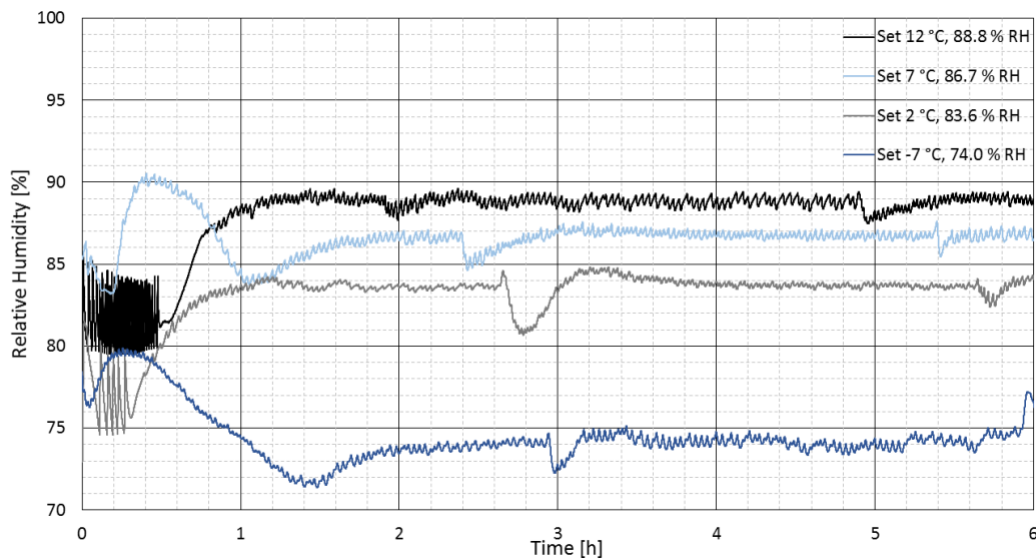


Figure 4.11 – Obtained humidity values over time for tests performed without heat load.

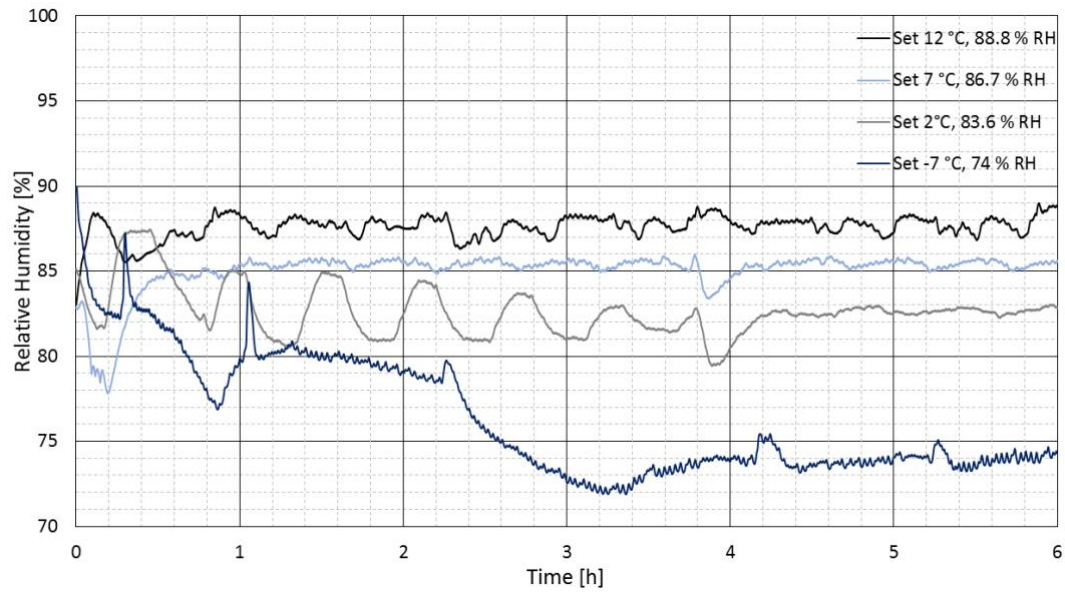


Figure 4.12 – Obtained humidity values over time for tests performed with heat load.

Considering the given intervals in Table 3.12, it is easily observable in the presented graphs that humidity does never go out of boundaries, being therefore unnecessary to build similar tables as for the temperature analysis. It could also be assessed that the relative humidity stability was better for higher temperatures and that the heat pump operation produced many oscillations due to the heat removal from the climate chamber (and injection during defrost cycles).

4.5 Conclusions

Regarding the data acquisition system, two problems were identified. Firstly, the humidity sensor was not calibrated at the time of testing, and the manually obtained calibration curve is not scientifically valid to support the use of this sensor when assessing humidity stability for release measurements. However, the sensor was later sent back to the manufacturer for recalibration, which together with the claimed uncertainty of measurement enables it to be fit for testing according to DIN EN 14511. Secondly, the PT100 temperature sensors inside the climate chamber, which according to the manufacturer are in compliance with the standard DIN EN 60751, are classified as class B sensors, meaning that temperature is being read with more than $\pm 0.3\text{ }^{\circ}\text{C}$ of accuracy, without even considering the data acquisition system which carries up to $\pm 0.2\%$ of the full scale value according to the manufacturer of the PLC to which the sensors are connected. Therefore, considering the maximum uncertainty imposed by DIN EN 14511 of $\pm 0.2\text{ }^{\circ}\text{C}$, the current data acquisition system does not fit the requirements for release measurements due to the temperature measurement equipment. In order to fulfil these requirements, the current available sensors inside the climate chamber would have to be replaced for Class AA PT100.

In terms of dynamic behaviour, a two hour period for stabilisation before heat pump measurements start is recommended upon changing the test conditions. The relative humidity values measured for a set point of $-7\text{ }^{\circ}\text{C}$ with heat load represented in Figure 4.11 (bottom curve) is a good example for this stabilisation period requirement. Although relative humidity was rapidly reaching to the desired set point, a defrost cycle interrupted this process and destabilized the system.

As for stability during continuous operation, relative humidity values were comfortably maintained within boundaries, whereas temperature reached closer to the limits, although these were not surpassed under no circumstance. It must be underlined that the read values diverged as far as $0.9\text{ }^{\circ}\text{C}$ from the set values. Considering that the limit imposed by DIN EN 14511 is $1.0\text{ }^{\circ}\text{C}$ and that the measurement was performed with an accuracy greater than $\pm 0.3\text{ }^{\circ}\text{C}$, this result is not conclusive in terms of standard compliance, as the real value could overcome the limits.

All in all, the observed behaviour of the climate chamber transcended the expectations, since such a stability was not expected, especially for low temperature set points. Considering that the maximum observed temperature deviation (excluding defrost periods) was of $0.9\text{ }^{\circ}\text{C}$ and that relative humidity values did not present an issue, it can be concluded that this climate chamber is currently fit to carry tests for internal assessments, even though the measurement system does not comply with DIN EN 14511. In order for valid release measurements to be performed on heat pumps and hybrid heat pumps using this climate chamber as a heat source, the existing

temperature sensors would have to be replaced with Class AA PT100, and the presented temperature measurements would have to be repeated so that the required stability could be proven whilst using adequate sensors. There is space for modification of the sensors positioning, in case it is necessary to achieve satisfying values.

Financially speaking, the cost of applying the mentioned modifications in order to achieve standard compliance must be weighed against the cost of outsourcing the release measurements, together with other aspects not related to measuring. The bottom line is that the climate chamber is capable of maintaining testing conditions to assess the performance of an appliance before it is sent to an external laboratory, meaning that the outcome of the release measurements can be predicted using this equipment as a heat source.

Chapter 5

Heat Sink Establishment

5.1 Introduction

The heat source for an air-to-water heat pump has been characterised. For the second part of the practical work done for this dissertation, a heat sink for the same type of appliances is established. It must dissipate heat from the hot water leaving the heat pump, ultimately in the same way as in a real-life scenario. While the characterisation described in the previous chapter was only about the observation of the climate chamber behaviour and further analysis, the tasks set for the heat sink were more complex, as they involve programming the control of the heat sink. The ultimate goal was to make this heat sink work in such a way that it can simulate the complex dynamic energetic behaviour of a home and with stable enough conditions so that it can even test appliances in accordance to DIN EN 14511. However, it would be an unrealistic prediction to achieve such targets within the time frame of this dissertation, so the aim was to study it and fully characterise its behaviour together with its components (mainly pumps and valves) as well as building a first version of a new control software operated via Simulink, which could then be used to make further studies until eventually the device gets ready for any kind of testing with heat pumps. In the end of this work, the goal is to have objective conclusions about the achieved results followed by meaningful suggestions about the next steps to take regarding this testing device. Is it worth to carry on its development? Or would it be too expensive to change it in such a way that it suits the laboratory needs? These are the sort of questions that must be answered. This work is therefore the first iteration for the full establishment of such a test rig.

This chapter is divided into three main parts: an assessment of the available hardware available on the physical setup together with an analysis of the entailed capabilities followed by a thorough explanation and establishment of the whole control system, including the built software and achieved results with it. Everything is summed up in the conclusion of this chapter, where an overview of the heat sink's current situation is explained and insight for the next steps regarding its development is given.

5.2 Physical Setup

The equipment comprising the heat sink can be divided into two parts – hydraulic and electric. The hydraulic includes every pipe and component where the heat medium (water) will circulate and perform energetic exchanges. The electronic part can be summed up by being a PLC (Programmable Logic Controller) composed of several slots where the components are connected as well as a basic safety circuit. This PLC system was used to modulate pumps and valves in such a way that the desirable energetic conditions were met. The control of this system is described on the next section.

The first task regarding the heat sink establishment was to assess the hydraulic scheme of the available equipment and it was quickly observed that it offered many possibilities in terms of operating modes. In order to start this whole establishment task with a simple step, the hydraulics were manually manipulated (by closing valves, for instance) in such a way that the achieved setup was simple to operate. It is represented in Figure 5.1 and will be the base for the work presented in this chapter.

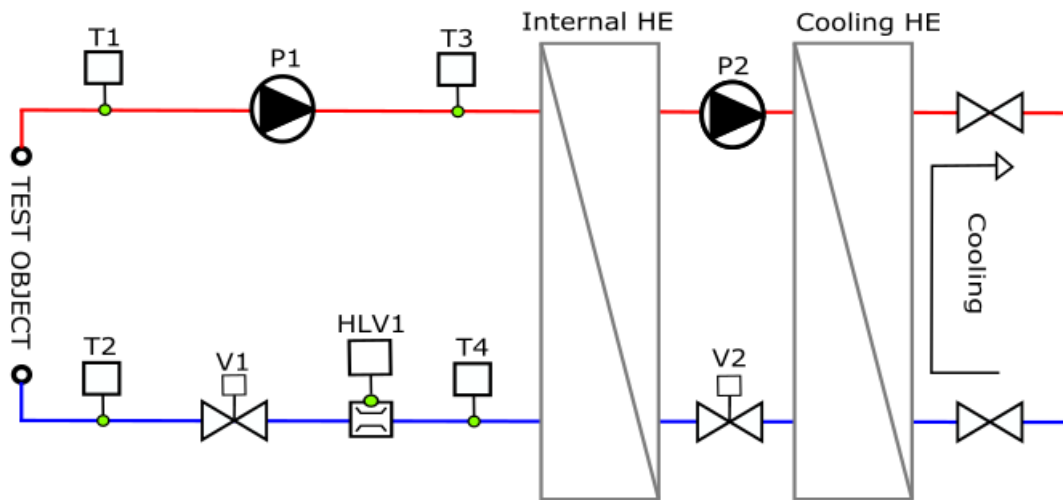


Figure 5.1 – Heat sink hydraulic scheme. Simplified from the original present in Annex 7.

The electrically connected components represented in the hydraulic scheme in Figure 5.1 are listed in Table 5.1.

Table 5.1 – List and description of the hydraulic components used for the present work

Name	Type	Description
T1	PT100 Temperature Sensor	Measures flow temperature from the test object
T2	PT100 Temperature Sensor	Measures return temperature to the test object
T3	PT100 Temperature Sensor	Measures flow temperature at the entrance of the internal heat exchanger
T4	PT100 Temperature Sensor	Measures flow temperature at the exit of the internal heat exchanger
HLV1	Magnetic induction flow meter	Measures volume flow going to the test object
P1	Circulation Pump	Can be modulated to increase or decrease power and subsequently volume flow.
P2	Circulation Pump	Can be modulated to increase or decrease power and subsequently volume flow.
V1	Modulating control valve with magnetic actuator	Can be modulated to a more closed or opened position, inducing different levels of pressure lost in the primary circuit, subsequently changing the volume flow.
V2	Modulating control valve with magnetic actuator	Can be modulated to a more closed or opened position, inducing different levels of pressure lost in the intermediate circuit, subsequently changing the volume flow.

Before going into further detail about the physical setup, the measurement uncertainty must be regarded. The temperature sensors used in this setup are the same model as the ones inside the climate chamber, whose minimum uncertainty has been assessed in the previous chapter (see Table 4.2). As such the current measurements carry an uncertainty more than two times larger than the one required by DIN EN 14511 for release measurements ($\pm 0.15\text{ }^{\circ}\text{C}$). As for the flow meter, it was firstly verified that it is installed according to the instructions on the manual, as it respects position and distance to nearest bend requirements. As such, the uncertainty of the measured value can be considered the one provided by the manufacturer as the equipment includes the data acquisition system and transmits the actual value. The manufacturer claims an accuracy of $\pm (0.02\text{ \% m.v.} + 1\text{ mm/s})$ for the considered model, where m.v. means “measured value”.

The accuracy refers to flow velocity because the sensor measures the voltage induced by the magnetic field of the flow and converts it to velocity with the following mathematical relation:

$$U = v \times B \times k \times D \quad (5.1)$$

where U is the measured voltage, v is the flow velocity, B is the induced magnetic field, k the and D the pipe diameter. The velocity is further converted into volumetric flow under the following relation:

$$\dot{V} = v \times A \quad (5.2)$$

where $\dot{V}$ is the volumetric flow and A the pipe area. So, the accuracy can be translated to volumetric flow by converting 1 mm/s to l/h for this pipe diameter, which is 25 mm. Therefore, the correspondent volumetric flow is given by:

$$\dot{V} = 0.001 \times \frac{\pi \times 0.025^2}{4} = 4.9 \times 10^{-7} \text{ [m}^3\text{/s]} = 1.8 \text{ [l/h]}$$

The accuracy of flow measurement is $\pm(0.02 \% m.v + 1.8 \text{ l/h})$, which is enough considering the requirements (see Table 3.5).

It can be seen in Figure 5.1 that the heat sink is composed by three circuits hydraulically separated by two heat exchangers. Starting from the right, a cooling circuit can be found where cold operating water from the nearby river Lahn (roughly at 15 °C, depending on the time of the year) circulates to a heat exchanger, extracting heat from the intermediate circuit, and has no electrically controllable components. The intermediate circuit acts as a controlling medium by modulating the volume flow going through both heat exchangers, which can be done through the circulating pump P2 or the modulating control valve V2. However, P2 was not connected to the PLC module, so only valve V2 was modulated to control the volume flow in this circuit. The intermediate circuit transfers heat from the primary circuit to the cooling one. The primary circuit is hydraulically connected to the test object, which could be any kind of water heat generator, and should behave as a real central heating system of a house. Therefore, the main control operations both in terms of inputs and outputs occur in this circuit. It has four temperature sensors, one volume flow meter, one circulation pump and one modulating control valve. As it can be concluded, the only directly controllable parameter is volume flow. For testing, it is desirable to control the return temperature T_2 and ΔT (difference between T_1 and T_2) which can be indirectly achieved by modulating the volume flow. The principles for controlling this parameters will be described in the next section.

Before proceeding to the development of a control program, the hydraulic behaviour of the system had to be assessed, i.e., comprehending how volume flow varies with the modulation of the pump P1 and valve V1. This was only possible to assess in the primary circuit since there was no volume flow measurement in the remaining ones. It must be pointed out that the results are only valid for the considered hydraulic setup and for the connected test object. If a different test

object is connected, the induced pressure drop will change, consequently imposing a different system characteristic curve. However, the main goal is to implement the methodology and such an analysis can be quickly done every time a new test object is connected.

Firstly, an analysis of the pump modulation was performed. As seen in equation 3.1 and 3.2, the heating power depends on the volumetric flow as well as in the temperature difference. By increasing the pump modulation, the volumetric flow rises, which for a constant temperature difference implies a higher heat power. Since it is desirable to maintain ΔT , this variable was fixed to different values in order to get a direct correlation between $\dot{V}$ and $\dot{Q}$. Developing on the heat equation 3.1 and 3.2, and considering $\rho = 1000 \text{ kg/m}^3$ and $c_w = 4.180 \text{ kJ}/(\text{kg} \cdot \text{K})$, the following relation was obtained:

$$\dot{V} = \frac{\dot{Q}}{4180 \times \Delta T} \quad (5.3)$$

The volume flow was measured for successive modulating signals with an increment of 5 % between each point. The values were obtained ten times and then averaged for each point. In an Excel table, the modulation value could then be converted in an equivalent heating power for different ΔT . The results from this assessment are shown in the graph contained in Figure 5.2

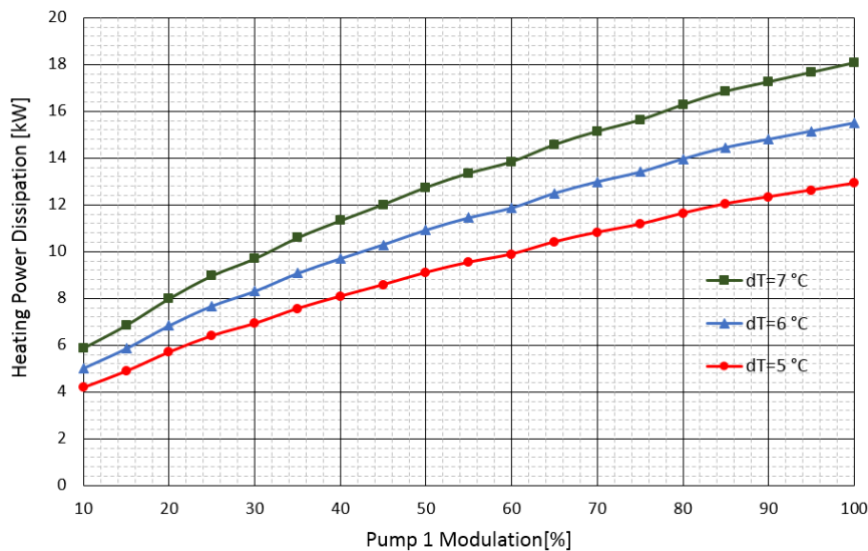


Figure 5.2 – Relation between heating power dissipation and pump modulation for different ΔT . Note that dT is equivalent to ΔT .

It can be observed that for the connected test object and for a $\Delta T = 5 \text{ }^\circ\text{C}$, there is a capacity for controlling 4 to 13 kW of heating power with modulation of the pump P1 alone. A modulation under 10 % is not possible, as it will provide the same power, until it hits 0 %. It can also be

concluded that the volume behaviour is roughly homogeneous throughout the whole modulation range, being therefore favourable for achieving stable conditions with the further described control. In order to assess the best way of controlling volumetric flow on this circuit as well as to analyse the possibility of obtaining flow values smaller than the ones obtained for P1 modulation of 10 %, measurements regarding valve V1 behaviour were performed.

Valve V1 has a magnetic actuator, which allows it to be electrically modulated. A 0 V signal will close it completely, blocking the entire flow to the test object, and a 10 V signal corresponds to the widest opened position possible. In the same way as the previous test, flow was read 10 times for each modulation value and an average value further calculated. The measurement was done while P1 operating at 10 % and 100 % in order to observe a potential difference. Therefore, the result is presented in terms of percentage of flow, where 100 % is the flow obtained when the valve is fully opened (100 % modulation). The representation of such results is contained in the graph of Figure 5.3.

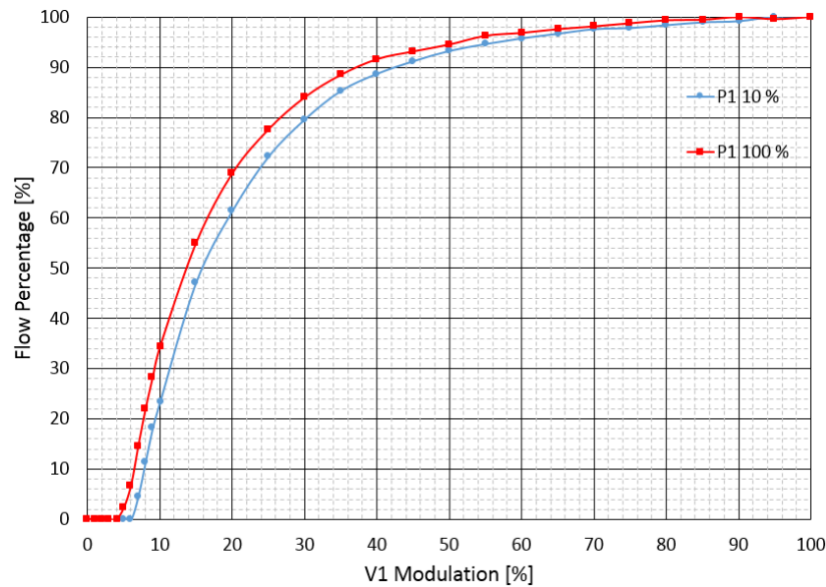


Figure 5.3 – Relation between valve V1 modulation and decrease/increase of volume flow.

A steep drop of flow is observed for modulation values inferior to 30 %, whereas for values above the variation is quite small, as the flow between 40 % and 50 % of valve positioning differs only in roughly 2 %, for example. As such, the conclusion drawn is that the control of volumetric flow control using V1 modulation should be avoided since it shows distinct variations for different modulation ranges. The control modules used are PID based, which are explained in the next section. However it is relevant to mention it, because these type of controllers must be tuned according to the system. As such, a fine tuning cannot be achieved for the entire modulation range with such a heterogeneous behaviour. Furthermore, a small variation of the modulation between 10 % and 30 % induces a large change in flow rate, meaning that the system could likely

oscillate around the desired value without never achieving it. It is believed that the obtained curve is a result of the shape of the physical actuator, which does not block equal percentages of the flow area as it is being closed. When the electric module of the valve is off, the valve gets completely closed, as this is the position with no induced force. As such, the valve must always be actuated when P1 is on, so that it allows water to flow to the test object.

5.3 Control system

The physical setup of the heat sink has been described, together with the capabilities it provides. In this section, the control system is described in detail, from the existing data transfer mechanisms to the actual creation of a Simulink program which modulated the hydraulic components in order to achieve the desired set points.

Before going into detail about the control setup itself, it is important to explain the energetic principles behind it. The heating power provided by the test object is given by the following equation:

$$\dot{Q}_{TO} = \dot{V}_1 \times c_w \times \rho \times (T_1 - T_2) \quad (5.4)$$

where $\dot{Q}_{TO}$ is the heating power provided by the test object in kW, $\dot{V}_1$ is the volumetric flow in the primary circuit in m^3/s , c_w is the specific heat value for water in $\text{kJ}/(\text{kg} \cdot \text{K})$, ρ is the water's density in kg/m^3 and T_1 and T_2 are the flow and return temperature in K respectively, as seen in the hydraulic scheme in Figure 5.1. For the same $\dot{Q}_{TO}$, c_w and ρ , a modulation of $\dot{V}_1$ will change the difference between T_1 and T_2 (ΔT), according to the following equation:

$$\dot{V}_1 = \frac{k}{\Delta T} \quad (5.5)$$

where k is given by:

$$k = \frac{\dot{Q}_{TO}}{c_w \times \rho} \quad (5.6)$$

Therefore, for the controlling ΔT to the desired set point, pump P1 is modulated accordingly. If ΔT is lower than it should, $\dot{V}_1$ has to be reduced so that ΔT rises since they follow an inverse relation. In reality, this volumetric flow modulation to the test object will only modify T_1 (flow temperature), as the return temperature depends on the amount of heat dissipated to the intermediary circuit. As such, larger the amount of water sent to the heater, the less its

temperature will be risen. The heating power dissipated on the intermediary heat exchanger is given by:

$$\dot{Q}_1 = \dot{V}_1 \times c_w \times \rho \times (T_3 - T_4) \quad (5.7)$$

where T_3 and T_4 are the temperatures read by the sensors according to the hydraulic scheme represented in Figure 5.1, i.e. at the inlet and outlet of the heat exchanger. However, $\dot{Q}_1$ actually depends on the heat absorbed by the intermediary circuit through the following relation:

$$\dot{Q}_1 = \eta_{HE} \times \dot{Q}_2 \quad (5.8)$$

where η_{HE} is the efficiency of the heat exchanger and $\dot{Q}_2$ the heat power absorbed in the intermediary circuit and consequently by the heating circuit. This means that if V2 is completely closed, the volumetric flow in the intermediary circuit will equal 0 and therefore $\dot{Q}_1$ will also equal 0. If these conditions were maintained with heating generation from the test object, the circuit temperature would rise to a point where the test object would shut off for security reasons. Therefore, if $\dot{V}_2$ rises, $\dot{Q}_1$ will rise accordingly. However, $\dot{V}_1$ depends on the modulation of P1 and T_3 on the set flow temperature (it is closely equal to T_1), which has been already explained. As such, modulating the flow in the intermediary circuit will have a direct impact on T_4 , as the remaining parameters cannot be changed. As stated before, the only available way of controlling volumetric flow in the intermediary circuit is to modulate valve V2, so as this valve is set to a more open position, $\dot{V}_2$ rises and T_4 decreases. The phenomenon here exposed will be proven later in this section.

5.3.1 Data system architecture

In this section, an explanation about the existing data system is provided, so that the reader can understand better where the Simulink program for controlling this heat sink fits and why it is needed. This program takes in the input from the measurement equipment available in the hydraulics, and processes the signal output to the actuators (pumps and valves) in function of the mentioned inputs as well as the settings selected by the user. In reality, the designed program in this work will be integrated in a more complex one which fully controls and reads parameters from the entire measurement system in the laboratory (including the climate chamber, heat pump and boiler). As such, a central data server application (i.e. software) that supports several communication protocols must establish the interface between the controller (Simulink) and the physical setup. Between the data server and the physical components there is usually a PLC which

processes data from sensors or to actuators, being the interface between these components and the data server. A representation of the described data architecture was drawn so that it can be easily understood. It should be mentioned that both the explanation and scheme do not go into detail about technical aspects of this system, such as data protocols since it is not relevant in the scope of this dissertation. The arrows can be considered as information exchange, even if the information is exchanged in different protocols or methods. Given that the present work was done to establish the heat sink, a more detailed view of that system was drawn. Therefore, one must also note that the squares representing the climate chamber and the heat pump entail complex system, which depend on the type of hardware and are not further described here. Figure 5.4 shows the mentioned representation for the data architecture system. The internal data server was given the name of TDS, which means Tiny Data Server and will be from here on called accordingly.

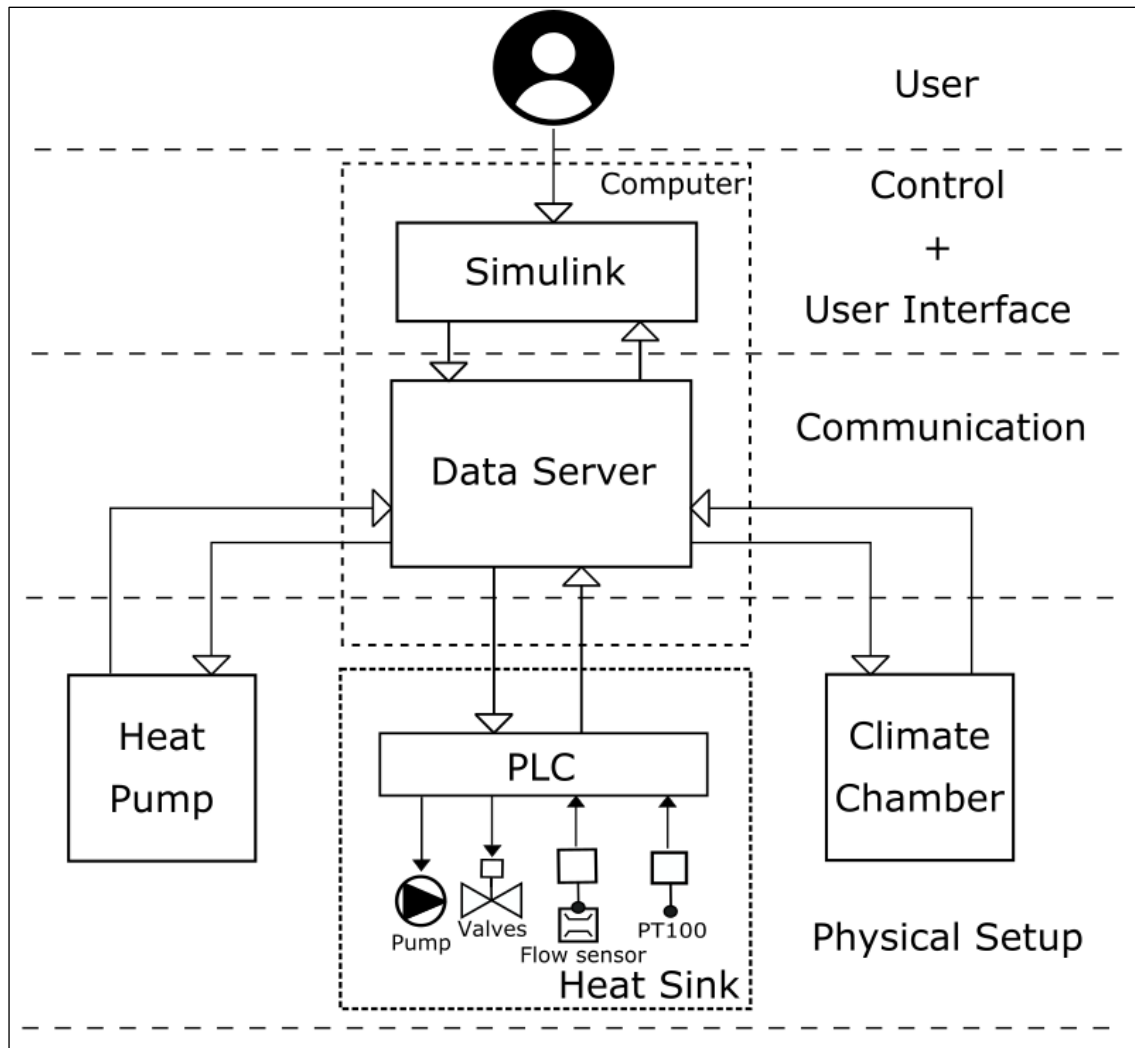


Figure 5.4 – Visual representation of the data exchange mechanism used in the laboratory.

The PLC can be programmed so that it controls and processes data from its inputs and to its outputs in a desired way, as its name suggests. Initially, the heat sink was controlled directly with the PLC and so its pumps and valves performed accordingly to the modulation processed inside the PLC. However, this program was not documented, so there was no certainty about how the system behaved under different conditions. For this reason, and in order to integrate the heat sink in the global data system represented in Figure 5.4, the whole thing was disregarded. Therefore, as the PLC must always be controlled by a program (which in this case is done in Codesys), a new one was done in which the instructions are purely about processing the data from and to TDS, so that the actual control is done by Simulink. The PLC acts as the master for the electrical configuration, meaning that its signals can overwrite any signal coming from the data server, so some safety features were also added into the new Codesys program. The relevant features for this work, considering the presented hydraulic scheme are the following:

- If V1 is set to 0 % (fully closed), P1 must be set to 0 %. This prevents potential damages on P1;
- V2 has a minimum signal set of 1 %. As P2 is constantly running, a minimum flow must be guaranteed by never fully closing V2.

Now that the control principles and the data system architecture have been described, the developed Simulink can be presented in the following section.

5.3.2 PID control fundamentals

In order to control the signal output of the valve and pump modulation, PID controllers were used inside Simulink, as will be further shown during the program's description. In spite of not being the focus of the performed study, an overview about these controllers should be done, so that the control program itself can be better understood.

Firstly, a representation of the control loop was drawn in order to elucidate the reader about the basic control operation performed. As the user defines the set values and the program receives input from the installed sensors, an error value is obtained (difference between set value and read value) and fed to the controller, where the output is generated and sent back to the physical setup. The data mechanisms for these information exchanges have already been described in the previous section. In the figure, the “Process” represents the whole hydraulic and energetic phenomena occurring within the physical setup.

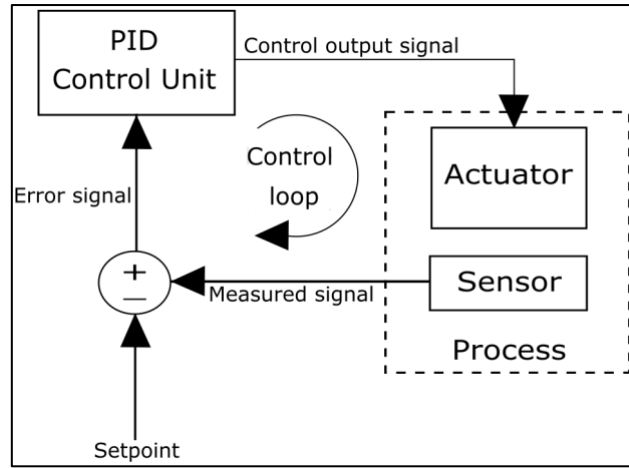


Figure 5.5 – Control loop representation.

PID stands for Proportional-Integral-Derivative and according to Ogata (1997) its signal output $C(t)$ is given by the sum of 3 terms, as shown in the following equation:

$$C(t) = K_p \times e(t) + K_i \times \int_0^t e(t) \cdot dt + K_d \times \frac{de(t)}{dt} \quad (5.9)$$

where $e(t)$ is the error signal and K_p , K_i and K_d are the proportional, integral and derivative gains respectively. A correct tuning of such parameters must therefore be achieved so that the control unit reacts correctly to the dynamic behaviour of the system. An example of the output result from such systems was taken from one of the performed tests and it is presented on Figure 5.6.

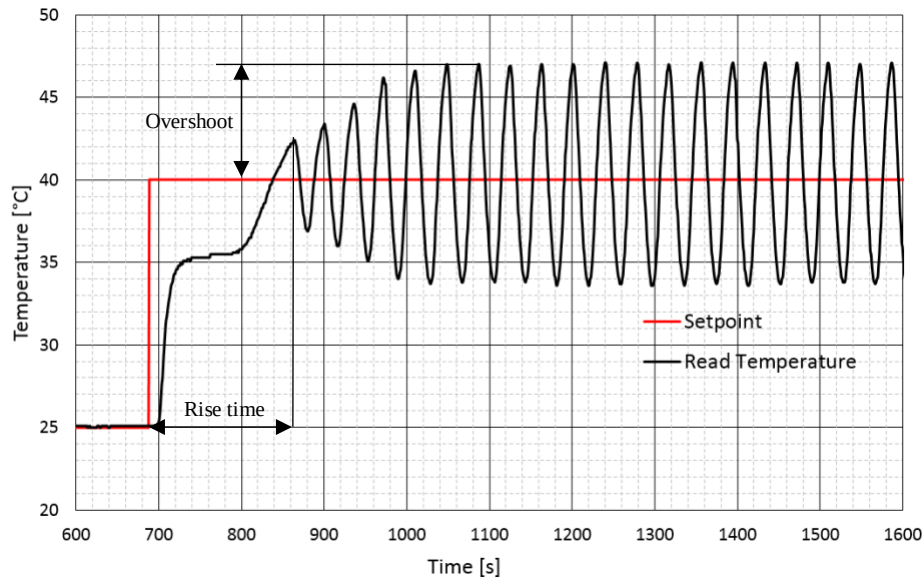


Figure 5.6 – Control behaviour observed during testing.

The presented example actually accounts for an incorrect tuning of the parameters for the considered system, since an oscillation of about 15 % of the set point is observed. However, it is a clear illustrative example of the behaviour of such control systems. In order to obtain fit values for each gain, previously tuned controllers were adapted. In case of the pump, an already existing PID controller for a circulation from another rig was used, the behaviour was then observed by building a similar graph as the one in Figure 5.6 and some principles about PID manual tuning presented by Ogata (1997) were applied. According to the author, the most popular method in the industry for PID controller tuning is the Ziegler-Nichols method, and it was the one previously used for the existing controllers in other heat sinks. However, it must be underlined that more attention should be given to this topic, as it could mean the difference between achieving or not stable enough conditions for testing. During the current work, as it was more about evaluating and establishing a first iteration of the heat sink, rather than optimising it, the topic was not further investigated, but some conclusions were taken from it.

5.3.3 Simulink program

Simulink is a graphical programming and simulation tool included in Matlab that allows control and modulation of data in complex dynamical systems. It is mainly composed by a graphical block diagramming tool, meaning that the creation of a program in this environment consists in customizing and connecting the available blocks, instead of a traditional script based interface. The established program for this work consists of five modules (or sub-programs):

- Data from TDS – this module processes the data from TDS, meaning that it reads and attributes each received value to its correspondent variable, which could be a given temperature or volume flow. TDS provides a vector composed by the values of every variable and this Simulink module separates it accordingly;
- Data to TDS – this module is responsible for gathering significant variables such as a pump's modulation. A vector with every value of each variable is created and sent to TDS;
- Write Data – this module is simply writing the values of every variable into a .mat matrix (Matlab file) every second. The resulting matrix is manually converted into an “.xlsx” file (Excel) and further processed into graphs, many of which are presented in this chapter;
- Setup – this is the module where the user interacts with the control system, providing essential inputs. The user can set the desired mode (explained ahead) as well as the values for certain variables, creating specific test conditions. The module reads the

input user set points and stores them in variables, which are then read in the calculation module;

- Calculation – this is where the actual control takes part. This module takes in the inputs from the TDS (which come from the heat sink) as well as the defined settings by the user and processes data in such a way that the desired conditions are achieved.

It was considered that it would not be interesting to present graphically every detail of the program. The setup and calculation modules are fully described as they allow the reader to understand the logic behind controlling the heat sink. This is intended to be a first version of the program whose main goals are to assess the capabilities of the heat sink and to be a first iteration for the establishment of a fully functioning test rig to measure at any possible conditions and with highly complex dynamic modulation. Therefore, this version was kept simple so that is easily understood by the reader and thereby the presented results are clear and conclusive.

Firstly, the user can select the mode at which he desires to operate the heat sink: automatic or manual. In manual mode, the value for each component modulation must be set (for example, P1 at 20 %, V1 at 80 % and V2 at 30 %). However, the output conditions will be unpredictable given the dynamic nature of the system. By choosing the automatic mode, the user must define the desired return temperature (T_2) and opt from volume flow control or ΔT control. As explained earlier in this section, in order to control ΔT , volume flow must be modulated. As such, if volume flow control mode is selected, a specific volume flow must be set, otherwise, the user must set ΔT . In order to set automatic mode, this variable must be set to 1 in the setup window shown in Figure 5.7. In the same way, if “Volume_Manual” is set to 1, the system will control the volume flow according to its set point. Otherwise, it controls the set ΔT .

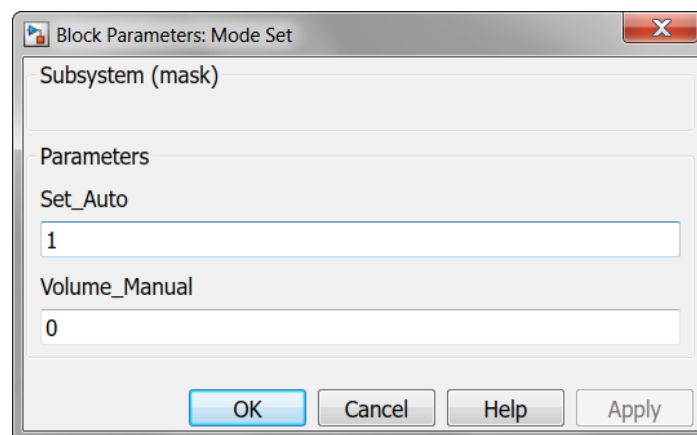


Figure 5.7 – Window for mode definition. When the “Set_Auto” value is set to 0, manual mode will activate.

The calculation module brings every input as well as set points and modulates the output signal accordingly. Figure 5.8 shows the sub-module responsible for modulating the signal of P1.

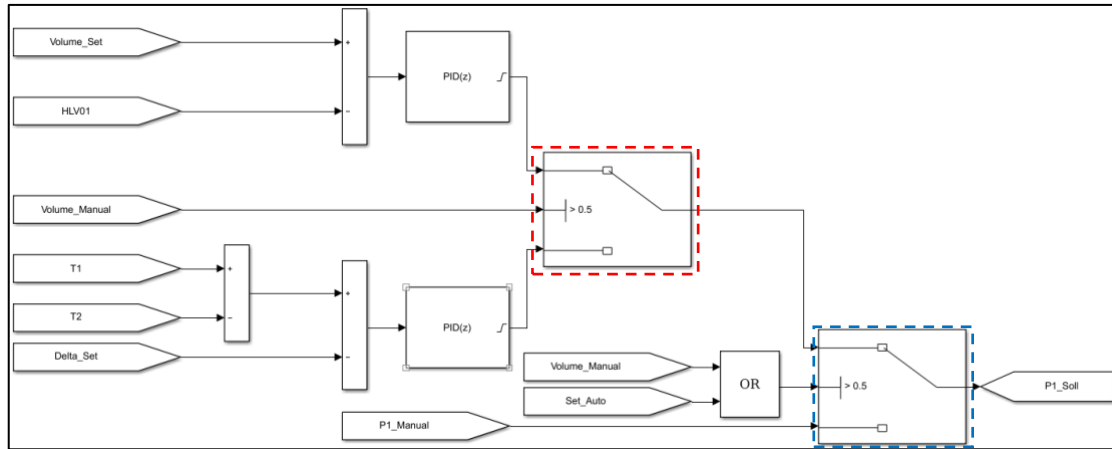


Figure 5.8 – Graphical representation of designed program for P1 modulation control.

The block surrounded by a blue border is where the mode selection occurs. If “Set_Auto” or “Volume_Manual” equal 1, the block switches the output signal to the upper position. Note that this means that “Volume_Manual” can be activated even if the automatic mode is off. The red border evidences the block that switches between volume flow control or ΔT control. In both cases, the input signals from the sensors are subtracted by the set-point, and the error signal is sent to the PID controller, which modulates the output signal according to its parameters, as explained previously. The module responsible for modulating V2 works in a similar way but takes in different inputs.

5.3.4 Control program outcomes

In this section, the achieved results for controlling each parameter are shown. In order to test the established Simulink program, a 15 kW oil boiler was connected to the heat sink so that a heat source was available. The boiler was considered a suitable heat source for testing the system for the first time, since it provides a constant and stable heat power after a start-up period of about ten minutes, which could be well defined by sending a 0-10 V signal through another channel. Each test was initiated when the Simulink program started running and every data was recorded with the “Write Data” module, as explained. The obtained Excel sheet was then used for analysing the data, by building graphs of the obtained values over time.

Firstly, the volume flow control was verified. Figure 5.9 shows its behaviour over time for three set points as well as P1 modulation.

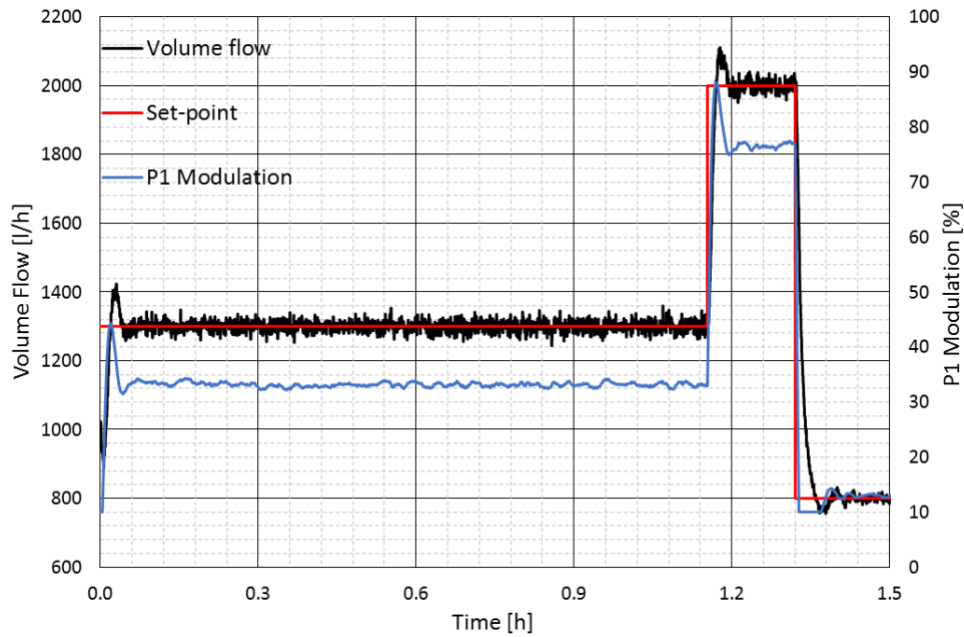


Figure 5.9 – Volume control and P1 modulation behaviour for different set points.

It can be assessed by the observation of the graph that the system responds to the set volume flow, resulting in an oscillation around the desired value. Figure 5.10 shows a more detailed view of this oscillation, which produces deviations up to 60 l/h when the set-point is 1300 l/h, corresponding to 4.6 % of the set value. The cause is believed to be due to the PID values of this controller which could be further tuned, however achieving 1 % of deviation required by standard DIN EN 14511 does not seem promising since even at a pump constant speed, deviations up to 10 l/h were observed.

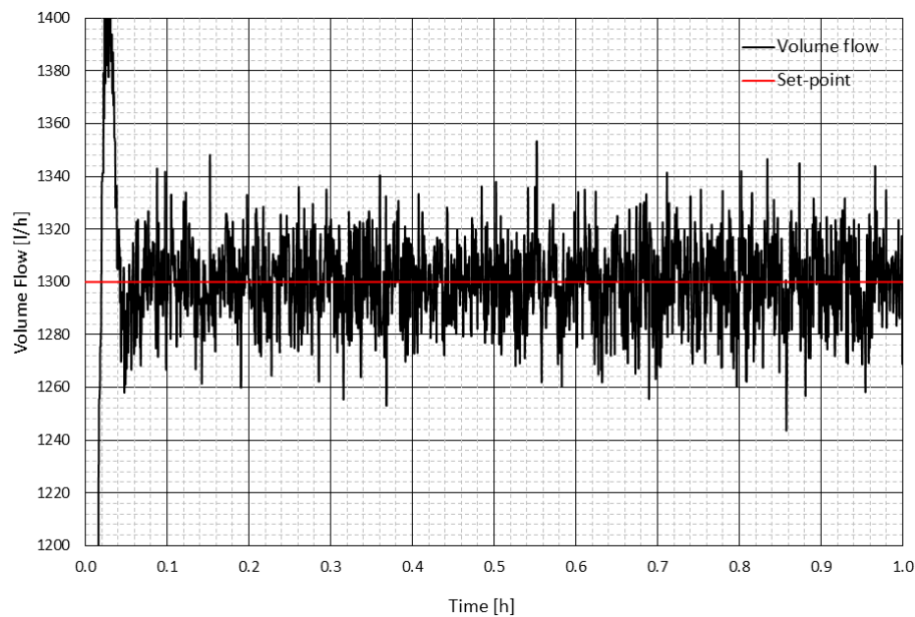


Figure 5.10 – Detailed view of volume flow control behaviour for a set point of 1300 l/h.

The return temperature was tested for two methods – in one of them, the temperature fed to the control was T4, which is at the outlet of the internal heat exchanger. This sensor was chosen because it produces a quick response to the valve modulation, hence making it easier to be manipulated, whereas T2 takes about 15 seconds to respond to a certain change in the valve V2. The results produced with this method are shown in Figure 5.11.

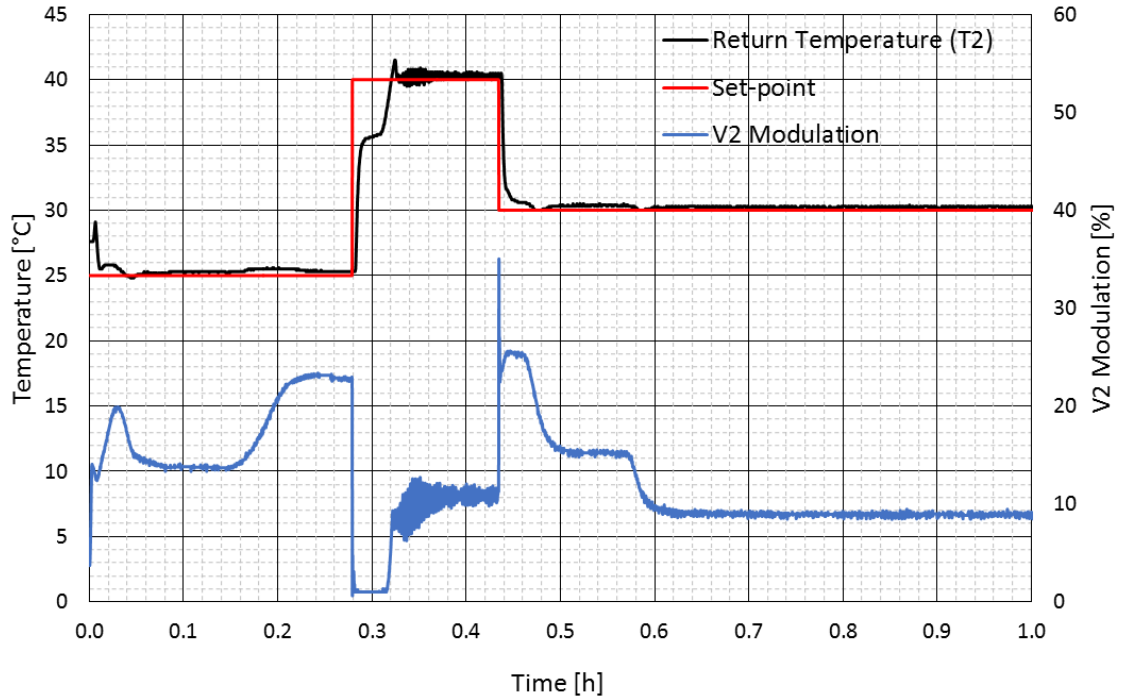


Figure 5.11 – Return temperature and V2 modulation control behaviour overview for different set points and using T4 as feedback signal.

The following topics can be discussed through the observation of the obtained graph:

- V2 modulation changes while the return temperature maintains its value – this happens due to deliberate changes made to the heating power output of the boiler, whose purpose was to observe this phenomenon in order to prove the control basis of the present energetic system proposed at the beginning of this section. For example, at around $t = 0.15$ h, the boiler's heating power was doubled, forcing the valve to open so that volume flow in the intermediary circuit was raised and consequently the dissipated heat as well. More examples of this behaviour (reducing the heat power) are shown in the figure;
- When the set-point is changed from 25 °C to 40 °C, the valve closes to its minimum (1 % due to safety reasons), which reduces heat dissipation also to the smallest value possible, until the temperature achieved by the boiler reaches the set-point, at which

the modulation starts rising, with a minor overshoot. This is a good observation of the response of the control to this energetically dynamic system.

- The higher the set temperature, more unstable the system gets – this is likely related to the temperature difference between the inlets of the different circuits. One must not forget about the behaviour of valve V1 shown in Figure 5.3. Valve V2 is the same model as V1, so it behaves in the same way, since P2 is also the same model as P1. As such, modulating V2 between 0 % and 30 % causes abrupt changes in volume flow, hence the higher instability observed at 40 °C.
- An off-set between the set temperature and the obtained return temperature can be observed and it is related to the fact that the value of T4 was fed to the control system instead of T2. There is a significant hydraulic distance between these two sensors as well as 2 closed bypasses, in which hot water can be stored (inside the pipes). Therefore, water gets to the tip of the heat sink where T2 is located at a higher temperature than the one at the outlet of the heat exchanger.

Due to the latter point and to the fact that using T2 as the reference value for its own controlling would be the most correct way of modelling V2, the program was changed accordingly and the behaviour was yet again observed. However, it produced oscillations up to 30 % of the set value since the response from the temperature was too slow, even after some manual tuning was attempted. As such, and in order to prove this behaviour, a new method was implemented and tested, whose results are shown in Figure 5.12. This method used a weighted average of the sensors along the pipes between T2 and T4 (not represented in the scheme present in this chapter, but on the complete one in Annex 7) as an input to the control system. The value read by T2 contributed 50 % to this average, and the one from T4 10%.

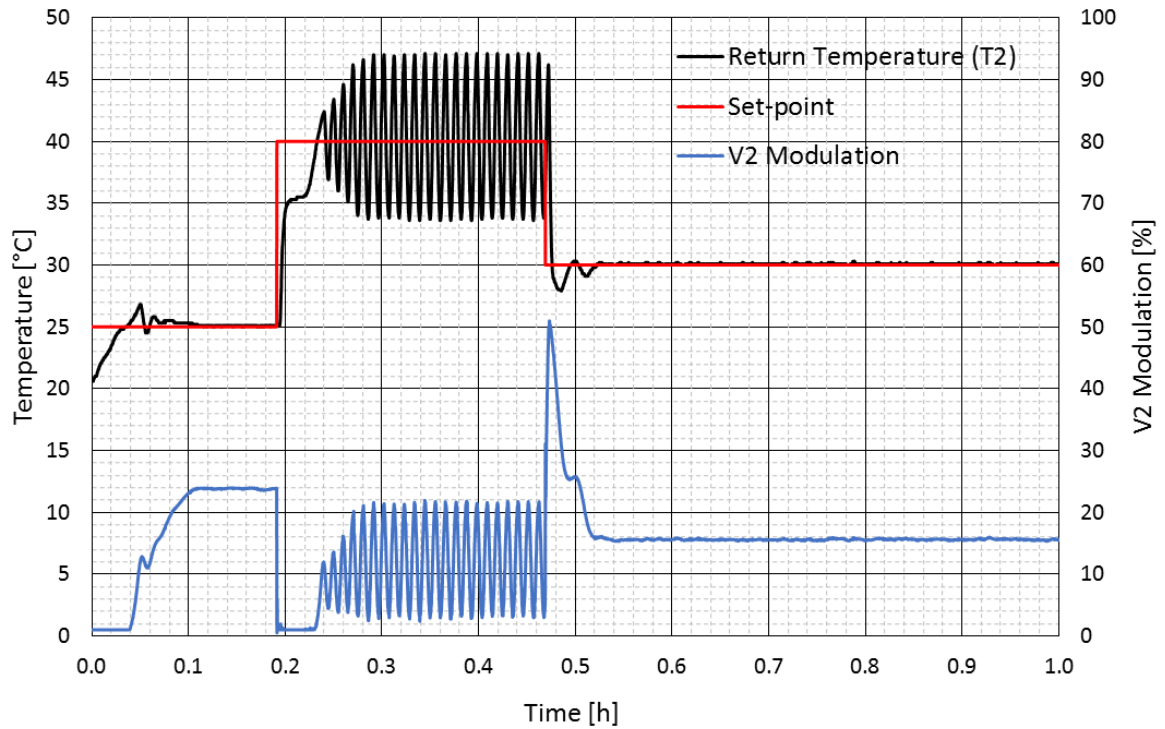


Figure 5.12 – Return temperature and V2 modulation control behaviour overview for different set points and using a weighted average of 4 sensors as feedback signal.

As it can be observed, the temperature is more stable in the lower set-points but highly unstable in higher set-points. The reason has been pointed out – the speed of response from the more distant sensors. Theoretically, the PID values could be better assessed for using T2 as the reference value for controlling and achieving stable conditions with it. But such task could not be performed for time related reasons, as more important factors were yet to be assessed. As such, through the observation of the off-set between T2 and T4 (0.3 °C), it was opted to go with the temperature read by T4 for the control system established for this work, but with an implementation of this off-set. So, the system subtracts 0.3 °C to the set-point and feeds that value to the PID controller. The obtained result with this method are observed in Figure 5.13.

It can be concluded that with the given method, the deviation is never greater than ± 0.2 °C once the system stabilizes. The abnormal behaviour observed between $t = 0.6$ h and $t = 0.7$ h accounts to an increase of the heating power to the maximum available power with this boil. The limits imposed by the standard DIN EN 14511 are wider when transient conditions are met, so this case does not raise any concerns.

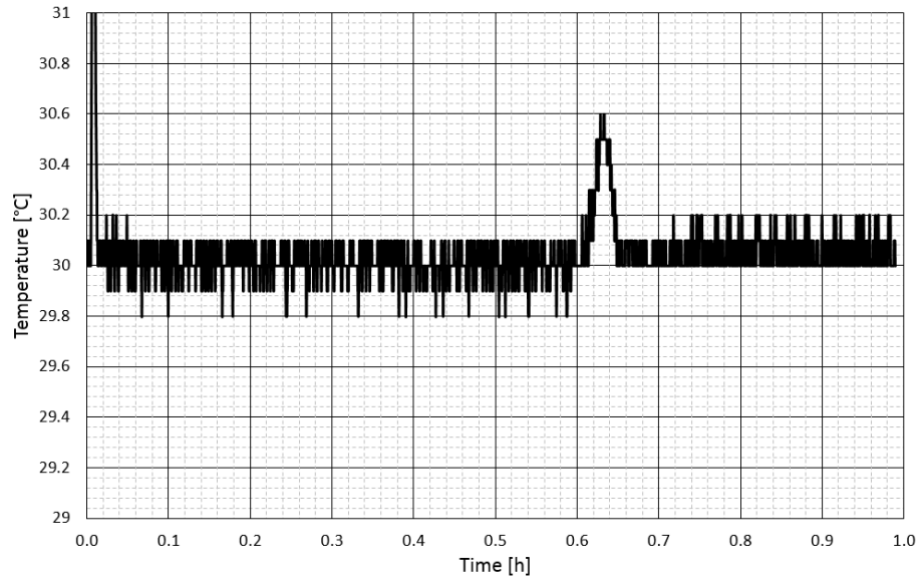


Figure 5.13 – Detailed view of return temperature control behaviour using T4 as feedback signal and an implemented offset.

Lastly, the ΔT control was tested with the same methodology, and the respective results can be analysed with the graph in Figure 5.14. After finding suitable PID values (which were taken from an already existing controller in a different test rig), satisfactory results were achieved, since the increase in heating power occurred at about $t = 0.6$ h resulted in a quick response to modulate up the circulation pump in order to increase volume and thereby maintaining the set ΔT . During this test, the return temperature was controlled with the latest presented method.

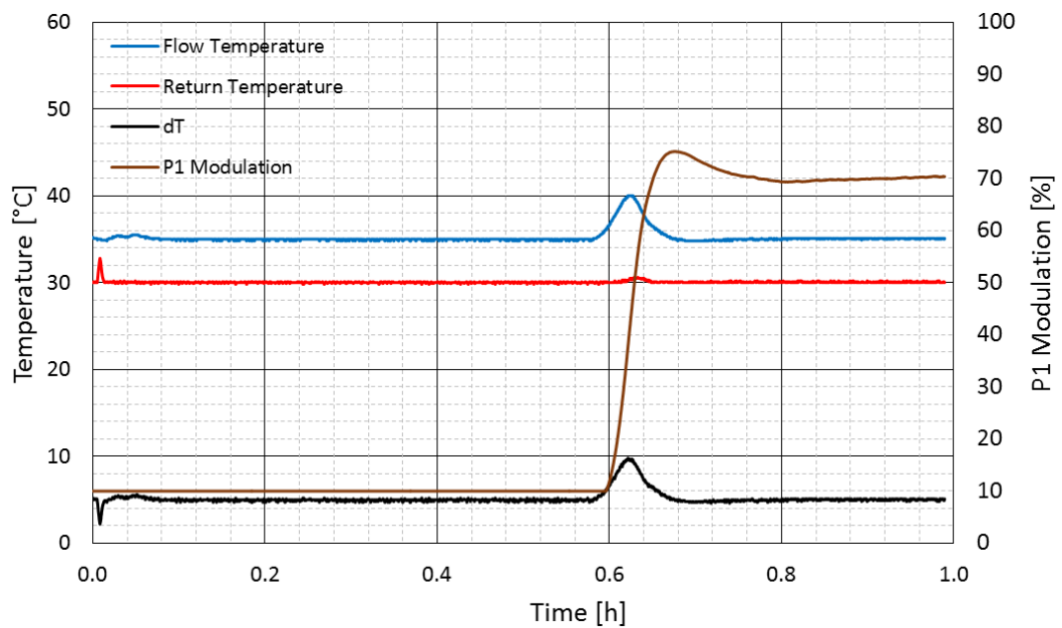


Figure 5.14 – ΔT control and P1 Modulation behaviour overview for minimum and maximum heating power obtained by the oil boiler. Note that dT is equivalent to ΔT .

5.4 Proof of functionality

So far, the environmental concerns behind the development of more efficient heating appliances have been exposed, followed by a description of hybrid heat pumps and the requirements and challenges associated with testing them. The heat source (air – climate chamber) available in the laboratory where this work was developed was then characterised and in this chapter, the heat sink which is responsible for dissipating the heat produced by the appliance being tested, was established by drawing the available hydraulic setup and evaluating its capabilities, as well as by creating a program in Simulink able to control volume flow, return temperature and ΔT so that testing conditions may be achieved with this equipment, but mostly to acquire knowledge about its behaviour, so that an educated guess can be made regarding the future application of this test rig and the steps that should be taken next. So, to sum the chapter up, a test was performed with a heat pump operating inside the characterised climate chamber and dissipating heat into the established heat sink.

Figure 5.15 shows the complete hydraulic representation for this test. Even though the boiler was not turned on for this case, it was included in the sketch due to the hydraulic relevance, since that it was used on the assessments performed on the heat sink control system and in this way, a possible setup for a hybrid heat pump testing is exposed. Both heat generators (heat pump and boiler) are controlled through an indoor unit, whose mixing valve controls what percentage of the outgoing flow (to the heat sink) comes from the heat pump and the boiler. For the present test, the valve was manually manipulated so that 100 % of the flow comes from the condenser of the heat pump, thereby achieving more demanding and dynamic conditions for the heat sink, rather than just a constant heat output, as it happens with the boiler. The IDU contemplates the control of the whole heating system. For the present test, the control of the IDU was set to achieve a flow temperature of 60 °C. This way, as long as such a temperature was not achieved, the heat pump was solicited at the maximum power and thereby the testing conditions were fully manipulated through the heat sink. The achieved results are exposed in Figures 5.16 and 5.17.

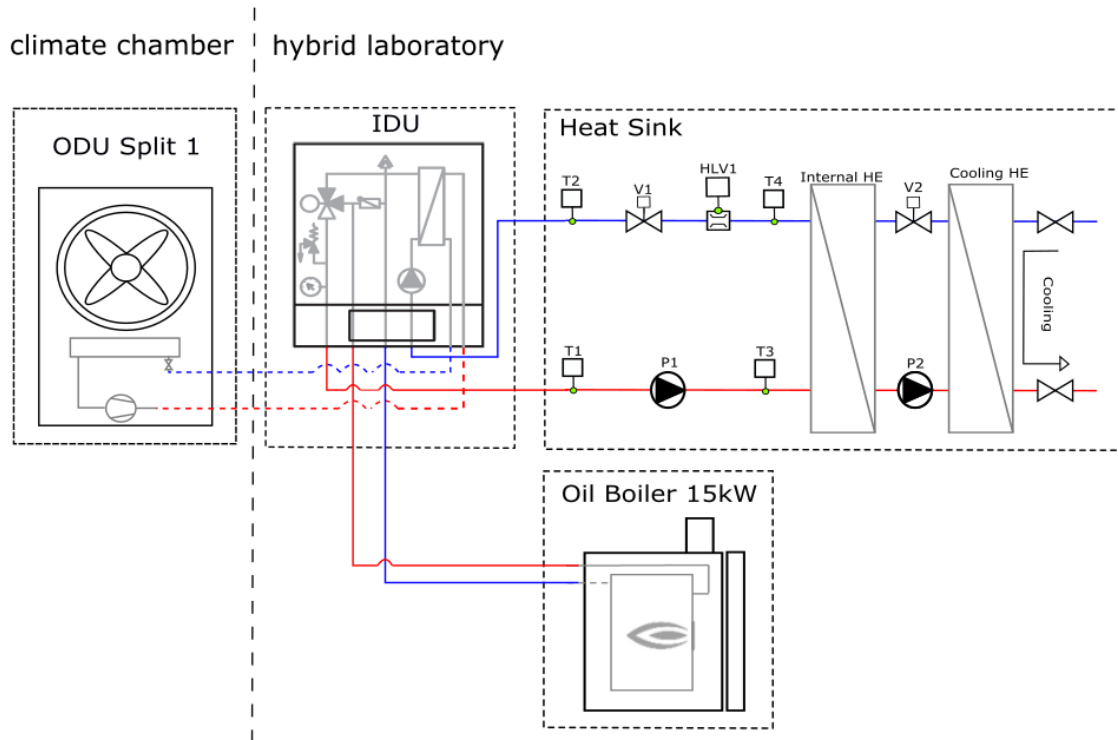


Figure 5.15 – Hydraulic scheme of a complete hybrid heat pump test rig. This setup was used for testing the heat sink.

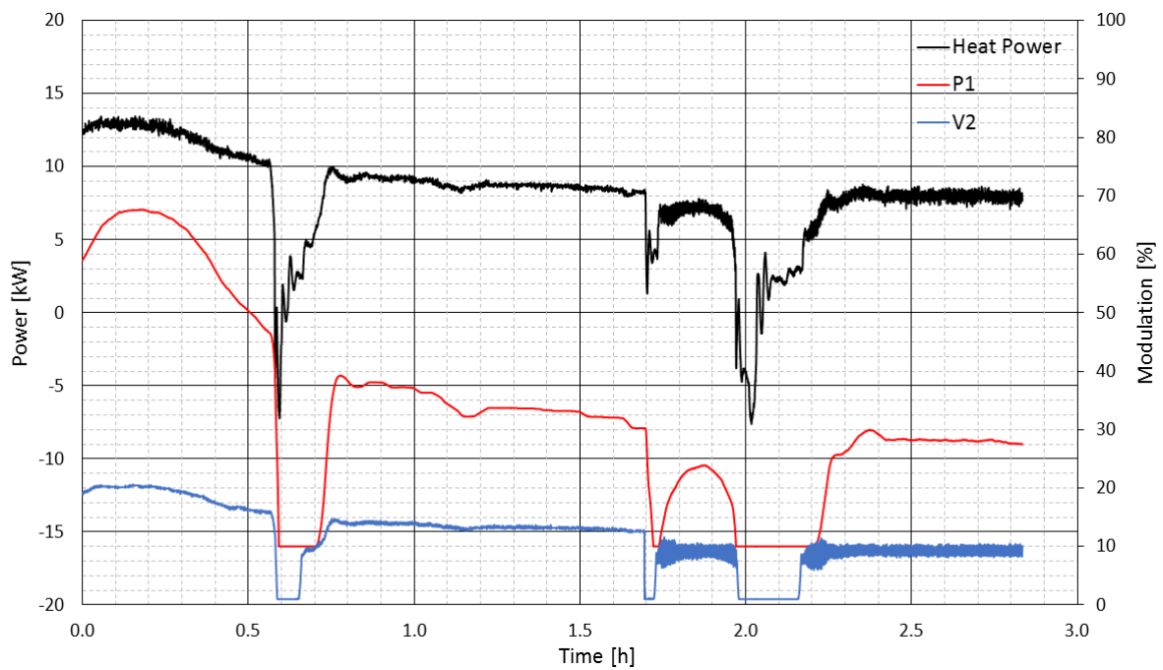


Figure 5.16 – P1 and V2 behaviour during heat pump testing including defrost cycles.

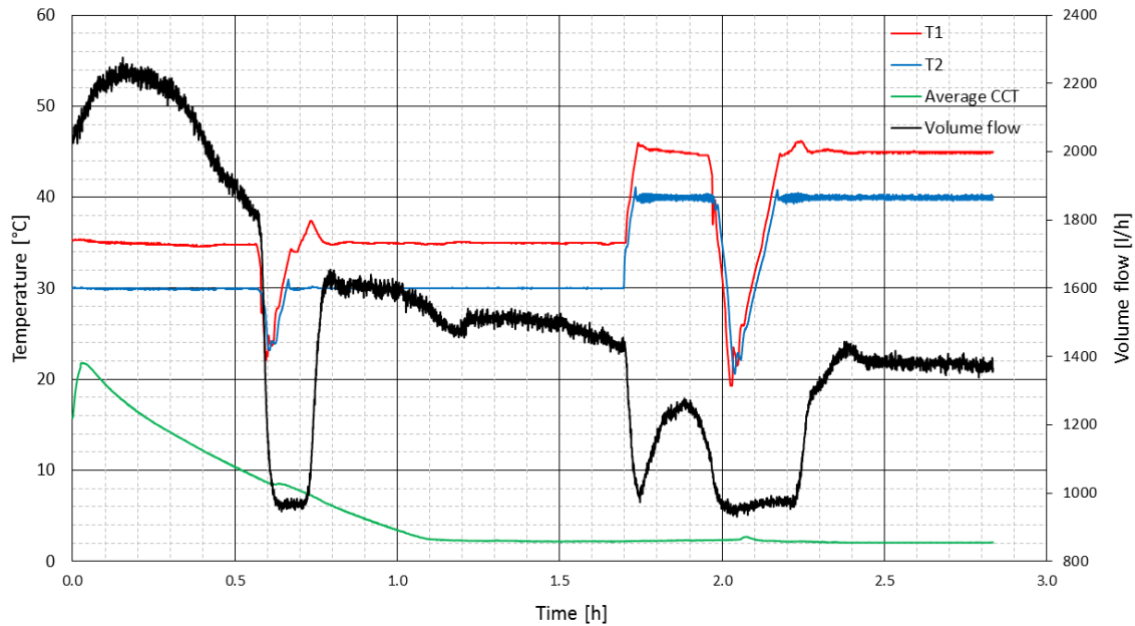


Figure 5.17 – Overview of the heat sink and climate chamber behaviour during heat pump testing including defrost cycles. Inlet and outlet temperatures, volume flow and air temperature average over time values are represented.

Figure 5.16 shows the heat power output provided by the heat pump together with the heat sink's circulation pump and valve modulation. A quick response to the dynamic conditions can be observed, especially during defrost cycles which occurred slightly after $t = 0.5$ h and at $t = 0.2$ h. As the heating power assumes negative values due to the reversion of the heat cycle, the flow temperature quickly drops, which demands a minimum modulation of the pump and the valve. So, from Figure 5.16 one can assess that the control works as it is responsive to unstable conditions. In Figure 5.17 the flow and return temperature fall during defrost cycles is evident, particularly the second one, where the set return temperature was 40°C and it got down to 20°C in less than 10 minutes. The volume flow shows a decrease in heating capacity for lower air temperatures, which can also be observed when comparing this graph to the previous one. Furthermore, when the set point was risen to 40°C (between $t = 1.5$ h and $t = 2.0$ h), it dropped further, which can be observed for $t > 2.5$ h, so that the defrost effect is disregarded. This evidences the theoretical assumption that the heating output of a heat pump decreases when the temperature difference between heat source and heat sink increases, which is the base for the need of developing alternative technology with backup heating for low outdoor temperature conditions, such as hybrid heat pumps. The backup heater would also be useful for avoiding low flow temperatures during defrost cycles. The high-performance oil boiler present in this setup would not react quickly enough, since it has a warm-up time of about five minutes, but a gas wall-hung boiler would. Note that these temperature drops could also be avoided with a hot water storage, not only for real life situations but also for testing.

5.5 Conclusions

At the beginning of this work, there was little to no knowledge about the heat sink, from the hydraulic setup to the control system. As such, the best decision seemed to be to start from the beginning, by gathering all the available information, draw the existing hydraulic setup and developing a new control program. After assessing the complete physical setup, which can be consulted in Annex 7, a simplified version was considered for this dissertation, by closing particular bypasses. Simplifying the hydraulic was the strategy for achieving the first iteration of the control program, so that main functionalities could be assessed and the results could be conclusive. An attempt to control the entire hydraulic at the first try with no previous knowledge about the components' potential problems would likely result in an inconclusive outcome, in which the key issues could not be identified.

Firstly, an analysis of the heating power dissipation at different ΔT was performed. It must be underlined that such an analysis is unique to the connected test object, since it is associated with the respective hydraulic characteristic. Connecting a different appliance will deliver a different pressure drop, changing the characteristic of the system, therefore changing the volume flow associated with the pumps' modulation. Nevertheless, the methodology is implemented, and the Simulink program allows the user to quickly make such an assessment. For the performed test, a maximum of 13 kW of heat could be dissipated at $\Delta T = 5\text{ }^{\circ}\text{C}$, meaning that an appliance with a higher nominal power, working at nominal conditions would overcome this ΔT even at maximum flow. At this point, it is not advisable to modulate the volume flow with the valve here represented as V1. Figure 5.3 shows its behaviour throughout the modulation range, which between 0 % and 30 % induces a steep volume flow rate change. The volume modulation with pump control show deviations from the set value up to more than 3 % (Figure 5.10), which already overcome the limits imposed for testing by DIN EN 14511. Therefore, V1 will induce even higher oscillations. However, in a more advanced version of the program, V1 could be used for small flow adjustments around the set point, as long as its signal does not go under 40 %.

The control of the return temperature and ΔT returned overall positive results since a quick reaction to dynamic conditions to meet the desired set point was observed in Figures 5.11 and 5.14. After trying different methods for controlling the return temperature, a satisfying result with minor deviations from the set point (maximum of 0.2 $^{\circ}\text{C}$) was achieved at 30 $^{\circ}\text{C}$. However, the system showed higher oscillations for a higher set point (observable in Figure 5.17). This variation of stability had been previously observed in Figure 5.12, and it was the reason for changing the feedback signal from the sensor on the outlet of the heat sink (T2) to the sensor on the outlet of the heat exchanger (T4). Even though it resulted in higher stability, the variation in stability is still there and will surely increase for higher temperature differences.

The conclusion for such a behaviour is that the PID values are only suitable for a given temperature set point range. This means that a system which appears to be stable at 30 °C, will not have the same behaviour at 60 °C. When consulting with colleagues from different laboratories, they reported a similar problem. Their solution was to change PID settings every time the set point was changed.

The final tests performed on a heat pump presented satisfactory results with a quick reaction from the heat sink to defrost cycles, which allowed it to achieve back the set conditions in under 10 minutes after defrost. The climate chamber presented the expected temperature stability, assessed from Chapter 4. The graph in Figure 5.17 represents therefore a proof of the whole system (heat source and heat sink) functionality to achieve air and water set points. The flow temperature could not be controlled by the heat sink during defrost, since its decrease is due to the reverse cycle carried out by the heat pump during this period. Standard DIN EN 14511 imposes limits for these periods up until 10 minutes after they end. It cannot be concluded that the whole system complies with the conditions imposed by DIN EN 14511 – the temperature sensors inside the climate chamber and in the hydraulic setup of the heat sink are not accurate enough and the heat sink does not yet present the desired stability for volume flow as well as for return temperature control over the entire set range (30 °C to 60 °C).

As such, considering the aforementioned topics, the following work is suggested for future improvement of the heat sink:

- PID control assessment – a detailed control model of the heat sink must be done in order to achieve desired PID values. To tackle the problem regarding the stability variation, a mathematical model that relates temperature set point and the value of each gain must be achieved and implemented in the control system so that it adjusts the controller according to the set point. A control engineer must be assigned with this specific task if the stability requirements in DIN EN 14511 are to be achieved;
- Use bypasses to overcome the lower heating power dissipation limit – in Annex 7, a full representation of the hydraulic setup shows a possibility of controlling the flow through a bypass with a mixing valve already connected to the PLC. This valve must be analysed in terms of behaviour and it can be used for having a higher control over the output volume flow as well as the return temperature;
- Use the buffer tank available on the hydraulic setup – this can also be seen in Annex 7, and the flow provided to the tank is controlled by V3, which is also connected to the PLC. The tank and the electrical heater were originally placed there for testing cooling operation (so the heat sink could become a heat source, by providing higher temperature flow than the ones it received). However, it would be interesting to use them to gain full control of the return temperature of the heat sink, even during

defrost cycles. The electrical heater could heat the water on the tank up to the return set point, and V3 would control the flow, so that the temperature drops observed in Figure 5.17 would not happen and a complete control of their values could be achieved;

- Develop a more sophisticated control – based on the assessments carried out during this dissertation, it is now possible to further develop the first iteration of the Simulink program. Ultimately, it would be interesting to introduce a heat load profile over a 24 hour period similar to a real house as an input and have the heat sink automatically control the energy dynamic behaviour accordingly.

Lastly, a regard must be done about the suitability of this equipment to test appliances in accordance to the limits imposed by the European standard DIN EN 14511. Whereas the climate chamber presented promising results regarding temperature stability which led to the conclusion that it could meet the desired boundaries provided that new sensors were installed, the case is not that simple for the heat sink. The hydraulic behaviour changes with each appliance connected and the limits could not be maintained for volume flow set point and for the entire range of return temperature control. Therefore, it is not worth to install expensive measurement equipment such as class AA PT 100 in this device before making further assessments on the stability of the entire system, mainly regarding PID tuning. It is advised to explore its potential capabilities in terms of carrying out complex tests possibilities (such as modulating a complex heat load profile over a long period of time) for internal assessment and for research and development purposes rather than investing time and money on trying to make it compliant with the standard, which could potentially never be achieved. The equipment's greatest potential lies on the opportunities to make useful tests and interesting assessment within a large range of heating appliances (during this dissertation, an oil boiler and an air-to-water heat pump were used) rather than maintaining strict limits for releasement measurements, which can be carried out with a more suited and simpler heat sink or even externally. The financial analysis of such possibilities must be carried out.

Chapter 6

Conclusions and Future Developments

“The EU still relies heavily on energy imports from non-EU countries, which provided 54.1 % of all energy consumed in 2015. The main supplier of energy to the EU in 2015 was Russia. It supplied 37.3 % of total gas imports, 32.9 % of imports of petroleum products and 29.1 % of solid fuel imports.”

The citation taken from a Eurostat report entitled “Europe 2020 indicators – climate change and energy” (Eurostat, 2018) presents a clear contextualization of the problem in which this dissertation is framed. This energetic dependence on foreign supply places the European Union in a politically fragile position. On the top of that, lie the environmental challenges described in this dissertation. A joint effort must be done by governments, companies and the population to increase the share of energy originated in renewable sources. But in reality, customers will prioritize price and comfort over environmental and energetic issues, posing a challenge for companies to remain competitive and profitable whilst developing efficient and environmental friendly solutions. Companies generating a significant share of their revenue by selling boilers should carry on developing and producing them. In fact, it is crucial to carry on improving gas boilers efficiency because there is a big share of the population who invested to connect their homes to the gas grid, so they are not prepared to give up on using gas driven appliances and invest yet again a big amount of money in a purely electrically powered heating system. Furthermore, one must also consider that electricity generation source is an important factor, as it would not make sense to have a large percentage of electrically driven appliances for heating without being able to provide the required electricity through renewable sources. Germany still produces about 50 % of its electric energy from coal and gas (Annex 2). Electricity prices also offer some resistance to the deployment of such system on the mass market, especially in countries such as Germany, Belgium and Portugal (Annex 3).

Due to the highlighted issues, the process of implementing electrically driven heating systems must be carried out slowly and progressively. Hybrid heat pumps are an interesting approach to this paradigm because they combine fossil fuels and electricity heat generation to overcome some of the drawbacks associated with air-sourced heat pumps in cold weather conditions. Not only they make sense from a performance and efficiency point of view, but also in terms of a practical approach to the issues presented on the previous paragraph. A hybrid heat pump that includes a gas condensing boiler will require a gas installation anyway, but at the same time will reduce the overall primary energy consumption when compared to a system using only gas. It is also a more efficient way of using gas to generate heat, rather than producing electricity with it to further use on a less efficient heat pump. In sum, these appliances are based on the premise that they achieve a better efficiency in terms of primary energy use, for the time being.

This dissertation and the time spent at the Research & Development facility in Lollar aimed at improving the conditions and general knowledge on using the climate chamber and a heat sink available in the laboratory to make releasement measurements for the energy label established by the European Commission for heating appliances. These measurements must achieve specific conditions introduced by the standard DIN EN 14511, which imposes boundaries over the temperature deviations inside the climate chamber (more specifically at the air inlet of the heat pump's evaporator) and water volume flow and temperature to and from the heat sink. It also limits the accuracy at which measurements must be carried out. The results presented in Chapter 4 for the climate chamber characterisation show that it achieves adequate temperature and humidity stability provided that a two hour stabilisation period is taken before testing starts. However, the results obtained for the heat sink were not so promising in terms of standard compliance. It does not fulfil the limits for temperature deviation over the entire range at which tests must be carried as well as for defrost cycles. These problems must be tackled with a PID analysis and further development of control features using more of its available hydraulic components. Before going further into problem solving, the goals for this equipment must be set clearly. Every temperature sensor both inside the climate chamber and in the heat sink have a higher uncertainty of measurement than the allowed limit, in some cases in more than double meaning that achieving laboratory conditions for testing in compliance with DIN EN 14511 will require not only engineering hours to solve the presented problems but also some financial investment to acquire sensors that achieve the mandatory accuracy. Therefore, the target must be to either achieve full compliance or to make some minor adaptations so that equipment serve the laboratory needs for internal research and development purposes. A financial analysis must be carried out considering that the releasement measurements could be performed in an external facility instead of equipping the laboratory to fulfil the standard requirements. To achieve a full compliance of the standard, the highlighted solutions in the conclusions of the two previous chapters must be taken into account to solve the problems observed during this work.

As for future developments, some other solutions for the upcoming paradigm of the heating market should be considered and studied. Firstly, there are other ways of increasing gas efficiency that should be analysed, such as gas engine heat pumps (GEHP), which are powered by a small gas engine to generate energy for the compressor unlike conventional heat pumps, which use a bigger amount of primary energy in electricity to feed the compressor. Some competitors have deployed such appliances, so the market should be thoroughly analysed because there might actually be space in the near future for these heating systems. Customers will potentially be interested in replacing an old gas boiler for a system using the same type of fuel but as efficient as possible. Therefore, more heating systems developers should look into the potential of such appliances to overcome the efficiency achieved by conventional condensing boilers. Secondly, hybrid systems could consist in combining different sources of energy with electricity, rather

than just fossil fuels. An interesting idea would be to join a biomass heat generator with a heat pump since the same principles of conventional hybrid heat pumps would apply but instead of being powered by fossil fuels, a waste-to-energy system would be implemented with the biomass generator, meaning that such appliances could decrease the use of gas and oil for heating. Considering the volatility and uncertainty of oil prices due to political shifts in the Middle East, this solution would be especially interesting for those looking to replace their oil powered heating systems, since it could potentially provide outstanding primary energy efficiency without compromising heating power and comfort (given that biomass heat generators can easily achieve high flow temperatures) and with no need for gas installations. Another topic that must be given a lot of attention is user experience. In general, costumers are not interested in understanding the complexity associated to their heating systems. They just wish to set temperature conditions for specific rooms at specific times of the day preferably from their smart phones and at the lower possible cost with maximum comfort, leaving the complexity of deciding what kind of appliances and energy sources that should be implemented in their homes for the developers of such systems. Therefore, heating system developers such as Bosch Thermotechnology should consider taking on also the supplier role by cutting out the middle man between them and the costumer and developing direct contact with them. This would make sense for a company like Bosch, which commits to provide technology solutions rather than limiting itself to producing specific appliances and devices. The existing know-how could be used to provide customers with a full assessment of the entire energetic model of their homes by analysing parameters such as solar potential and gas connections to implement an optimal and automated heating system, which requires little to no inputs from the user end.

Lastly, given that this is an engineering dissertation, a brief remark must be done about future disrupting technologies in the energy sector. Smart grid technology consists of a decentralised network that allows every household to be an energy producer just as much as a consumer. In a home or a building equipped with solar panels as well as a few small wind generators, the unutilised produced energy could be fed to the grid, which would then send it to a node of the network where energy is being consumed, with a smart management system. It has the potential to unlock ultimate global energy efficiency. Centralised systems will become more and more obsolete, not only in the energy sector, but also in finance and supply chain. So, for a “global supplier of technology and services” such as Bosch to continue thriving in 50 years, improving the efficiency of its products will not be enough. Blockchain applications for smart grid and energy management technologies should be investigated as they will revolutionise the energetic paradigm just as with banking, finance, supply chain and logistics.

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References

Agora Energiewende, 2017. *Heat transition 2030*, Fraunhofer-Institut für Windenergie und Energiesystemtechnik (IWES).

AISIN Group, 2018. [Online] Available at: <http://www.aisin.co.jp/ghp/english/>

ASHRAE - American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2010. *2010 ASHRAE Handbook - Refrigeration*. Atlanta: ASHRAE.

ANSI/ASHRAE Standard 55-2010, 2010. *Thermal Environmental Conditions for Human Occupancy*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc..

Balaras, C. A., Gaglia A.G., Georgopoulou E., Mirasgedis S., Sarafidis Y., Lalas D.P., 2005. European residential buildings and empirical assessment of the Hellenic building stock, energy consumption, emissions and potential energy savings. *Building and Environment*, Issue 42, pp. 1298–1314.

Baxter, V. & Moyers, J., 1985. Field-measured cycling, frost and defrosting losses for a high efficiency air source heat pump. *ASHRAE Trans.* 91 (2), pp. 537-554.

BDEW, 2015. *Wie heizt Deutschland? - BDEW Studie zum Heizungsmarkt*.

BOSCH Group, 2018. [Online] Available at: <https://www.bosch.com/our-company/>

Bundesministerium für Wirtschaft und Energie, 2018. *BMW I - Energiewende*. [Online] Available at: <https://www.bmwi.de/Redaktion/DE/Dossier/energiewende.html>

Bundesverband der Deutschen Heizungsindustrie, 2017. *BDH: Verband für Effizienz und erneubare Energien*. [Online] Available at: <https://www.bdh-koeln.de/presse/daten-fakten.html>

Clean Energy Wire, 2018. [Online] Available at: <https://www.cleanenergywire.org/>

DKE Deutsche Kommission Elektrotechnik Elektronik Informationstechnik im DIN und VDE, 2008. *DIN EN 60751 - Industrielle Platin-Widerstandsthermometer und Platin-Temperatursensoren (IEC 60751:2008)*, Berlin: DIN - Deutsches Institut für Normung.

Dongellini, M., Naldi, C. & Morini, G. L., 2015. Seasonal performance evaluation of electric air-to-water heat pump systems. *Applied Thermal Engineering*, pp. 1072-1081.

European Commission, 2010. *Directive 2010/30/EU of the European Parliament and of the council of May 2010*.

European Commission, 2018. [Online] Available at: <https://ec.europa.eu/energy/eeepf-labels/>

Eurostat, 2018. [Online] Available at: http://ec.europa.eu/eurostat/statisticsexplained-/index.php/Europe_2020_headline_indicators

Fraunhofer ISE, 2018. [Online] Available at: <https://www.energy-charts.de>

Grassi, W., 2018. *Heat Pumps - Fundamentals and Applications*. Pisa: Springer.

Howard, R. & Bengherbi, Z., 2016. *Too Hot to Handle?*, London: Policy Exchange.

- Junkers, 2018. *Junkers Website*. [Online] Available at: <https://www.junkers.com>
- Kemna, R., 2014. *Average EU building heat load for HVAC equipment*, Delft: Prepared for the European Commission DG ENER C.3.
- Klein, K., Huchtemann, K. & Müller, D., 2013. Numerical study on hybrid heat pump systems in existing buildings. *Energy and Buildings* 69 (2014), pp. 193-201
- Ogata, K., 1997. *Modern Control Engineering*. 3rd Hrsg. New Jersey: Tom Robbins.
- Qiao, H., Vikrant, A. & Reinhard, R., 2018. Modeling of transient characteristics of an air source heat pump with vapor injection during reverse-cycle defrosting. *International Journal of Refrigeration*, pp. 24-34.
- Satyavada, H. & Baldi, S., 2018. Monitoring energy efficiency of condensing boilers via hybrid first-principle modelling and estimation. *Energy*, pp. 121-129.
- Technical Committee CEN/TC 113, 2013. *DIN EN 14511 - Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling*, Berlin: DIN - Deutsches Institut für Normung.
- Technischen Komitee CEN/TC 113, 2017. *DIN EN 14825-Luftkonditionierer, Flüssigkeitskühlsätze und Wärmepumpen mit elektrisch angetriebenen Verdichtern zur Raumbeheizung und -kühlung – Prüfung und Leistungsbemessung unter Teillastbedingungen und Berechnung der jahreszeitbedingten Leistungszahl*, Berlin: DIN - Deutsches Institut für Normung.
- The Boston Consulting Group, 2018. *Klimapfade für Deutschland*. [Online] Available at: http://image-src.bcg.com/Images/Klimapfade-fuer-Deutschland_tcm108-181356.pdf
- The European Parliament and the Council of the European Union, 2010. Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the energy performance of buildings. *Official Journal of the European Union*, Band L 153, pp. 13-35.
- UK Government - Department of Energy & Climate Change, 2012. *Emissions from heat - Statistical Summary*, London: s.n.
- Vignali, G., 2016. Environmental assessment of domestic boilers: A comparison of condensing and traditional technology using life cycle assessment methodology. *Journal of Cleaner Production*, pp. 2593-2508.
- Wang, S. K., 2000. *Handbook of air conditioning and refrigeration*. McGraw Hill.

Annex 1 – An overview of the Bosch Group

The Bosch Group is a global supplier of technology and services established back in 1886 in Stuttgart by Robert Bosch (1861-1942). It currently employs more than 400,000 associates in 60 countries and reached a total of 78.1 billion euros of sales revenue in 2017 (BOSCH Group, 2018). The group is divided into four business sectors: mobility solutions, industrial technology, consumer goods and energy and building technology, where Thermotechnology is inserted.

Bosch aims to lead the supply of technology in promising areas such as automated driving and connected mobility services, as well as contributing to a sustainability with energy efficient and low carbon solutions for mobility, heating and power tools. As such, it invested more than 7 billion euros in research and development in 2017.

The Thermotechnology division is responsible for developing energy efficient heating systems for domestic hot water, central heating as well as for commercial and industrial applications.

Bosch Thermotechnik GmbH in Lollar is one of the German locations of the Thermotechnology division and the largest production plant for this sector, with about 1250 employees. It was founded in 1731 as a foundry and only in 2003 it became owned by Bosch with the Buderus acquisition. Gas and oil cast iron boilers with a power range of 15 – 1200 kW are produced in this facility. However, given that the foundry's capacity surpasses the boiler's production rate, it is also used to produce other casting products such as train brakes, motor shells, gears and other cast mechanical components.

Being one of the main locations for Bosch Thermotechnology, especially for the boiler market, it comprises a research and development department in which this master dissertation project was done.

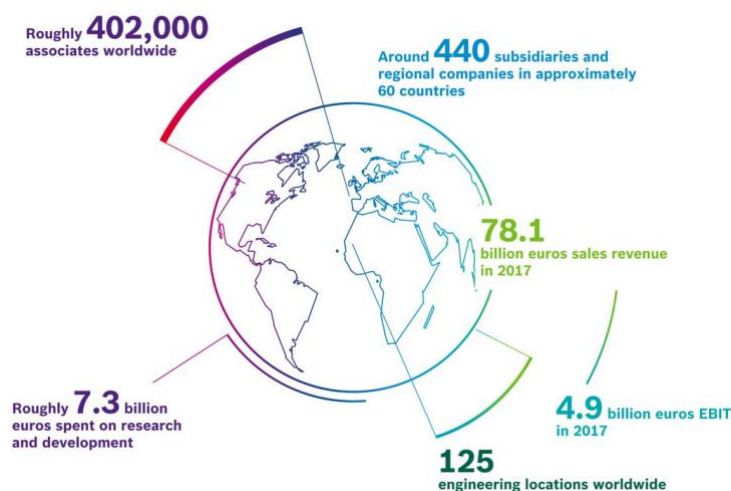


Figure A-1.1 – The BOSCH Group in numbers. Source: <https://www.bosch.com>.

Annex 2 – Statistics regarding energy production and consumption

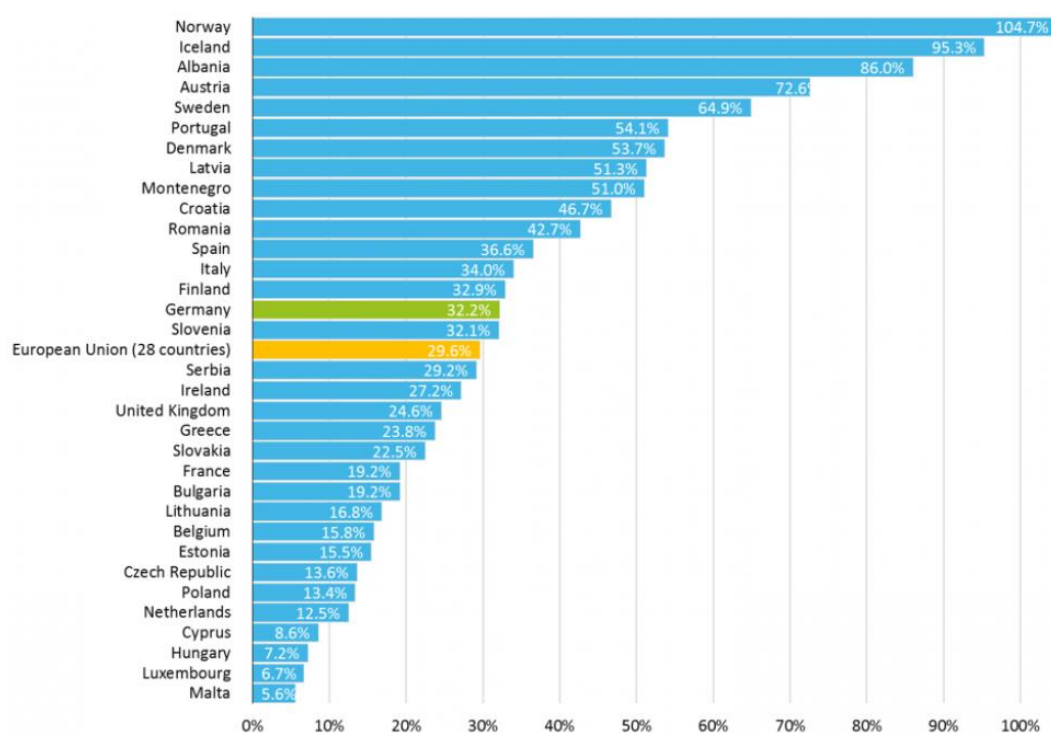


Figure A-2.1 – Share of electricity from renewable sources in gross electricity consumption in European countries in 2016. Source: <https://www.cleanenergywire.org>.

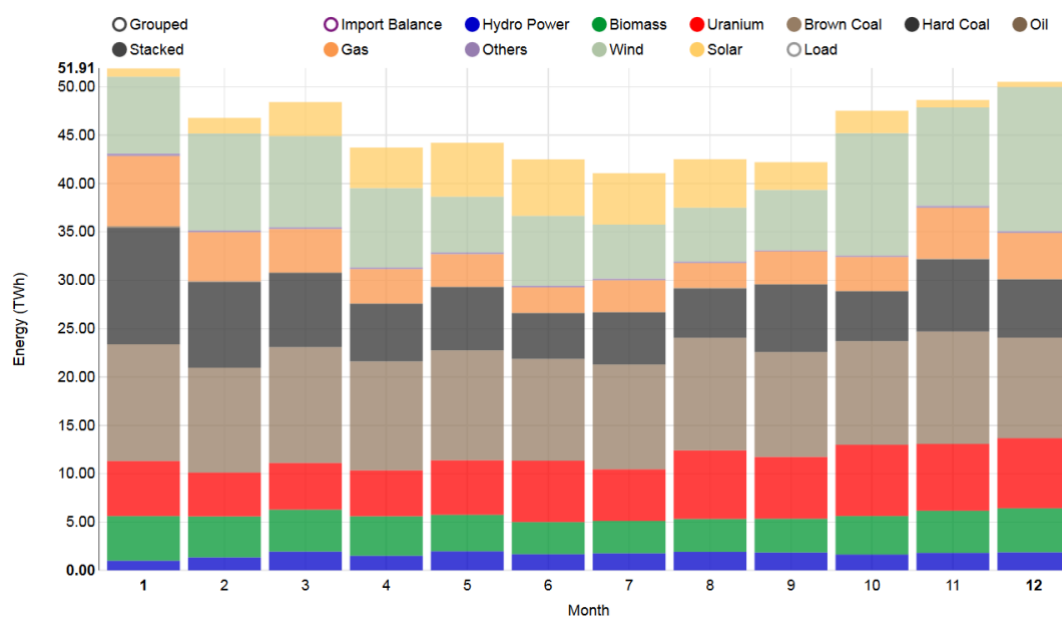


Figure A-2.2 – Monthly electricity generation in Germany in 2017. Source: Fraunhofer ISE 2018.

Annex 3 – Electricity and gas prices throughout Europe

In the present annex, the electricity and gas prices for different European countries in 2017 as well as the ratio between them according to the Eurostat website are presented. It was evidenced in Chapter 2 the influence this parameter might have on the adoption of electricity powered appliances and further reduction on fossil fuel usage. The electricity price in Germany as well as the high ratio between electricity and gas prices in Germany and Belgium must be highlighted. The price of gas in these countries is five times lower than the electricity price for the residential sector (in a primary energy based comparison).



Figure A-3.1 – Electricity prices in €/kWh for different European countries. Source: Eurostat 2018.

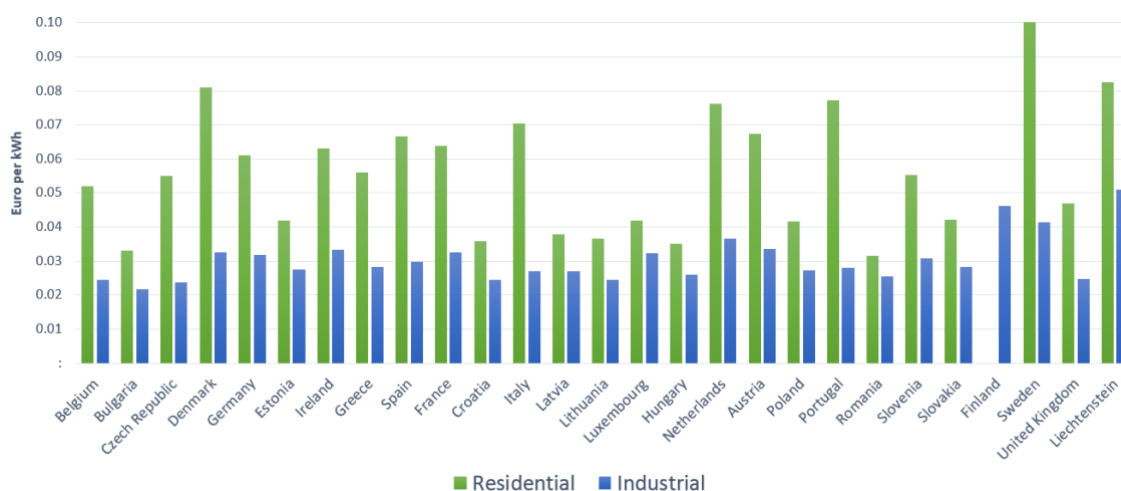


Figure A-3.2 – Gas prices in €/kWh for different European countries. Source: Eurostat 2018.

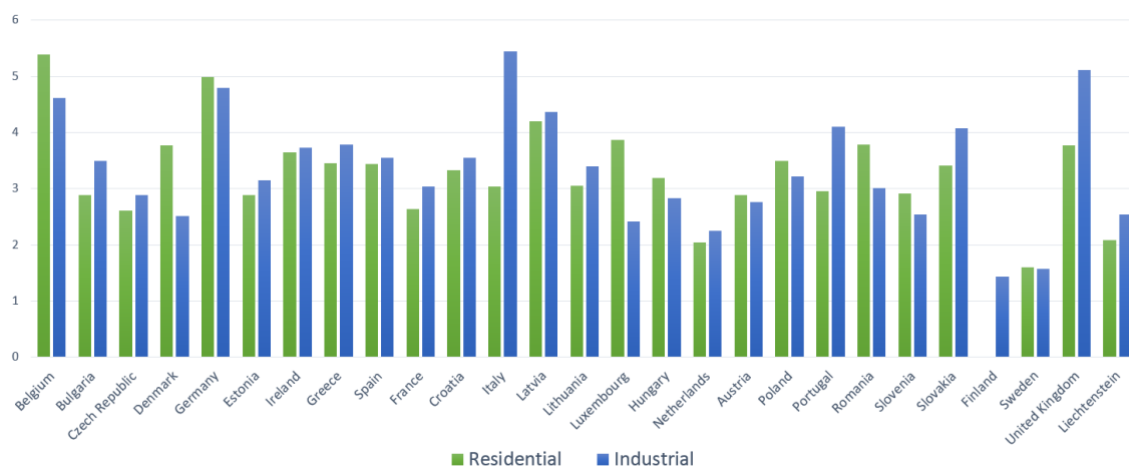


Figure A-3.3 – Ratio between electricity price and gas price for different European countries.

Source: Eurostat 2018.

Annex 4 – Tabled bin hours for $SCOP_{ON}$ calculation in different climates

This annex presents the reader with the bin hours considered for $SCOP_{ON}$ calculation. The bin hours are based on statistical data and characterize cold, average and warm climates, which have been defined in Chapter 3.

Table A-4.1 – Bin hours for different climates

Bin number (j)	Temperature (T_j) [°C]	Hot climate hours	Average climate hours	Cold climate hours
1 to 8	-30 to -23	0	0	0
9	-22	0	0	1
10	-21	0	0	6
11	-20	0	0	13
12	-19	0	0	17
13	-18	0	0	19
14	-17	0	0	26
15	-16	0	0	39
16	-15	0	0	41
17	-14	0	0	35
18	-13	0	0	52
19	-12	0	0	37
20	-11	0	0	41
21	-10	0	1	43
22	-9	0	25	54
23	-8	0	23	90
24	-7	0	24	125
25	-6	0	27	169
26	-5	0	68	195
27	-4	0	91	278
28	-3	0	89	306
29	-2	0	165	454
30	-1	0	173	385
31	0	0	240	490
32	1	0	280	533
33	2	3	320	380
34	3	22	357	228
35	4	63	356	261
36	5	63	303	279
37	6	175	330	229
38	7	162	326	269
39	8	259	348	233
40	9	350	335	230
41	10	428	315	243
42	11	430	215	191
43	12	503	169	146
44	13	444	151	150
45	14	384	105	97
46	15	294	74	61

Annex 5 – Testing procedures presented in DIN EN 14511

In order to understand the testing procedure for heat pumps imposed by DIN EN 14511 and predicts the various possible scenarios upon testing such devices, a collection of information is here presented. Firstly, the different periods and procedures are presented, followed by Figure A-5.1 which is an example of a testing procedure and finally Figure A-5.2 shows a flow chart with the whole procedure for any possible testing outcome, with reference to the previously presented definitions. For this reason, the topic number of each definition was maintained. Note that the mentioned values from Table 4 and Table 5 are presented in this dissertation in Table 3.7 to 3.9. Table 4 presents the deviations for a steady state test and Table 5 presents the deviations for a transient test. The following text is therefore a complete citation from the standard DIN EN 14511:

4.4.4.2 Preconditioning period - The test room reconditioning apparatus and the heat pump under test shall be operated until the test tolerances specified in Table 4 are attained for at least 10 min. A defrost cycle may end a preconditioning period. If a defrost cycle does end a preconditioning period, the heat pump shall operate in the heating mode for at least 10 min after defrost termination prior to beginning the equilibrium period. It is recommended that the preconditioning period end with an automatic or manually induced defrost cycle when testing at application rating conditions for outdoor air stated in Table 3 and in Table 18 to 21 of EN 14511-2:2013. For units having defrost cycles at the standard rating condition, the water flow rate shall be set at the corresponding inlet/outlet water temperatures 20 min after the end of a defrost cycle, manually or automatically induced.

4.4.4.3 Equilibrium period – The equilibrium period immediately follows the preconditioning period or the defrost cycle and a recovery period of 10 min that ends a preconditioning period. A complete equilibrium period is one hour in duration.

4.4.4.4 Data collection period – The data collection period immediately follows the equilibrium period. Data shall be sampled at equal intervals that span every 30 s or less, except during defrost cycles as specified below. During defrost cycles, plus the first 10 min following defrost termination, data used in evaluating the integrated heating capacity and the integrated power input of the heat pump shall be sampled more frequently, at equal intervals that span every 10 s or less. When using the indoor air enthalpy method, these more frequently sampled data include the change in indoor-side dry bulb temperature. When using the calorimeter method, these more frequently sampled data include all

measurements required to determine the indoor-side capacity. For heat pumps that automatically cycle off the indoor fan during a defrost, the contribution of the net heating delivered and/or the change in indoor-side dry bulb temperature shall be assigned the value of zero when the indoor fan is off, if using the indoor air enthalpy method. If using the calorimeter test method, the integration of capacity shall continue while the indoor fan is off. The difference between the leaving and entering temperatures of the heat transfer medium at the indoor heat exchanger shall be measured. For each interval of 5 min during the data collection period, an average temperature difference shall be calculated, $\Delta T_i(\tau)$. The average temperature difference for the first 5 min of the data collection period, $\Delta T_i(\tau = 0)$, shall be saved for the purpose of calculating the following percent change:

$$\% \Delta T = \frac{\Delta T_i(\tau = 0) - \Delta T_i(\tau)}{\Delta T_i(\tau = 0)} \times 100 \quad (\text{A.1})$$

4.4.4.5 Test procedure – When a defrost cycle ends the preconditioning period - If the quantity $\% \Delta T$ exceeds 2,5 % during the first 70 min of the data collection period, the heating capacity test shall be designated a transient test (see 4.4.4.7). Likewise, if the heat pump initiates a defrost cycle during the equilibrium period or during the first 70 min of the data collection period, the heating capacity test shall be designated a transient test. If the above conditions do not occur and the test tolerances specified in Table 4 are satisfied during both the equilibrium period and the first 70 min of the data collection period, then the heat capacity test shall be designated a steady-state test. Steady-state tests shall be terminated after 70 min of data collection.

4.4.4.6 Test procedure – When a defrost cycle does not end the preconditioning period

4.4.4.6.1 If the heat pump initiates a defrost cycle during the equilibrium period or during the first 70 min of the data collection period, the heating capacity test shall be restarted as specified 4.4.4.6.3.

4.4.4.6.2 If the quantity $\% \Delta T$ exceeds 2,5 % any time during the first 70 min of the data collection period, then the heating capacity test procedure shall be restarted as specified in 4.4.4.6.3. Prior to the restart, defrost cycle shall occur. This defrost cycle may be manually initiated or delayed until the heat pump initiates an automatic defrost.

4.4.4.6.3 If either 4.4.4.6.1 or 4.4.4.6.2 apply, then the restart shall begin 10 min after the defrost cycle terminates with a new equilibrium period of one hour. This second attempt shall follow the requirements of 4.4.4.3 and 4.4.4.4 and the test procedure of 4.4.4.5.

4.4.4.6.4 If the conditions specified in 4.4.4.6.1 or 4.4.4.6.2 do not occur and the test tolerances specified in Table 4 are satisfied during both the equilibrium period and the first

70 min of the data collection period, then the heating capacity test shall be designated a steady-state test. Steady-state tests shall be terminated after 70 min of data collection.

4.4.4.7 Test procedure for transient tests – When, in accordance with 4.4.4.5, a heating capacity test is designated a transient test, the following adjustments shall apply. To constitute a valid transient heating capacity test, the test tolerances specified in Table 5 shall be achieved during both the equilibrium period and the data collection period. As noted in Table 5, the test tolerances are specified for two sub-intervals. Interval H consists of data collected during each heating interval, with the exception of the first 10 min after defrost termination. Interval D consists of data collected during each defrost cycle plus the first 10 min of the subsequent heating interval. The test tolerance parameters in Table 5 shall be determined throughout the equilibrium and data collection periods. All data collected during each interval, H or D, shall be used to evaluate compliance with the Table 5 test tolerances. Data from two or more H intervals or two or more D intervals shall not be combined and then used in evaluating Table 5 compliance. Compliance is based on evaluating data from each interval separately. The data collection period shall be extended until 3 h have elapsed or until the heat pump completes three complete cycles during the period, whichever occurs first. If at an elapsed time of 3 h, the heat pump is conducting a defrost cycle, the cycle shall be completed before terminating the collection of data. A complete cycle consists of a heating period and a defrost period; from defrost termination to defrost termination. For a multiple refrigerant circuit units, the data collection period is 3 h whatever the state of cycling of the different refrigerant circuits.

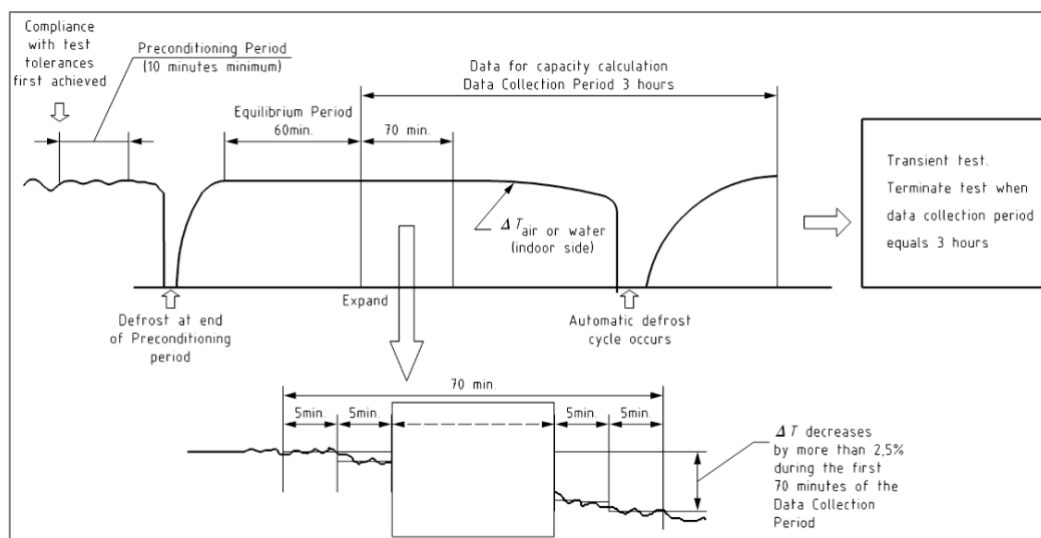


Figure A-5.1 – Transient heating capacity test with one defrost cycle during data collection period. Source: DIN EN 14511.

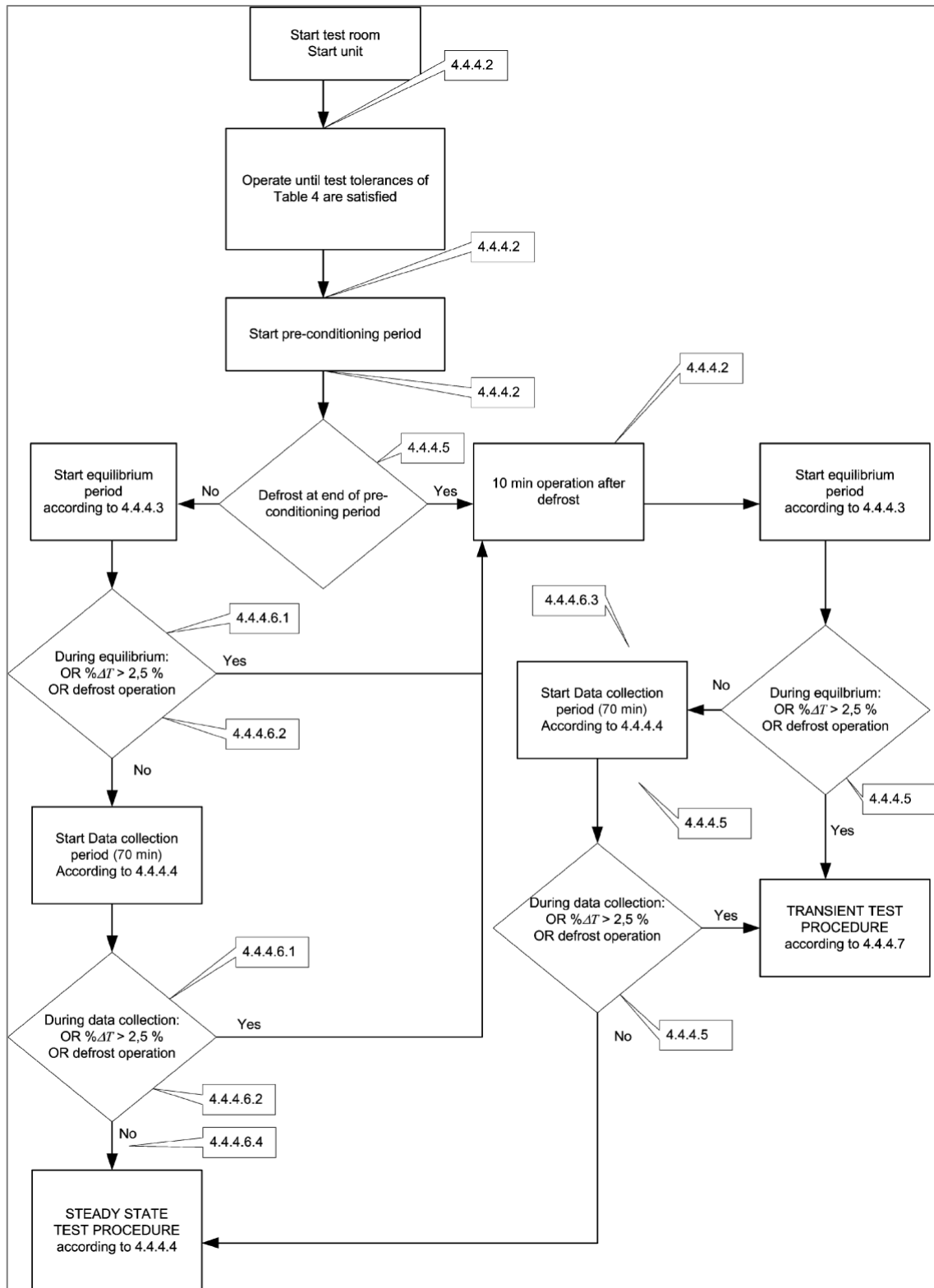


Figure A-5.2 – Flow chart for testing. Source: DIN EN 14511.

Annex 6 – Results obtained from the climate chamber tests

This annex complements the information presented in Chapter 4 regarding the climate chamber characterisation. Not every graph was presented given that the results shown in the chapter together with the ones in this annex are enough for assessing the stability of the temperature inside the climate chamber around the air inlet of the evaporator.

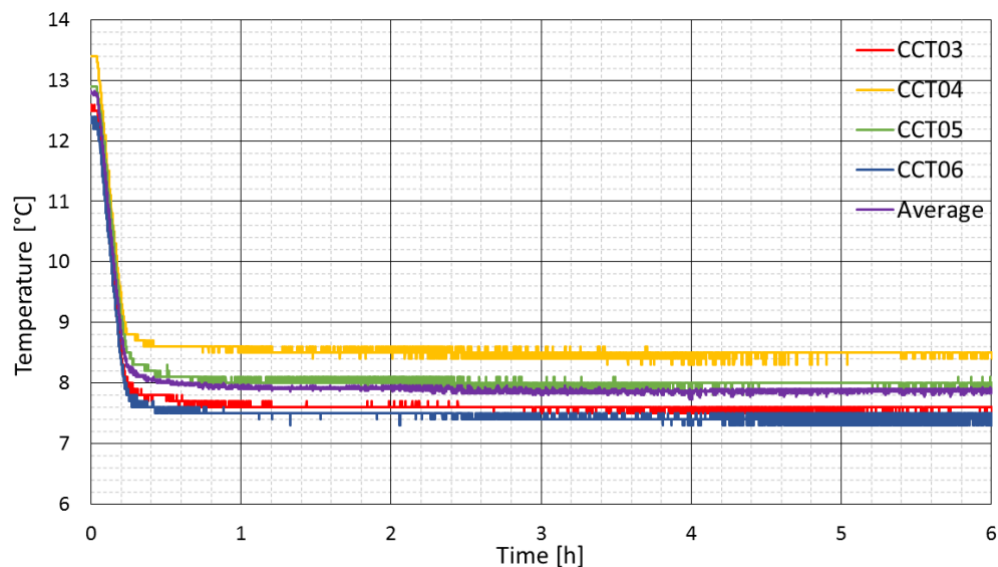


Figure A-6.1 – Temperature values over a 6 hour period for 4 sensors and a set point of 7 °C and 86.7 % RH (Test 2).

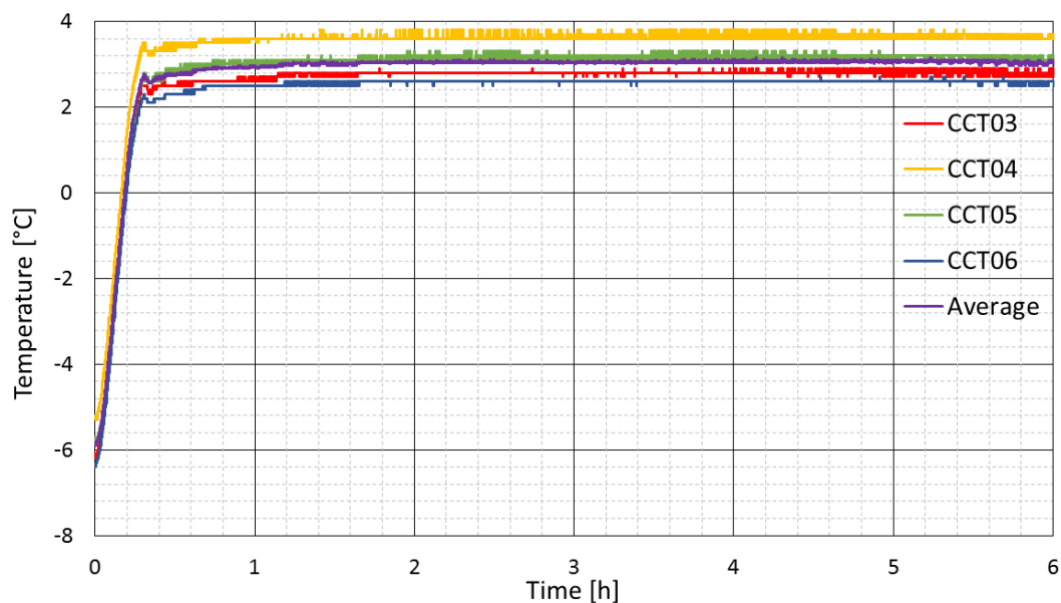


Figure A-6.2 – Temperature values over a 6 hour period for 4 sensors and a set point of 2 °C and 83.6 % RH (Test 3).

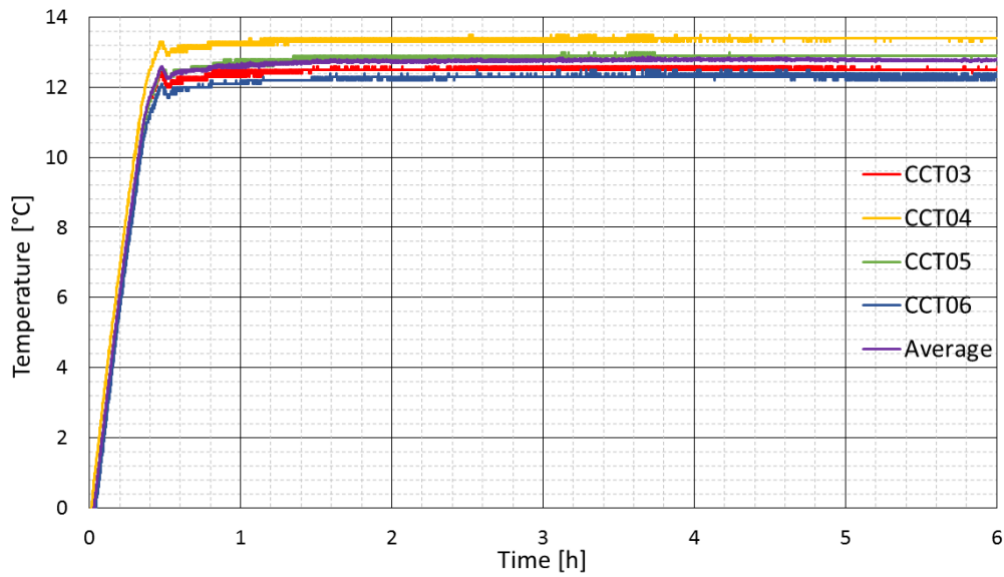


Figure A-6.3 – Temperature values over a 6 hour period for 4 sensors and a set point of 12 °C and 88.8 % RH (Test 4).

Figures A-5.4 to A-5.6 show the graphs obtained for tests with heat load. Periodical defrost cycles can be observed but temperature stability was not affected by them.

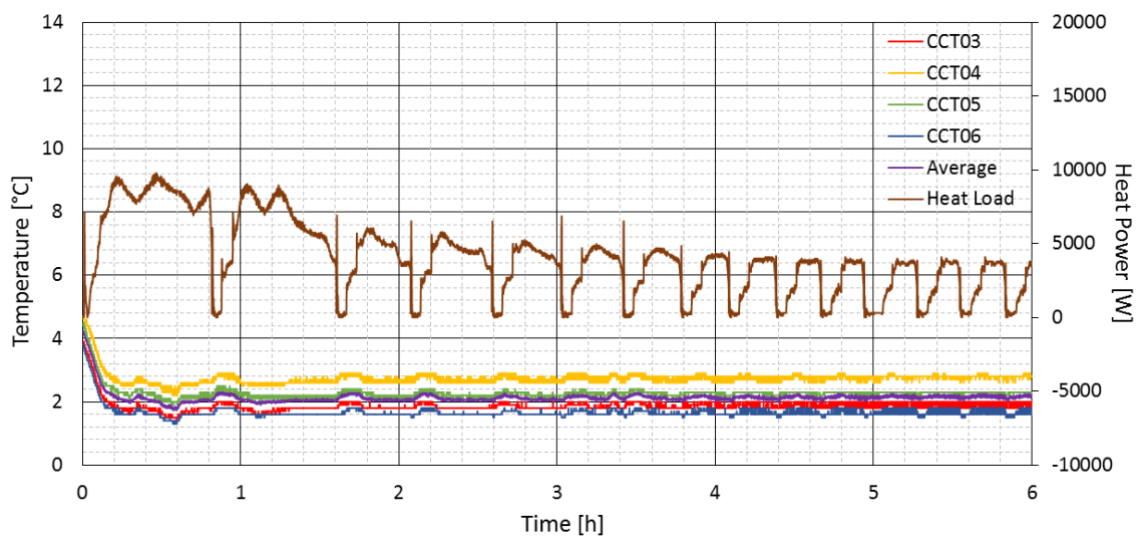


Figure A-6.4 – Heat load and temperature values for 4 sensors over a 6 hour period. Set point of 2 °C and 88.8 % RH and water at 65 °C (Test 9).

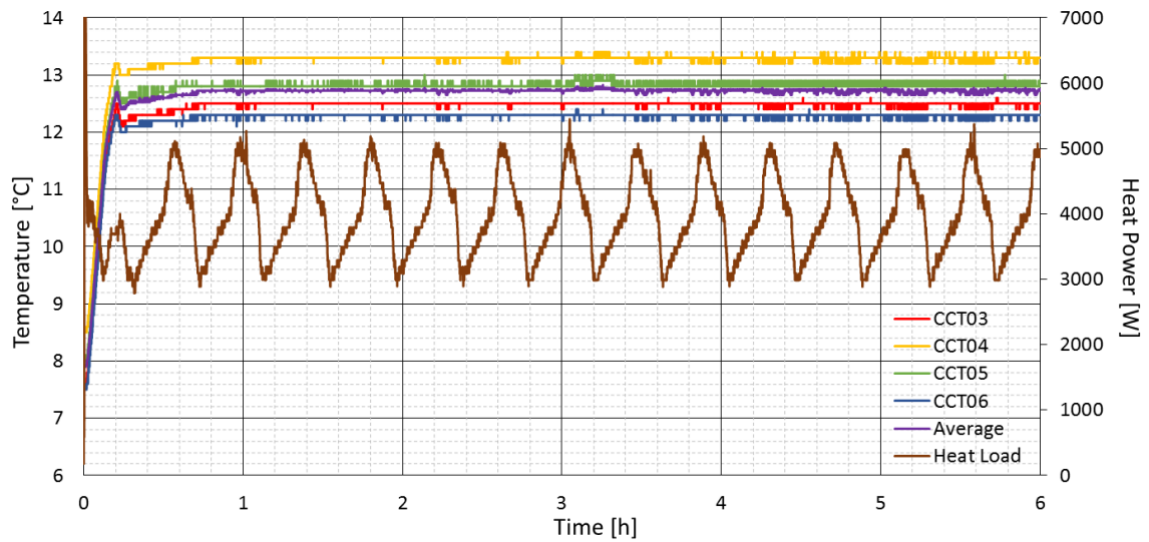


Figure A-6.5 – Heat load and temperature values for 4 sensors over a 6 hour period. Set point of 12 °C and 88.8 % RH and water at 35 °C (Test 13).

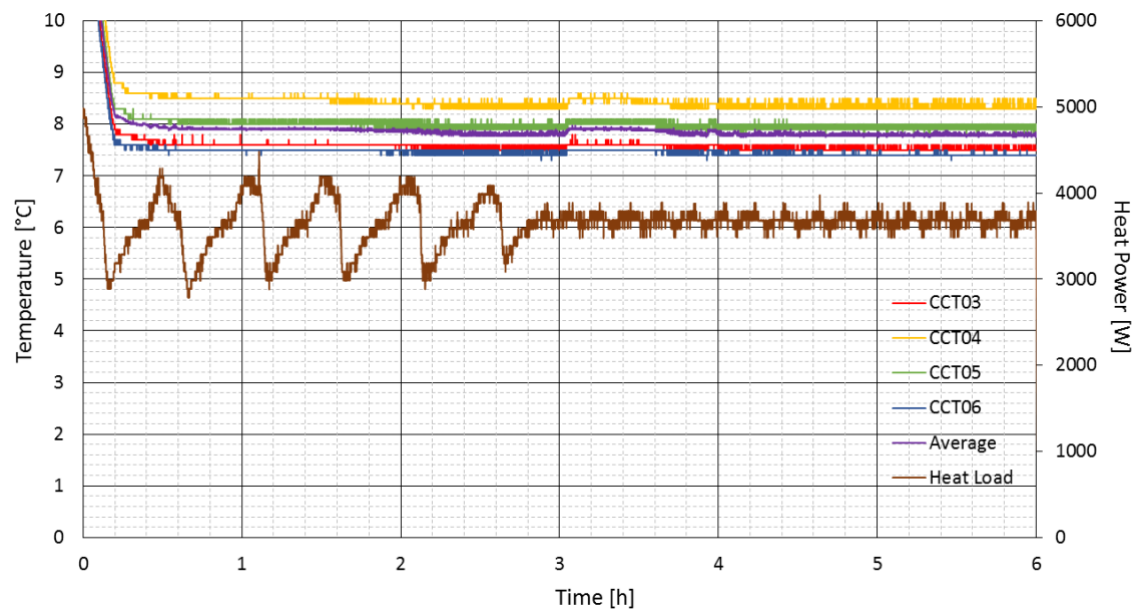


Figure A-6.6 – Heat load and temperature values for 4 sensors over a 6 hour period. Set point of 7 °C and 86.7 % RH and water at 35 °C (Test 14).

Annex 7 – Heat sink full hydraulic representation

This annex presents the full hydraulic setup available for the characterised and developed heat sink in Chapter 5. As explained previously, this setup was simplified in order to make correct and conclusive assessments on a few components and a simple circuit, instead of having a complex hydraulic and therefore not being able to identify the source of a given behaviour. Note that the actual names of the components are different from the ones used during Chapter 5. From now on, the given names of the hydraulic components stand for the representation presented in this annex. For the simplification, the following procedures were taken:

- The bypass on the left side (the closest the test object) was manually closed on the represented valve;
- The other bypass was closed by introducing an instruction on the PLC program to always modulate valve V1 to 100 %, so that it was always getting the flow from the test object;
- Valve V3 was set to 100 % on the PLC program, so that the circuit to the buffer tank was closed, given that this was not used during the work performed on this dissertation.

After such simplifications were done, P1 and P2 were considered to be only one pump (P1) as they are aligned and the signal sent to each other was the same, meaning that 10 % modulation corresponded to both pumps were modulated to 10 %, for example. As for P3 and P4, they were also considered to be only one (P2) as P3 was actually turned off. The sensors' names were also adapted for Chapter 5 so that they would make sense in the considered setup. The original hydraulic drawing can be seen on the next page, in Figure A-6.1.

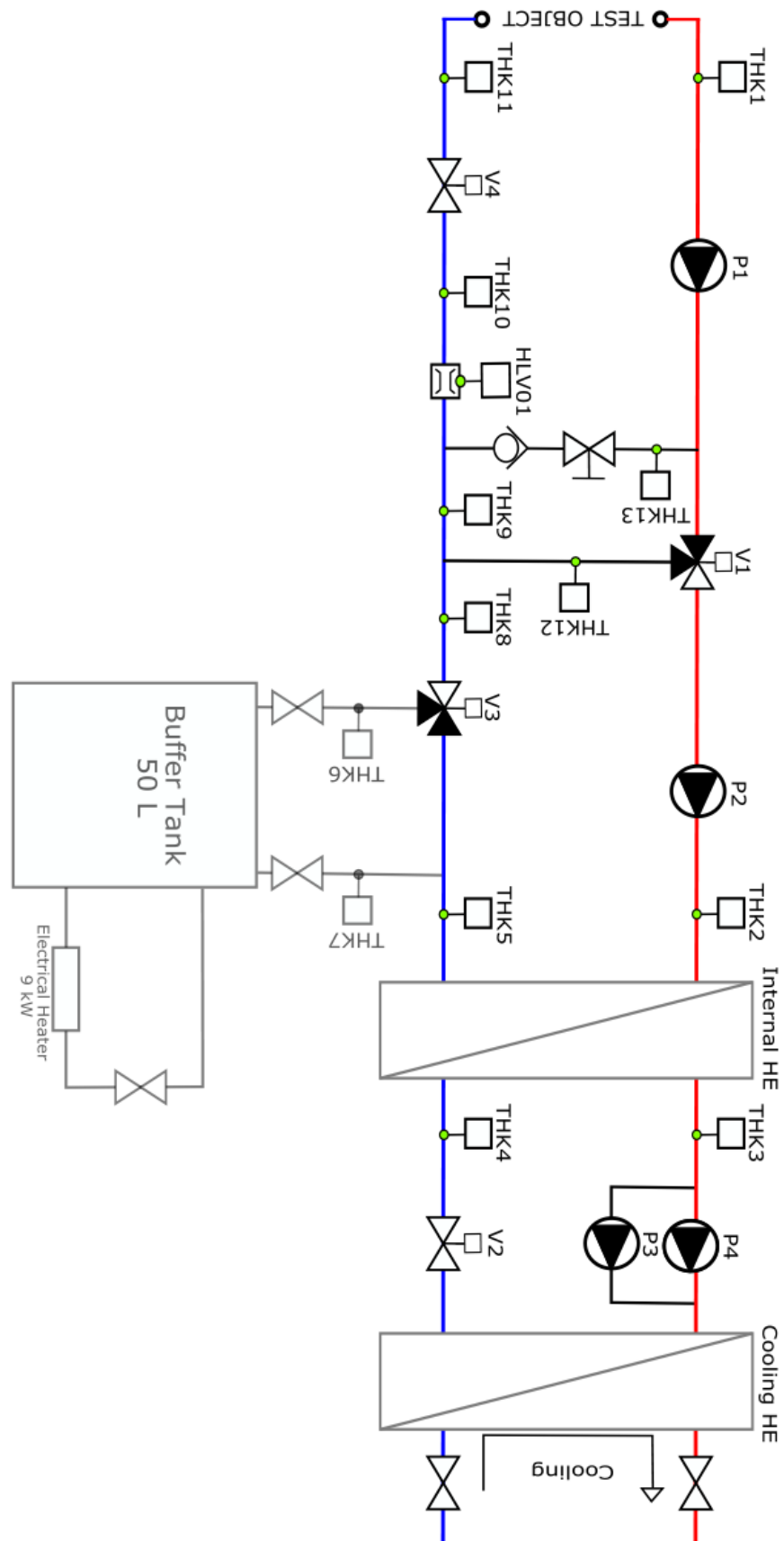


Figure A-7.1 – Representation of the real hydraulic setup available on the heat sink studied during Chapter 5.