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Control of Liquid Cooling Unit

Evaluation of Present Control and Suggestion for Improvements

Master of Science Thesis

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Göteborg, Sweden, 2010

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Cover:
Illustration of the ground-based radar systems Arthur and Giraffe AMB from Saab Electronic Defence Systems. [1] [2]

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SUMMARY

The electronics and the manned compartment in the two mobile ground-based radar systems, Arthur and Giraffe, at Saab Electronic Defence Systems are cooled by a Liquid Cooling Unit (LCU). This LCU is today controlled by means of relay logic and is robust, but not flexible for changes. Furthermore, the noise level due to the condenser fans inside the LCU is high and a decrease in this noise level is desirable.

In this work the system, as it is today, has been analyzed and the possibility for a more flexible control has been investigated. The focus in present thesis has been to evaluate the LCU and today's control of it. A model of the LCU has been built in Simulink. A LCU has been used to perform measurements on, in order to identify the behaviour of the system as well as to obtain an understanding of the dynamics of the system.

Two approaches have been made to examine how the control of the system can be improved. The first one is the possibility to control the frequency of the condenser fans, and the second is the possibility to control the frequency of the compressor. The results of this evaluation show that a frequency control of the fans can be implemented and reduces the noise level for an average LCU load situation. This noise reduction has been verified by experimental tests. However, a frequency control of the present compressor is not an option due to the safety limits of the compressor model.

As a mean of implementation the authors suggest a BEA control and communication system in combination with an NFO frequency converter. Both of these units are available within the company, the frequency converter has been tested together with the condenser fans, and the implementation of the control and communication system can be tested as a future work.

Keywords: frequency control, fan control, refrigeration systems, Saab, capacity control, BEA multiplex

Preface

This M.Sc. thesis has been conducted at, and in collaboration with, Saab Electronic Defence Systems in Göteborg. It is a part of the examination of the Master program *Systems, Control and Mechatronics* at Chalmers University of Technology and has been carried out during the spring of 2010.

The authors would like to thank all Saab employees who have been involved in this project, or by other means contributed to the results presented in this report. Special thanks to Peter Bung, the supervisor at Saab, for his encouragement, guidance and support throughout the work with this thesis. Thanks are also directed to Rolf Nilsson for his help with the BEA multiplex system, Håkan Olsson and the other personnel in Fordonshallen for all assistance with practical problems that have occurred, Daniel Dermark for his assistance with electrical connections and in situations of unexpected difficulties, Henrik Egerbo for help with the experimental setup, and Jan Rydén, Torgny Hansson and Maria Söderström for planning and administration of the project.

Several of the Ph.D's and professors at the department of Installationsteknik at Chalmers University of Technology have also contributed to the making of this thesis with their knowledge and experience of refrigeration systems. Amongst these are Mattias Gruber, Torbjörn Lindholm and Per Fahlén.

Another thanks is directed to Knut Åkesson, the supervisor and examiner at Chalmers, and who have contributed with important comments and guidance regarding the thesis.

Last, but not least, a special thanks is directed to the author's friends and families for all their support during the work with this thesis.

Göteborg, June 2010
Monica Johansson & Rebecka Villiamsson

Contents

Nomenclature

Abbreviations

ACU	Air Conditioning Unit
BEA	Bus Electronic Architecture
ECU	Electric Cooling Unit
EGW	Ethylene Glycol Water
LCU	Liquid Cooling Unit
R-134a	The refrigerant 1,1,1,2-Tetrafluoroethane
PCB	Printed Circuit Board

Terminology

Ambient Mode	Cooling by ambient temperature
Active Mode	Cooling by the vapour cycle system
Active Cooling	Cooling with the help of compressor
Bypass	No need for cooling, but the compressor is on and bypassed

Symbols

C	Capacitance [F]
C_p	Specific heat capacity [kJ/(kg K)]
F	Force [N]
h	Enthalpy [kJ/Kg]
n	Fan speed [rpm]
P	Pressure [psi]
q	Volume flow capacity [m^3/s]
Q_i	Energy [kJ]
$\dot{Q}$	Heat transfer rate [kJ/m]
R	Resistance [Ω]
s	Entropy [kJ/(kg K)]
t	Time [m]
T_i	Temperature [$^\circ C$]
V	Voltage [V]
W	Work [kJ]
x	Displacement [mm]

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1 Introduction

Ground-based radar systems are part of the Saab Electronic Defence Systems product portfolio. The electronics and manned compartment in the mobile ground-based radar systems, Arthur and Giraffe AMB, are both cooled by the same type of Liquid Cooling Unit (LCU). Figure 1 shows Arthur and Giraffe AMB. The LCU supplies a tempered mixture of Ethylene Glycol and Water (EGW) that flows through the system to the different cooled spaces, and then back again to the LCU. The cooled spaces comprise two separate units, the Electronics Cooling Unit and the Air Conditioning Unit. The flow of EGW to and from these units is shown schematically in figure 2.

Figure 1: *The two ground-based radar systems Arthur and Giraffe AMB. Both of them are equipped with Liquid Cooling Units. [1] [2]*

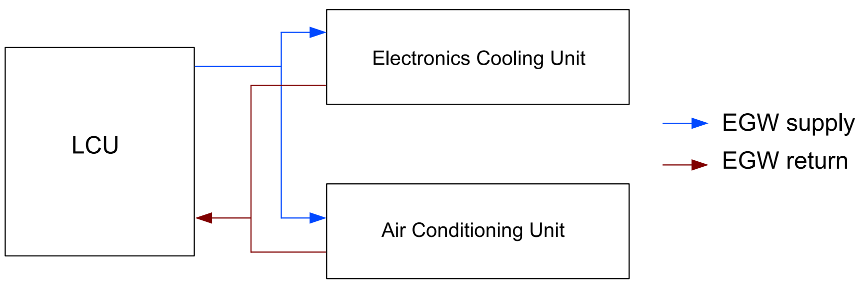

Figure 2: *Illustration of the interaction of the Liquid Cooling Unit with related components.*

Today the control of the LCU is done by means of relay logic. The input from the system to the control unit is given by temperature and pressure switches. An evaluation of whether a certain multiplex system, Bus Electronic Architecture (BEA), could be implemented to replace the relay logic was of interest. This multiplex system has been developed by AB Volvo and is supplied by the company Consat Engineering. If the BEA system could be used the flexibility of the system would increase.

The main focus of the thesis was however to evaluate the present control system of the LCU, and to investigate if frequency control could be implemented to reduce the noise level caused by the condenser fans in the LCU. In order to analyze the present control of the system, as well as to test a new control of the condenser fans, several experimental tests were performed. An investigation of whether or not the LCU could have another sort of capacity control was also performed in this thesis work.

1.1 Background

Saab Electronic Defence Systems (previously Saab Microwave Systems) is a part of Saab AB and is a leading provider of airborne, ground-based and naval radar systems. Two of their mobile ground-based radar system solutions are Giraffe AMB and Arthur [1] [2].

To reduce the risk of overheating the electronics in these vehicles, and to keep the manned compartment at a suitable temperature, in a wide range of possible ambient temperatures, a LCU supplies a tempered mixture of EGW to an Air Conditioning Unit and an Electronics Cooling Unit. The temperature of this EGW needs to be maintained at a constant temperature in a wide span of ambient temperatures and load situations.

The same type of LCU is used in both Arthur and Giraffe AMB, the only difference being the sizes of the components. The control of the LCU is the same in these two units and the LCU performs according to today's demands. However, the control is not flexible to any changes of these demands, nor to any readjustments of the system. In case of a need for lower EGW temperature supplying the Air Conditioning Unit or the Electronics Cooling Unit, the entire control part needs to be altered physically.

The LCU consist of three main parts, the condenser unit (condenser heat exchanger with 2 fan pairs), the compressor unit (evaporator heat exchanger, pump, compressor and diverter valve) and an expansion tank, see figure 3. The main goal of the LCU is to chill EGW and send the cooled EGW to the warm regions of the vehicle. The warmed-up EGW is then returned to the LCU to be cooled again. In order to accomplish this goal, the LCU make use of a refrigerant, R-134a (1,1,1,2-Tetrafluoroethane), to cool the EGW in a vapour compression refrigeration cycle. The outside air is also used to cool the EGW and to deposit the excessive heat produced in the vehicles.

The pump in the compressor unit makes sure that the EGW flow through the LCU, out to the consuming units and back again. The compressor increases the pressure and maintain the flow of the refrigerant.

The LCU has three modes of operation; Ambient mode, Active mode and Redundancy mode. Ambient and Active mode are the two main principles of operation mode in the LCU, and the Redundancy mode is a safety mode when a failure has occurred in the vapour compression cycle while in Active mode.

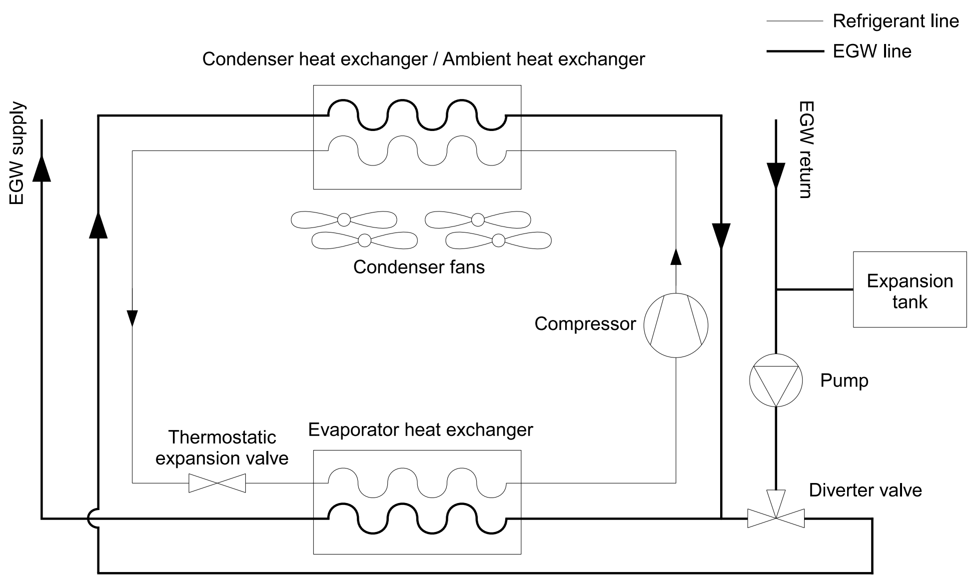

Figure 3: *Illustration of the main components of the LCU.*

In Ambient mode the refrigerant is not used, thus the compressor is off, and the EGW is cooled by the ambient air. The EGW passes the condenser heat exchanger, where the ambient air and the condenser fans are used to cool the EGW, see figure 3. On the way out of the LCU it also passes the evaporator heat exchanger, but the heat exchanger is inactive since the refrigerant is not moving. The fans work in pair and are either on or off. In this mode the fan activity is controlled by the EGW supply temperature.

In Active mode the compressor is on and the refrigerant flows through the evaporator heat exchanger to cool the EGW, thus the EGW does not pass the condenser heat exchanger. In this mode the fans are used to cool the refrigerant and they are controlled by the refrigerant pressure after the compressor.

The fans switches on and off frequently in both Ambient and Active mode. This frequent on and off behaviour is unwanted due to the noise level caused by the fans. By reason of this, there was an interest in evaluating the possibilities to frequency control fans to decrease this noise level. An evaluation of the capacity control of the compressor was also of interest.

1.2 Purpose

The purpose of this project was to analyze the control of the LCU used today, and also to evaluate and present a suggestion for a more flexible control. This solution should include both software and hardware which are compatible with the existing surrounding systems. It was desirable to evaluate the possibility to implement the solution with the BEA multiplex system and including sensors.

The on/off manner of the fans arose the interest of evaluating whether implementing frequency control of the fans would decrease the noise level. The possibility to capacity control the compressor was also of interest to be evaluated.

The purpose can be segmented into different steps, according to the list below.

- Modelling the system
- Evaluating the noise level of the condenser fans
- Evaluating frequency control of the fans
- Evaluating other capacity controls of the compressor
- Evaluating the BEA multiplex system for the implementation of the designed controls
- Testing new controls, theoretically and practically

1.3 Methodology

To accomplish the desired goals, it was necessary to gain knowledge about the current control system. This was carried out by studying documents, test data and circuit diagrams provided by Saab, through discussions together with the company's supervisor and through specially designed tests on an LCU.

1.3.1 Software

The simulation model of the present control system in Ambient mode was done with MATLAB/Simulink, it was based on documented data from previous tests on the system. When developing and evaluating the new control system through simulation, the model of the present control system was used as a basis for comparison as well as old test data.

A course in the BEA multiplex system was attended, to evaluate the possibilities to use this multiplex system to implement a new control system. Before the course, material about this system and also simulation tools of this system was used to gain theoretical and practical knowledge about BEA.

1.3.2 Experimental Tests

The information given by Saab was enough to gain knowledge about the Ambient mode. There was not enough data on the internal refrigerant line of the system when in Active mode, thus experiments were carried out on the system. The experimental setup was built to have the possibility to simulate load (warm the EGW) and also to simulate different surrounding temperatures. A heater and a styrofoam sauna was used to simulate these different conditions. Tests were carried out to measure the noise level of the condenser fans, both when they work as today and at different frequencies.

A lot of surrounding equipment, to collect data, was necessary for the experimental tests, such as thermocouples, pressure sensors, data logger etc. Of those, pressure sensors had to be purchased.

1.4 Exclusions

Other parts of the vehicle, such as the Air Cooling Unit and the Electronic Cooling Unit, were not accounted for when analyzing and evaluating the LCU. The return EGW from these units was considered a return from an unknown system.

There was no possibility to practically implement frequency control on both the fans and the compressor at the same time since only one frequency converter was available for testing. It was not possible to frequency control the compressor, since it could harm the existing compressor.

1.5 Structure of the thesis

Section 1 is the introductory part with a short background of the LCU system and the motivation for the thesis.

Section 2 presents the general theory behind refrigeration systems as well as methods to control them, to increase the understanding of the LCU system. It also includes information about the BEA multiplex system, which is to be evaluated for possible future usage in the control of the LCU.

Section 3 explains the different parts of the LCU system, how it function and its present control system.

Section 4 presents and explains how the experimental tests were carried out, the results from them and also a short discussion.

Section 5 analyzes the present control of the LCU based on information from section 2 and results from the experiments.

Section 6 presents the creation process of the simulation model and also the results from testing a new control.

Section 7 discusses and summarizes the thesis and gives suggestions for further work.

Appendix A includes additional background theory about sensors, and frequency inverters.

Appendix B includes the block diagrams used for simulations created in Simulink.

2 Background Theory

This section presents the general theory behind refrigeration systems like the one in the LCU, as well as methods to control them.

2.1 About Vapour Compression Refrigeration Cycles

In this section the basic theory behind vapour compression refrigeration cycles will be described briefly, including common methods for analyzing these systems. This section, including its equations, is based on information from [3], [4], [5], [6] and [7].

The purpose of all refrigeration cycles is to move heat from a cold space to a hotter one. In the case with vapour compression refrigeration cycles, the heat is transferred from the cold space to the hot space by the use of a refrigerant as medium. An illustration of the basic principle is shown in figure 4.

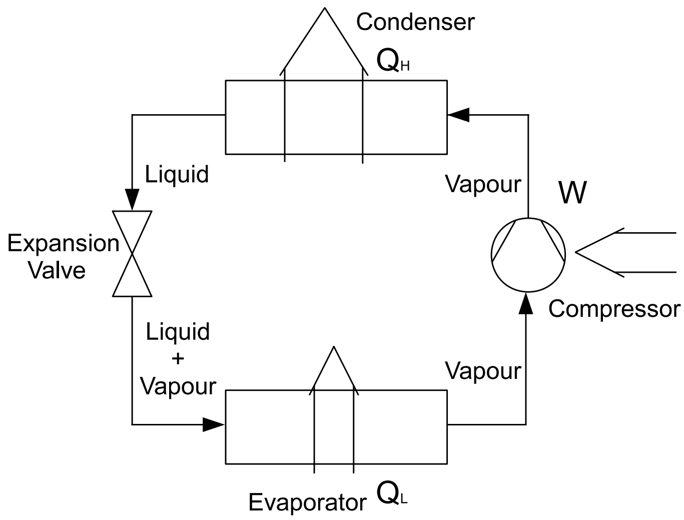

Figure 4: *The principle of a vapour compression refrigeration cycle.*

The refrigerant undergoes two phase changes during its course around a closed loop. The purpose of these phase changes is that during a phase change from liquid to gas form, the substance absorbs energy (Q_L) at a relatively constant temperature. In the same way, a phase change from gas form to liquid form releases energy (Q_H) which can be absorbed by the environment. The refrigerant evaporates while in contact with the cold space and thus removes energy from the cold space. Thereafter the refrigerant condenses while in contact with the hot space, thus dumping the extra energy.

Heat does not naturally flow in the direction towards higher temperatures. This is the formally stated in the second law of thermodynamics. So, for the refrigerant to be able to remove heat from the colder space, the temperature of the refrigerant needs to be lower than the temperature in the cold space. For the same reason, the temperature of the refrigerant needs to be higher than the hot space while condensing in that region. In order to achieve this, the refrigerant is compressed after it leaves the cold space. When the refrigerant leaves the cold space, it is still cold but have changed to a slightly superheated vapour (the temperature is slightly above the saturation temperature for that pressure), and when the pressure increases in the compressor the temperature is also increased. Thus, the saturation temperature increases, see figure 5 for an illustration of the relationship between saturation temperatures and saturation pressures for two different cooling media commonly used in refrigeration cycles.

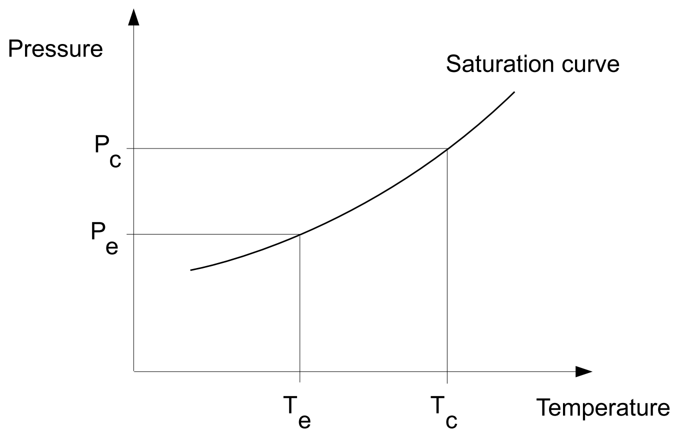

Figure 5: *Illustration the relationship between pressure and temperature for the evaporation point and for the condensation point.*

This cycle process cannot occur without work input, in this case from the compressor. Without the compressor, the saturation temperatures of the refrigerant could not be altered in the same convenient way, and the heat flow direction could not be controlled. The basic energy conversion relation for refrigeration processes can be described according to equation 1. As can be seen in the equation, the heat transfer to the hot region is always more than the heat transfer from the cold region.

$$Q_H = Q_L + W \tag{1}$$

In order to compare different refrigeration systems, a Coefficient Of Performance (COP) was defined by Sadi Carnot in 1824 [7]. The COP is defined as shown in equation 2. In a similar way to standard efficiency calculations, the COP relates the beneficial output, i.e. the actual cooling, to the work added to the system in order to achieve this desired output.

$$\begin{aligned} COP &= \frac{\text{desired output}}{\text{required input}} = \frac{Q_L}{W}, \quad W = Q_H - Q_L \\ \Rightarrow COP &= \frac{1}{\frac{Q_H}{Q_L} - 1} \end{aligned} \tag{2}$$

2.1.1 Processes in the cycle

To identify the thermodynamic changes of the refrigerant during the cycle, the properties temperature (T), pressure (p), enthalpy (h) and entropy (s) are used. The refrigerant undergoes four main changes during the cycle: evaporation (1-2), compression (2-3), condensation (3-4) and expansion (4-1). These four processes alter the properties of the refrigerant in different ways. To illustrate the changes, two main types of diagrams are used: P-h diagram and T-s diagram. The standard appearance of these diagrams for the ideal vapour-compression refrigeration cycle can be seen in figures 6 and 7. The ideal cycle is often used as a comparison to verify the efficiency of real cycles.

Step 1-2: Evaporation. In the ideal case, this process keeps both the temperature and the pressure constant, while enthalpy and entropy increases. For real cycles, this process often increases the temperature slightly as a safety margin so that the refrigerant leaving the evaporator is somewhat superheated. Thus the compressor can be guaranteed to solely compress vapour, and not liquid.

Step 2-3: Compression. In the ideal case the entropy is kept constant during this process. The pressure, temperature and enthalpy increases. In the real case, the temperature increases to a slightly higher level than the saturation temperature (superheated vapour). Furthermore, the entropy is not held constant in the real case.

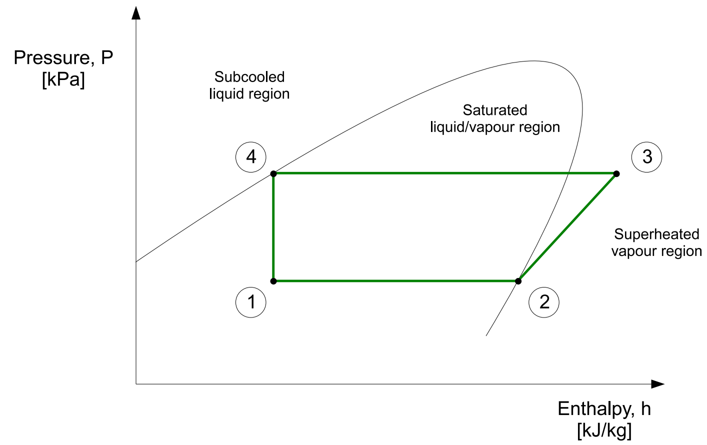

Figure 6: *A basic refrigerant Ph-diagram showing the refrigeration cycle in green.*

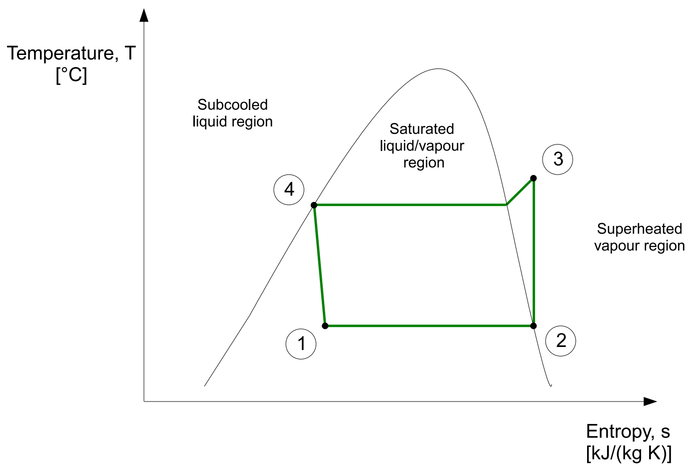

Figure 7: *A basic refrigerant Ts-diagram showing the refrigeration cycle in green.*

Step 3-4: Condensation. In the ideal case the refrigerant enters the condenser unit as a saturated vapour and all of the energy that is released to the environment equals the enthalpy change between saturated vapour and saturated liquid at that pressure level. In the real case, the refrigerant is slightly superheated as it enters and the temperature first drops to the saturation temperature for that pressure level.

Step 4-1: Expansion. In the ideal case the pressure and temperature are reduced in this step without any loss in enthalpy. In the real case, there is a loss in enthalpy as well since the process is never completely isolated from the surroundings.

In the ideal case, i.e. a totally reversible process, the COP can be simplified as shown in equation 3. This form of COP is called the Carnot Coefficient Of Performance. These temperatures are specified in Kelvin.

$$COP_{Carnot} = \frac{1}{\frac{T_H}{T_L} - 1} \tag{3}$$

2.2 Control of Refrigeration Cycles

There are several parts of the refrigeration loop that needs to be controlled in order for the system to work properly, and to ensure that all the different components does not take harm. This section treats different common strategies for controlling the refrigeration cycle and it is based on information from [8] and [4].

One of the risk issues regarding vapour compression refrigeration cycles is the evaporator. In the case of an incomplete evaporation, some liquid would be left in the vapour after the evaporator, but since the compressor usually only handles gas phases, this could damage the compressor. Should the evaporation only be precisely enough to guarantee 100 % gas at the evaporator outlet, pressure drops in the pipes leading the gas to the compressor could result in some of the gas changing back into liquid (since a pressure drop implies an energy drop and the saturation temperature for that lower pressure is lower than for the pressure directly after the evaporator). To minimize the risk of liquid in the compressor, the evaporation process must be regulated and ensure a slight overheating of the vapour to compensate for pressure drops in the pipes.

A common way to ensure that the evaporator produced enough is by using a mechanical solution consisting of a thermobulb at the outlet of the evaporator that checks the evaporator outlet temperature. Should the temperature be too high or too low, the valve inlet to the evaporator is adjusted accordingly.

Another important process of the vapour compression refrigeration systems is the condensation. This must of course also be controlled. For an air-cooled condenser, the process relies on the air temperature, and the air flow through the heat exchanger. The simplest way of controlling this condensation would be to keep air and refrigerant temperatures, plus air flow, constant. This is not always an option. With varying air temperatures, the air flow must also vary to adjust the condensation process.

A common way to adjust to this problem is through condenser fans. One or several fans operate on an on-off basis to ensure sufficient blow-through air for the condensation process. The control of this on-off cycling is usually based on compressor discharge pressure or temperature.

2.2.1 Capacity control of refrigeration systems

To adjust the cooling for varying load conditions, several methods for capacity control of refrigeration systems have been developed. The simplest one is on-off cycling of the compressor. With this method, the compressor is switched on and off repeatedly to generate a cyclic behaviour of the refrigerant flow, and thereby also of the cooling. Some of the other commonly used capacity control methods are hot gas bypass and variable-speed compressors. More about capacity control can be found in [4], [8], [7], [9], [10] and [3], and the information given herein is based on these sources.

Hot gas bypass One way to reduce the number of start-ups and shutdowns of the compressor is to simulate the on-off cyclic behaviour by bypassing the condenser, expansion device and evaporator, and lead the hot and high pressure discharge gas straight back into the suction side of the compressor. This way, the active cooling caused by the refrigeration system is avoided without shutting down the compressor. Cycling this form of bypass behaviour with normal refrigerant flow results in a cooling pattern similar to the one obtained with the on-off control method. Bypassing the hot gas straight back to the compressor inlet leads to an increase in refrigerant pressure and temperature; work is still added to the refrigeration system, but no major heat flow is leading this energy away from the system. To deal with this problem, some of the liquid refrigerant from the low pressure side of the cycle can be injected into the hot bypassed gas to cool it down. From an energy efficiency point of view this hot gas bypass control method is inefficient and wastes energy. The cooling capacity is reduced without reducing the input compressor work.

Variable-speed compressors Most compressors can operate at a different speed than it was designed for, but due to the necessary mechanical lubrication of the compressors they can seldom operate at speeds below 50 % of their design speed. Today, some of the various-speed compressors can go down to 1/3 of their top speed [11]. Compressors can be speed regulated either stepwise or continuously. Continuous variable-speed regulation is efficient for reduced load conditions [4]. The actual speed reduction is usually performed by use of a frequency inverter. Some compressors have built-in speed variation.

As mentioned in [4], there are six main design options to deal with variable-load cases:

1. One large compressor. Results in low efficiency and wasted capacity at part load cases.
2. One large compressor with inbuilt capacity control. A good option for cases when the variable load never goes below 50 % of full load.
3. Three small compressors, of which one has capacity control and the two other's do not.
4. Three small compressors with different capacities. Mix and match according to the current demands.
5. Three compressors with parallel control. Not always a good recommendation due to non-linear input power with capacity turn-down. Inefficient compressors.
6. Three compressors without parallel control.

2.2.2 Frequency control of fans

This section contains information about the theory behind frequency control of fans. It is based on information presented in [12], [13], [7] and [14], and the interested reader is referred to these sources for more details.

The Affinity laws relate a varying speed of a fan (or a pump) to the generated flow, pressure head and power; the relations for flow and power can be seen in equations 4 and 5. There is also a version of the same laws specifying these properties as functions of alternating impeller diameters instead of alternating speeds.

$$\frac{q_1}{q_2} = \frac{n_1}{n_2} \tag{4}$$

$$\frac{Power_1}{Power_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{5}$$

As can be seen in the equations, the relationship between the generated airflow of a fan and the speed of it is linear. This is of course an approximation, since other fluid dynamic aspects also interfere, but it serves as a basis to build on when dimensioning fans. The laws also assume that the fan efficiency for the two operating points is the same.

In refrigeration cycles, speed control of condenser fans generally reduces the power consumption significantly at the same time as it allows for a more flexible control of the condensation process [14]. It is also notable that two condenser fans operating on half speed is sometimes used instead of one fan at full speed during night-time for air conditioning units, in order to reduce sound level at night [7].

2.3 Bus Electronic Architecture

This section will give a short overview of how the Bus Electronic Architecture (BEA) system work, and also a short description of what a Controller Area Network (CAN) is, since it is used in the BEA system. The information used for this section is based on [15] and slides from a BEA course given by Consat Engineering (Jonas Williamsson) at Saab Electronic Defence System.

CAN This network controls and handles data communication that occurs between several electrical control units. Only two wires are needed, CAN high and CAN low. These wires are twisted around each other to suppress noise. At each end of the wire pair, there is a need for a termination resistance, which is connected between CAN high and CAN low.

The communication between the different electrical control units in a CAN system function in the following way:

- The data information (”1” and ”0”) is arranged in a frame.
- Each frame has an identifier which is unique.
- One frame forms a message that is used for communication.
- When one control unit sends a message, the others listen.
- CAN has a protection against message collision.

AB Volvo developed the Bus Electronic Architecture (BEA) to manage the complexity increase of electrical systems in busses. It is a multiplex system, i.e instead of having one control unit there are several control units distributed over the system. The BEA system is built up by reprogrammable hardware modules and they communicate through a CAN interface. This improves the flexibility, reduces the cabling complexity and it also creates the possibility to have electronic fault diagnostics.

BEA consist of two parts:

1. Five different hardware modules
2. The software development environment Volvo Multiplex Toolbox - VMT

2.3.1 The hardware modules

All of the five BEA modules (MasterID, CECM, I/O A, I/O B, and AIC) are reprogrammable nodes (connection points). Their functionality and behaviour in the electrical system is programmed in software.

MasterID The MasterID is the interface between the electrical systems and the operator. It has a memory back-up of the application software, which ensures that all the rest of the modules in the multiplex system have the correct software. It also has a built-in web server that gives the possibility for diagnostics and fault-searching for the electrical system. Any PC can be connected to this module and gives the possibility to easily get information about the system such as fault diagnostics if something went wrong. Faults can be seen in the MasterID web interface fault log, but the fault log is erased if the power to the CECM is cut. It has no inputs nor outputs.

CECM - Central Electronic Control Module It is the systems main computer and has 19 inputs and 17 outputs. The CECM controls and monitors the communication with the I/O modules, to ensure that the correct software is installed in the correct module.

I/O A - Input/Output Model type A It is a reprogrammable module with 15 inputs and 7 outputs. Its main use is to register inputs, such as sensors, and also to supply low power loads. The highest total output current of this module can be 3 A.

I/O B - Input/Output Model type B It is a reprogrammable module with 7 inputs and 10 outputs. Its main use is to drive high-current and analog outputs. The analog outputs are always PWM (Pulse Width Modulation). The highest total output current of this module is 30A.

AIC - Advanced Information Centre The module is a computer with a display. This computer registers statistical data about the multiplex system that can be of interest for the user. The display gives the possibility to show information to the user about the performance of the electrical system and also to control settings.

The BEA system has a "Plug and Play" function which imply that if any of the modules, CECM, I/O - A or I/O - B, breaks down they can easily be replaced by a new one. If any of the modules is replaced the system will be configured automatically and operate normally.

The different input and output types can be overviewed in table 1.

Table 1: *The inputs and outputs which are available in the BEA multiplex system.*

Inputs		**Outputs**	
HDI	High Digital Input	HDO	High Digital Output
LDI	Low Digital Input	LDO	Low Digital Output
HLDI	High/Low Digital Input	HLDO	High/Low Digital Output
AI5V	Analogue input	HPO	High PWM Output
RAI	Relative Analogue Input	LPO	Low PWM Output
RES	Resistive input	5V ref	5V reference voltage
Cntr	Pulse counter		

Note that only voltage of the value $0 - 5000$ mV is possible in AI5V. Current is not possible to have as an input, instead it has to be changed into voltage.

In figure 8 it is shown how the different modules are connected and the connection to a PC is also included.

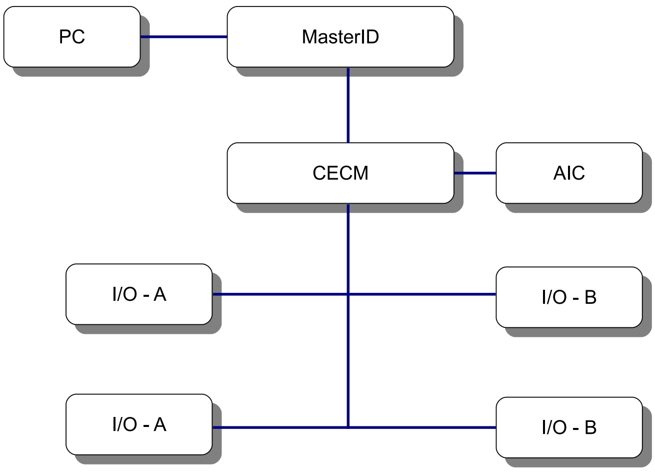

Figure 8: *Illustration of how the BEA modules and a PC are connected.*

2.3.2 VMT

VMT consist of the VMT editor and the VMT server. The editor consist of an XML file editor, a function programming editor (ISaGRAF) and a compiler. The programming tool makes it possible to create software that can be downloaded to the modules. There are also functions to test and debug the created software.

The server handles revision history and updates files needed for the VMT. In the server there is a "component database" where all the created functions and configurations are safely stored. This enables that multiple developers can work on the same project.

3 The Liquid Cooling Unit

This section gives information about the LCU system and its control system, with information provided from internal documents at Saab.

The Liquid Cooling Unit (LCU) consist of three main parts, the condenser unit, the compressor unit, and an expansion tank, see figure 9. In the condenser unit, there are four fans to circulate air in a heat exchanger. This air is used to cool the EGW and the refrigerant.

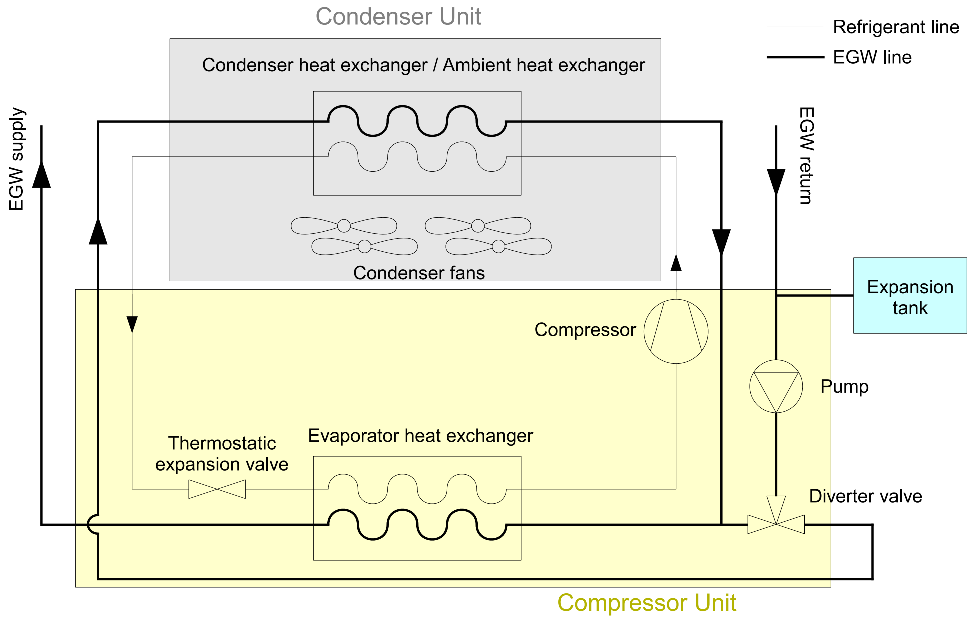

Figure 9: *Illustration of the three parts that the LCU consist of, the condenser unit, the compressor unit and the expansion tank.*

In the compressor unit there is a pump, which pump EGW through the system, a compressor to increase the pressure and maintain the flow circulation of the refrigerant and an evaporator heat exchanger. Before the refrigerant enters the evaporator heat exchanger it passes through a thermostatic expansion valve. The expansion tank is connected to the EGW in the compressor unit. This tank allows the EGW to expand or contract due to changes in pressure and temperature, and makes it possible to add fresh EGW to the system.

There are two different filters in the compressor unit, one for each fluid. Their main task is to remove unwanted particles or other contaminations. In the compressor unit there is also a liquid receiver incorporated before the filter of the refrigerant to act as a buffer and thus controlling a steady flow into the evaporator heat exchanger. At the same time it prevents reverse flow back into the condenser heat exchanger.

3.1 Controlling The LCU

There are two main principles of operation for the LCU today, Ambient mode and Active mode. In the Active mode, the refrigerant is processed through the system by the use of the compressor to actively cool the EGW. In Ambient mode, the EGW is solely cooled by the use of the ambient air in the heat exchanger of the condenser unit. No refrigerant is circulating, i.e the compressor is off.

The switching between Ambient mode and Active mode is done by a diverter valve, see figure 3. If the LCU is in Ambient mode it switches to Active mode if the ambient temperature increases to $0°C$, and if the LCU is in Active mode it switches to Ambient mode if the ambient temperature decreases to $-5°C$, see figure 10. The compressor is turned off at an ambient temperature of $-6°C$, if the LCU has switched from Active to Ambient mode. If the LCU switches from Ambient to Active mode the compressor is turned on at an ambient temperature of $-1°C$, see figure 10.

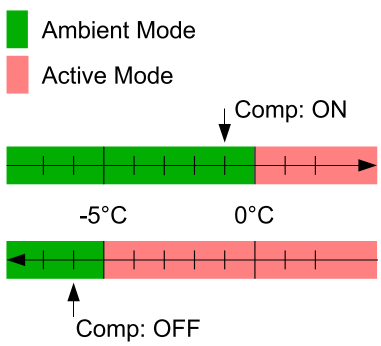

Figure 10: *Illustration of when the controller switches between Ambient and Active mode due to the ambient temperature. And when the compressor is turned on and off due to mode switches.*

In the control today, the four fans operate in pairs and in an on-off manner. This leads to three distinctive fan operation modes: No fans, two fans, or all four fans. The switching between these fan operation modes is based on refrigerant pressure in the case of Active mode, and in the case of Ambient mode the switching is based on EGW supply temperature.

3.2 Ambient Mode

In Ambient mode the refrigerant is not used at all, thus the compressor is off to save energy. The cooling of the EGW is done in the condenser unit using the ambient air, see figure 11. To increase the heat transfer, the fans can be switched on, thus pumping more air through the heat exchanger. The control today is based on signals from a temperature sensor mounted at the supply end of the EGW line (the EGW leaving the LCU). At some temperature trigger level, the first pair of fans is switched on. At a slightly higher temperature trigger level, the second pair is also switched on. These trigger levels are shown in table 2.

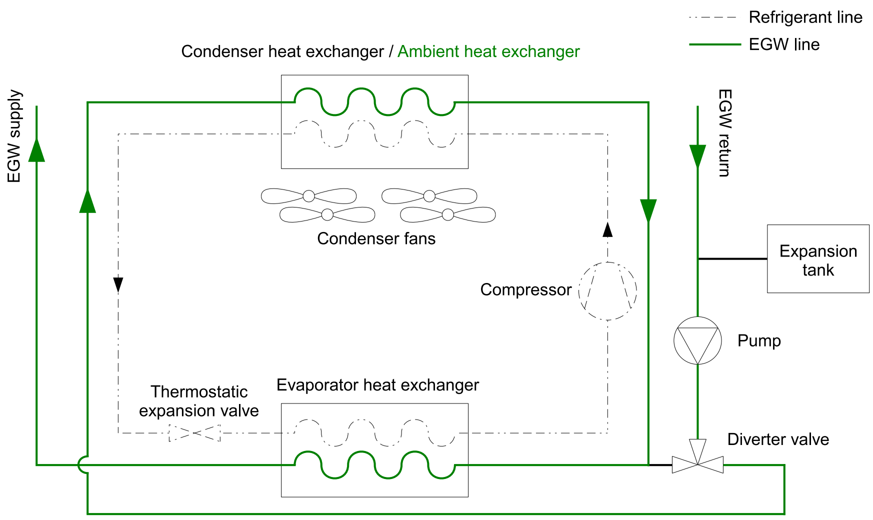

Figure 11: *Illustration of Ambient mode. The EGW follows the green line.*

In ambient mode, the control is said to maintain the EGW supply temperature at $11 \pm 5°C$.

Table 2: *The trigger temperatures controlling the fans in Ambient mode.*

	Fan Pair 1	**Fan Pair 2**
ON	$> 11.5°C$	$> 14°C$
OFF	$< 6°C$	$< 10.5°C$

3.3 Active Mode

In Active mode the compressor is on and induces a flow of the refrigerant. In this mode the EGW is cooled by the refrigerant in the compressor unit instead of the ambient air in the condenser unit, see figure 12. The refrigerant cycle is thoroughly described in section 3.5.

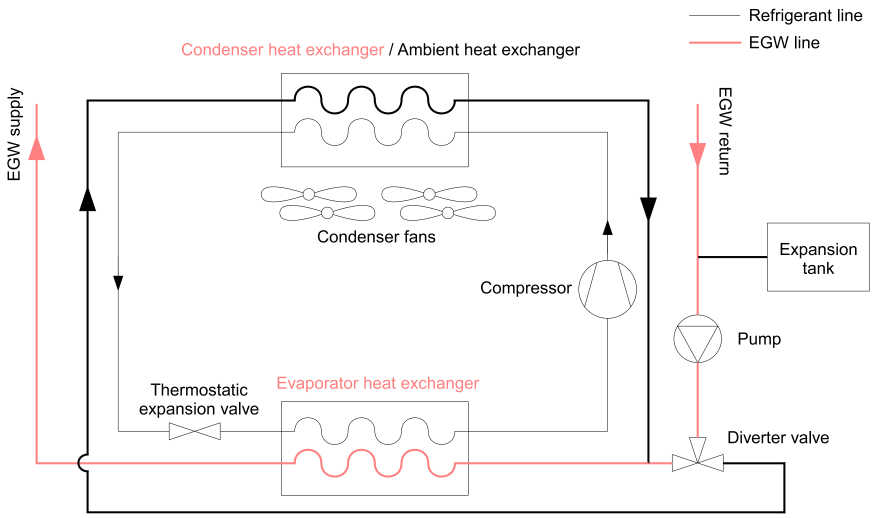

Figure 12: *Illustration of Active mode. The EGW follows the salmon line.*

The control is said to maintain the EGW supply temperature at $5 \pm 2°C$. It should be noted that the fans in this mode control the cooling of the refrigerant, and not the EGW as the case is for Ambient mode. It should also be noted that the system is required to supply a colder EGW while in Active mode compared to when in Ambient mode. This is due to the assumption that more cooling is required in the vehicle at higher ambient temperatures.

3.4 Redundancy Mode

There is a Redundancy mode that is used when a failure in the vapour cycle refrigeration system has been detected while in Active mode. It is activated when the EGW supply temperature is more than $14°C$ and the return EGW temperature is at least 4 degrees higher than the ambient temperature. While in Redundancy mode, the EGW follows the same path as in Ambient mode, i.e. it is cooled in the condenser unit heat exchanger instead of in the evaporator heat exchanger. The fans are in this case controlled in the same manner as when in Ambient mode.

3.5 The Refrigerant Line

The compressor starts the refrigerant cycle by compressing the fluid. The superheated gas is thereafter led into the condenser heat exchanger where it is cooled by the ambient air (with or without help from the fans). The gas condenses in this process, and the output from the condenser is thus liquid refrigerant at almost the same high pressure. The liquid is led to a liquid receiver, see figure 13, which acts as a buffer to both keep the flow to the evaporator at a constant level, and to prevent any liquid from being pushed back into the condenser heat exchanger. As the liquid leaves the liquid receiver, it enters a filter/drier, see figure 13, where unwanted particles are removed from the refrigerant and contaminants and/or water is removed. After the filtering of the refrigerant there is also a sight glass to allow for visible verification of the quality of the refrigerant and to check the chemical substance, see figure 13.

The liquid line solenoid valves control the change of path of the refrigerant, between the Active Cooling loop, see figure 14, and the Bypass loop, see figure 15. Before the liquid enters the evaporator heat exchanger, it passes through a thermostatic expansion valve where it is expanded to a lower pressure. In this process some of the liquid evaporates naturally since the fluid now is close to its boiling temperature for this specific pressure. To make sure that all the liquid entering the evaporator heat exchanger is evaporated into gas the thermostatic expansion valve is controlled by a thermobulb connected to the outlet of the heat exchanger, comparing the temperature of the fluid before (via a capillary tube) and after the heat exchanger. The outlet temperature of the refrigerant is thereby maintained at a level of $5 \pm 4°C$. The slightly superheated refrigerant is sucked into the compressor and thus the cycle is completed. The path through the evaporator heat exchanger, see figure 14, of the refrigerant is called Active cooling, see the part about Hot gas bypass in section 2.2.1.

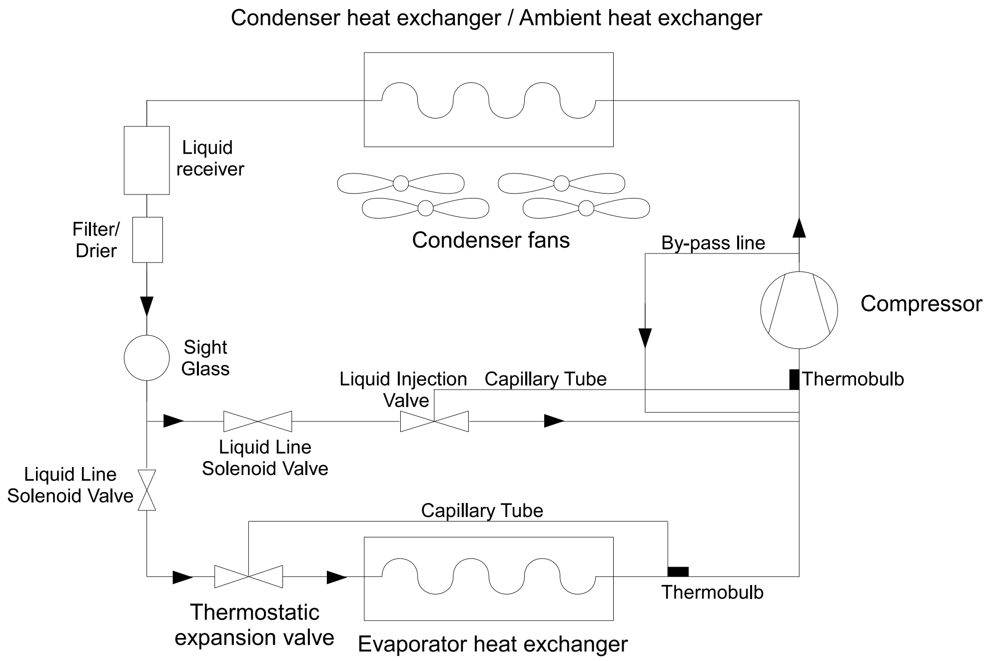

Figure 13: *Illustration of the refrigerant line and its components.*

The fans can help in cooling the refrigerant. The control of the fans is based on a pressure switch mounted at the outlet of the compressor see figure 16. At a set pressure level, the first pair of fans is switched on, and at a second higher pressure level, the second pair is also switched on. The set pressure levels are shown in table 3. When the return EGW does not need to be cooled the refrigerant bypasses the evaporator through a liquid line where it is evaporated by mixing it with refrigerant vapour coming out from the compressor. This vapour is connected to the liquid line through a hot gas bypass line. This path see figure 15, where the refrigerant bypasses the evaporator heat exchanger is called Bypass, see the part about Hot gas bypass in section 2.2.1. The temperature of the refrigerant at the inlet of the compressor is controlled in the same way as in the evaporator heat exchanger, with a thermobulb and a capillary tube.

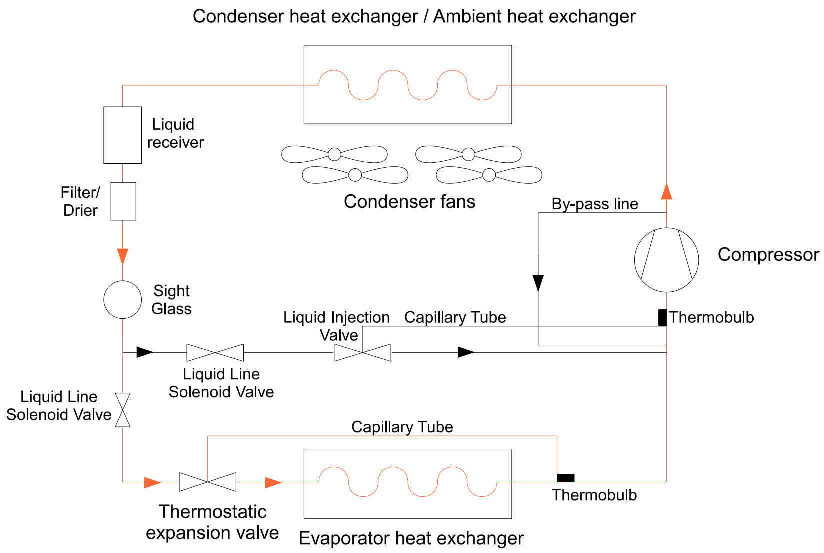

Figure 14: *Illustration of the refrigerant path when in Active Cooling.*

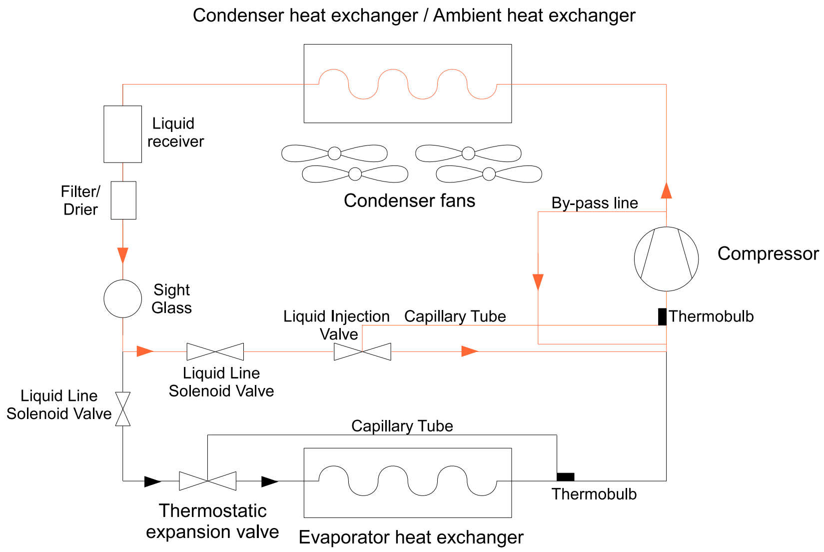

Figure 15: *Illustration of the refrigerant path when in Bypass.*

Table 3: *The trigger pressure values controlling the fans in Active mode.*

	Fan Pair 1	**Fan Pair 2**
ON	$>$ 110 psi	$>$ 220 psi

3.6 Sensors

There are two main properties that are measured in order to control the LCU, temperature and pressure. For more information about how different sensors work see appendix A.1.

There are two RTD sensors in the EGW line. One is placed in the condenser unit to measure the ambient temperature, see figure 16. This temperature information is used for the control of the diverter valve. The other one is placed at the EGW supply outlet and it is used to control the fans when in Ambient mode, see figure 16. The trigger values of the fan control is shown in table 2, in section 3.2.

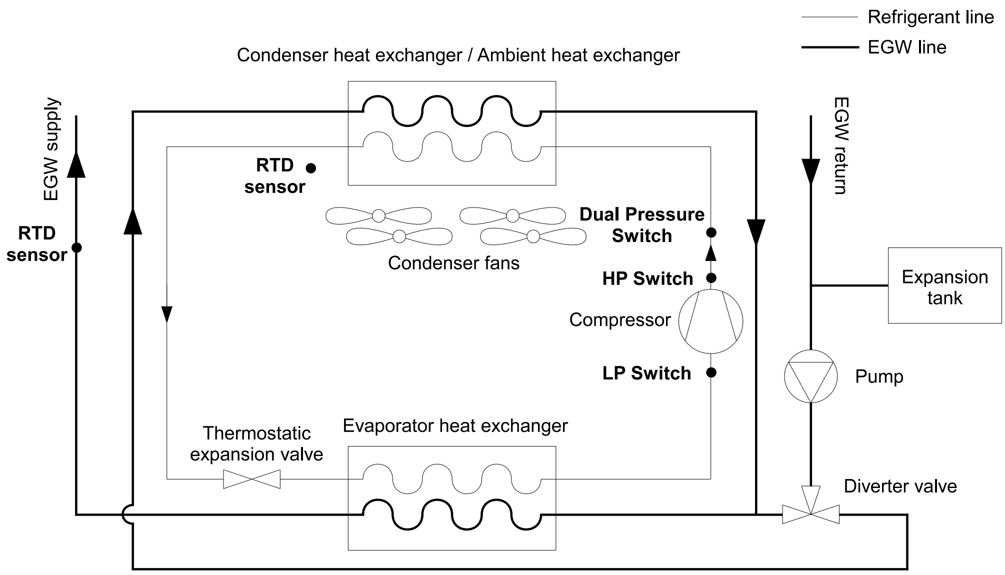

Figure 16: *Illustration of the positions of the sensor in the LCU.*

In order to monitor the behaviour of the refrigerant line, a dual pressure switch is attached to the outlet side of the compressor, see figure 16. This is set to signal as two different trigger values are reached, as described in section 3.5 and shown in table 3.

In addition to this dual pressure switch there are two other pressures switches. One is set to signal at dangerously high pressure (HP switch) at the outlet of the compressor, see figure 16 and the other is set to signal at dangerously low pressure (LP switch) at the suction side of the compressor, see figure 16 and see section 3.7 for more details.

3.7 Additional Safety Measures

To protect the compressor in the refrigerant line from too low or too high pressures, there are pressure switches at the inlet and the outlet of the compressor, see figure 16. These sensors signal a warning when certain limits have been exceeded, as listed in table 4, and if this happens the compressor is turned off. If the compressor is turned of due to too high pressure, the fault indicator HP FAULT is illuminated, and if it is due to too low pressure, the fault indicator LP FAULT is iluminated. When in Ambient mode, these warning signals are switched off, since a lower pressure naturally occurs when the compressor is not operating.

The pump controlling the flow of the EGW keeps a differential pressure between return end and supply end of about 300 kPa (3 bar). As a safety measure, there is a pressure sensitive valve linking the return end with the supply end which opens if the differential pressure exceeds 400 kPa (4 bar). In this way, an automatic pressure relief function is included for the EGW loop.

After the EGW line pump, the liquid enters a flow switch. This switch monitors the flow and sends a signal if the flow falls below a certain threshold value. It also allows the expansion tank to be filled without interfering with the LCU cycles.

Table 4: *The trigger pressure values to turn off the compressor when potentially harmful pressure levels are detected.*

	High Pressure	**Low Pressure**
Fault indicator	$> 340 \pm 0.7$ psi	$< 2 \pm 2$ psi

4 Experimental Tests

Measurements were carried out on the LCU control system to analyze the system. Temperatures and pressures were measured, and also signals from different relays on the PCB were measured for the understanding of the system. The noise level from the fans as they work today and also as they are frequency controlled were of interest to investigate and measurements were carried out. Measurements with a pressure setpoint were also carried out, to investigate how the frequency control of the fans affect the system.

It was not possible to carry out experimental tests by frequency controlling the compressor, due to the fact that it could only be controlled in the interval of 45 - 60 Hz (Patrik Larsson at Refrico AB, 2010). This is no enough to achieve any significant information about the effect it would have on the system since the Hot gas bypass still would be necessary.

4.1 Experimental Setup

The setups for the measurements on the LCU control system and the noise level measurements are illustrated and explained in this section. The equipment used for the two different setups are also listed. The electrical connections done for the experiments are illustrated and explained as well.

4.1.1 Setup for measurements on the LCU control system

The tests were made with thermocouples and pressure sensors attached to different locations on the LCU. A styrofoam sauna (styrofoam walls and roof with a sauna aggregate) was built around the LCU to increase the surrounding temperature and a heater was attached to the EGW line to increase the load. A view from above of the positions of the different parts used in the experimental setup can be seen in figure 17.

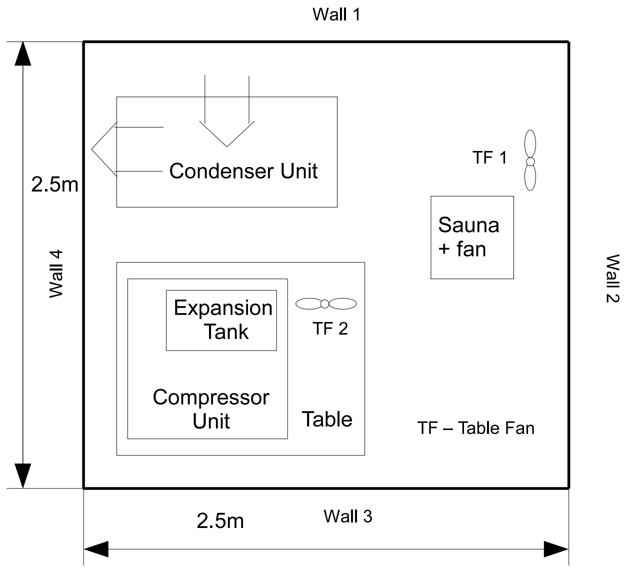

Figure 17: *The experimental setup for the measurements on the LCU control system, seen from above. This illustration is not according to scale.*

The following list consist of the equipment used for the tests on the LCU control system.

- A complete LCU, which includes a condenser, a compressor unit and an expansion tank for the EGW (item 5, item 2 and item 1 in figure 18).
- A table (on wheels) with the compressor unit on it, and the expansion tank (item 1 in figure 18) placed on top of the compressor.
- Beneath the table, with the compressor unit and the expansion tank on, there is a tank to create a delay in the EGW flow (item 3 in figure 18).
- 20 liter EGW mixture of 55 vol% ethylene glycol, and 45 vol% water.
- A heater (4 kW) to heat the EGW once it has been cooled by the LCU (this is to simulate load from the vehicle). The heater is item 4 in figure 18.
- A power supply, *LTronix* PowerSupply B32-20R (Sn: GS 2689) set to 28 VDC to feed the LCU.
- A power supply, *Powerbox 3100* (Sn: 9711121-104797) set on 27 VDC to feed the pressure sensors.

- 10 thermocouples (type K) to measure the temperatures.
- 3 different pressure sensors. A *Foxboro* pressure transmitter (piezoresistive sensor, SN: NEA 16E819B, mechanical connection 1/4" − 1/2") with a working range of 0-50 bar. One *MBS 3000* pressure transmitter (piezoresistive sensor, SN: 060G1430, mechanical connection G 1/4 A), with a working range of 0-25 bar and a second *MBS 3000* pressure transmitter (piezoresistive sensor, SN: 060G1105, mechanical connection G 1/4 A), with a working range of 0-40 bar. All three sensors have current as output.
- An *Agilent* Data Acquisition/Switch Unit (SN: 34970A), with a 20-Channel Armature Multiplexer (SN: 34901A) to register temperatures, pressures and control signals. *BenchLink* Data Logger Software was included and was used to handle the measured and registered data.
- A *Tylö* sauna aggregate, type A8 (SN: 26699847, 8 kW) to heat up the surrounding air, with an *Ebm Papst* fan (SN: W2E200-D138-05, 64/78W, 2600/2900 rpm) mounted 20 cm above the aggregate to circulate the heated air.
- 4 walls (á 6.25m^2) and a roof (6.25m^2), all five are made of styrofoam, to isolate the LCU test area, see figure 17 and figure 18. Together with the sauna aggregate, the experimental area became a styrofoam sauna.
- Two *Biltema* table fans (art.n: 35-972, 30 W, 30 cm wing diameter), see figure 17 and figure 18, to make the heated air from the sauna aggregate circulate within the styrofoam walls.
- A laptop PC for storing the registered temperatures, pressures and signals.

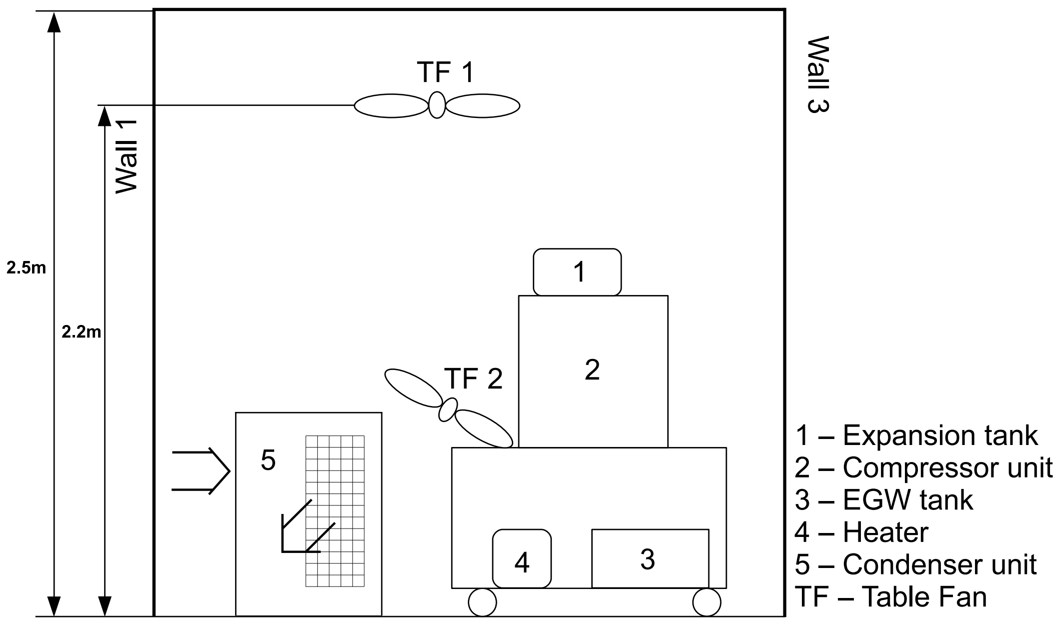

Figure 18: *The experimental setup for measurements on the LCU control system, seen from the side of wall 4. This illustration is not according to scale.*

Figure 19 shows the physical measurement locations of where the different temperatures, listed in table 5, and pressures, listed in table 6, were measured.

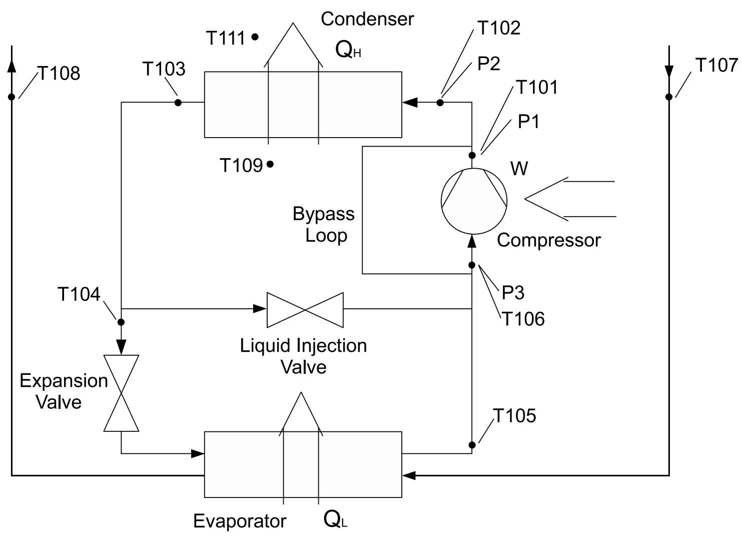

Figure 19: *Illustration of the different measuring points for temperatures and pressures, see table 5 and table 6.*

The 10 different temperatures that were measured in the LCU, through the logger, are listed in table 5.

Table 5: *The different temperatures measured in the LCU.*

T101 $[°C]$	Temperature after the compressor.
T102 $[°C]$	Temperature before the condenser.
T103 $[°C]$	Temperature after the condenser.
T104 $[°C]$	Temperature before the thermostatic expansion valve.
T105 $[°C]$	Temperature after the evaporator.
T106 $[°C]$	Temperature before the compressor.
T107 $[°C]$	The EGW return temperature.
T108 $[°C]$	The EGW supply temperature.
T109 $[°C]$	The air temperature at the input of the condenser.
T111 $[°C]$	The air temperature at the output of the condenser.

At three different points in the refrigerant line, P1, P2 and P3, see figure 19 and table 6, the pressure of the refrigerant was measured. All three pressure sensors act as current transmitters, but the logger only had two inputs for current signals, thus one of the pressure sensors was connected to a voltage input in the logger.

Table 6: *Measured pressures in the refrigerant line.*

P1 [**mA**]	Pressure after the compressor within the bypass loop.
P2 [**V**]	Pressure before the condenser.
P3 [**mA**]	Pressure before the compressor within the bypass loop.

The pressure sensors in the measuring points P1 and P3 were individually connected in series with the 27 VDC supply and the two current inputs in the logger, see figure 20.

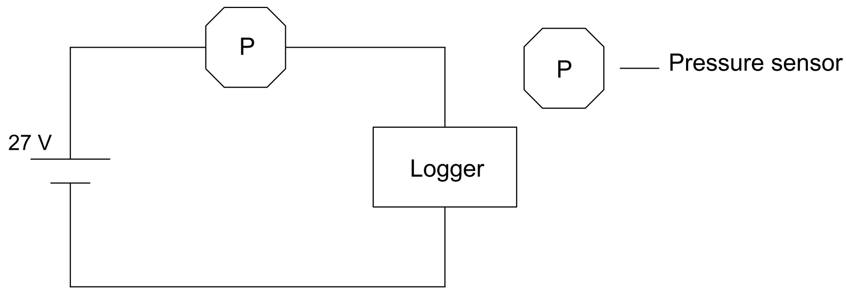

Figure 20: *Circuit diagram of the connection of the pressure sensors, P1 and P3.*

The output from the pressure sensor in the measuring point P2 was changed from current to voltage by means of a resistor in order to connect it to the voltage input of the logger. The sensor was connected in series with the 27 VDC supply and the logger, as the other two pressure sensors. To measure the voltage from the sensor a resistance, R, with the value of 250 Ω, was connected in parallel to the voltage input of the logger. Figure 21 shows how the pressure sensor in P2 was connected. The pressure data from all three pressure sensors were registered in amperes and volts, and could later on be transformed into psi.

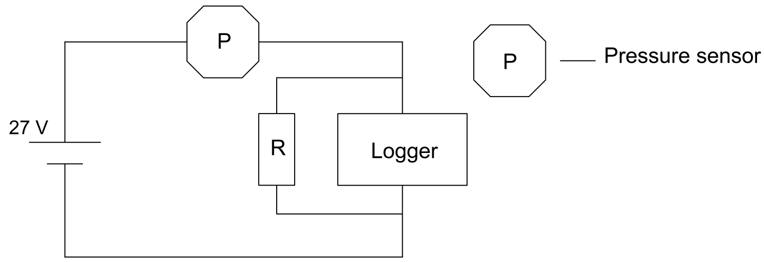

Figure 21: *Circuit diagram of the connection of the pressure sensor P2, where $R = 250\ \Omega$.*

The control signals for fan pair 1, fan pair 2 and Active Cooling were measured over their individual relays on the Printed Circuit Board (PCB), and directly connected to one voltage input each on the logger.

Table 7: *Measured signals from the PCB.*

S113 [V]	Signal from fan pair 1.
S114 [V]	Signal from active cooling (if 0 then bypass loop active).
S115 [V]	Signal from fan pair 2.

4.1.2 Setup for noise level measurements

The noise levels from the condenser fan were measured. During these tests the compressor unit was off, and disconnected from the condenser fans (soft starters were disconnected), thus the voltage supply to the fans was external.

The equipment used is listed below:

- The condenser unit of the LCU.
- A compact acoustical analyzer, *Acoustilyzer*, was used as a microphone to measure the condenser fan noise.
- A frequency converter, *NFO Sinus* (SN: BL 220027, 3-phase 400 V, 15 kW), was connected to the condenser fans to control their frequency during the tests.

- A power supply, *Powerbox 3100* (Sn: 9711121-104797), to feed the fans, through the frequency converter, to control their frequency.

The settings on the Acoustilizer, microphone, were:

Function: SPL/RTA. Sound Level Meter, Actual Sound Pressure Level.

Filter: RLB - Revised Low Frequency B-curve. Applicable for loudness measurements, and correlates the best to the subjective experience by humans.

The microphone was placed at a hight of 0.88 m, and at a distance of 1.4 m from the condenser, see figure 22. The power supply was connected as voltage input to the frequency converter, and the input setting for voltage was $0 - 10$ V. The maximum output frequency was set to 50 Hz, thus 0 V corresponds to 0 Hz and 10 V corresponds to 50 Hz. This made it possible to control the frequency of the fans by changing the input voltage to the frequency converter.

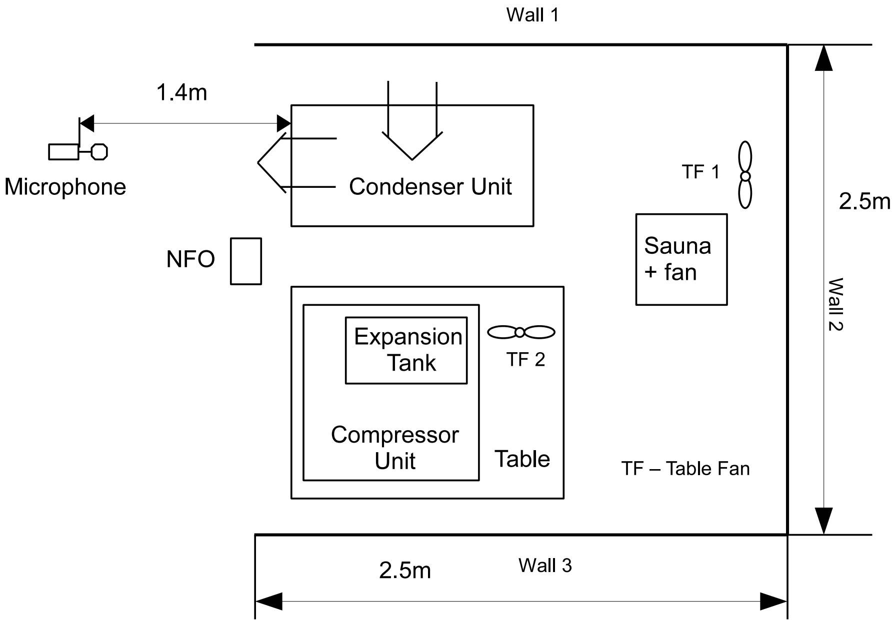

Figure 22: *The position of the microphone for the noise level test. This illustration is not according to scale.*

4.1.3 Setup for pressure setpoint measurements

The built-in PI controller of the frequency inverter was tested and the results measured. During this test the compressor unit was on, but the LCU control system was disconnected from the condenser fans, and the voltage supply to the fans came from the frequency inverter.

The measurement setup resembles the one described in section 4.1.2 on most points, except for that no microphone was used in this setup. The equipment used is listed below:

- The condenser unit of the LCU.
- The compressor unit of the LCU, although the power supply connection to the condenser unit was disconnected.
- A frequency converter, *NFO Sinus* (SN: BL 220027, 3-phase 400 V, 15 Kw), was connected to the condenser fans to control their frequency during the tests.
- A power supply to feed the fans, through the frequency converter, to control their frequency.
- The same temperature sensors as described in table 5.
- The pressure sensor at position P1, as described in table 6.

The settings on the frequency converter were as described in table 8.

Table 8: *Frequency converter settings for the pressure setpoint measurements.*

ControlMode	PI-reg	**RegKp**	1.00
ParGroup	PI-reg	**RegTi**	5.0s
OpMode	Fix-1F	**MinFr**	0 Hz
R-Fix1	8,6	**MaxFr**	50,0 Hz
ActMin	0 bar	**Unit**	bar
ActMax	50 bar	**AinAct**	0-10V
RegAmp	-1		

4.2 Measurements on Present Control System

The measurements on the current system were carried out at three different surrounding air temperatures, 23°C, 30°C and 40°C. For each temperature there were tests performed with two different loads, one with no load and one with the heater on (4kW load). To achieve higher temperatures in the isolated experimental area, the sauna aggregate was on. To fasten up the heating process, and to achieve an even temperature in the air and in the equipment, the heater was turned on during this process. Only the pump in the LCU was active during the heating process, to keep the EGW circulating, which enabled the heating of the EGW to be evenly distributed.

In the styrofoam sauna the free space was very limited, which caused problems for the outlet air of the condenser, since it had nowhere to go except back into the condenser. This situation did not resemble its real and normal working conditions. Therefore the wall 4, seen in figure 23, was moved to create an opening for the air outlet of the condenser, to enable the heated air to leave the condenser and the styrofoam sauna.

There was also a need for preventing the outlet air of the condenser to be sucked in as inlet air to the condenser again. For this reason a special construction (made of particle boards and a black plastic bag, see figure 24) was made for regulating the direction of the airflow, to resemble its real working situation.

When tests were carried out at the temperature 40°C, it was necessary to put a particle board above the outlet of the condenser, seen in figure 25. This was done to inhibit the heated air within the styrofoam sauna to be mixed with the lower temperature on the outside, thus reach the desired temperature level and keep it stable.

The two table fans and the fan mounted on the sauna were always turned on, to maintain the same airflow in the styrofoam sauna when the measurements where performed.

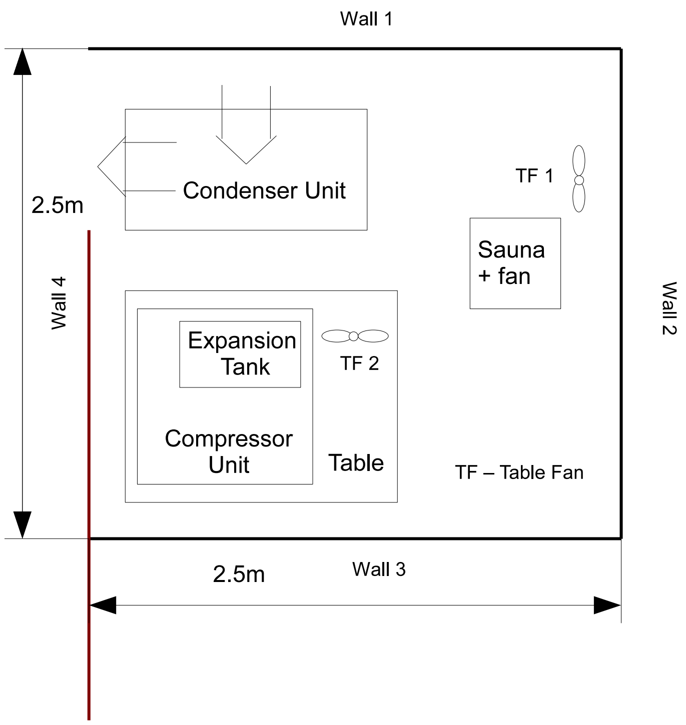

Figure 23: *Wall 4 (the red one) was moved to enable the outlet air of the condenser to leave the styrofoam sauna. This illustration is not according to scale.*

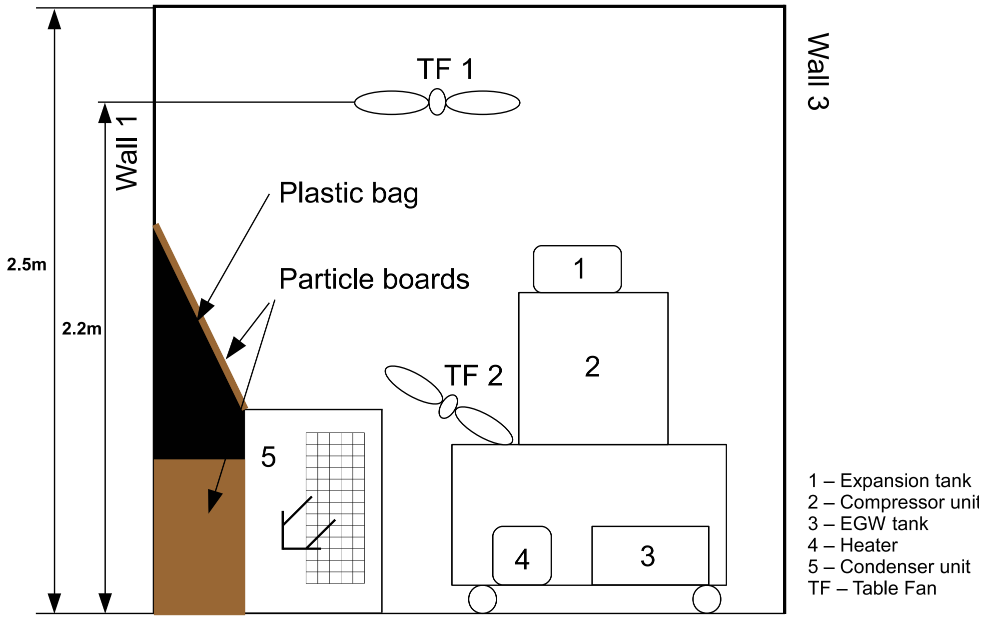

Figure 24: *Illustration of the plastic bag and the particle board construction to regulate the direction of the airflow, seen from wall4. This illustration is not according to scale.*

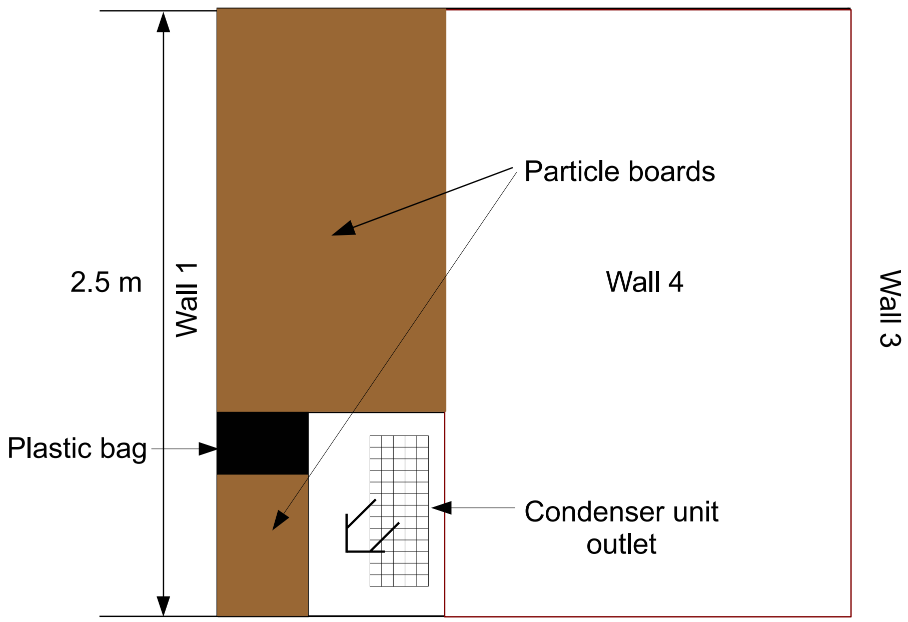

Figure 25: *Illustration of the position of the particle board to maintain the styrofoam sauna temperature at* $40°C$*, seen from wall 4. This illustration is not according to scale.*

4.2.1 Surrounding air temperature at approximately $23°C$

For measurements at a surrounding temperature of approximately $23°C$, there was no need for the sauna aggregate to be on, since the surrounding temperature at the location was already around the desired level. There were two tests carried out. One with the heater off, see graph in figure 26, and one with the heater on, the graph is shown in figure 28. At this temperature only fan pair 1 switched on.

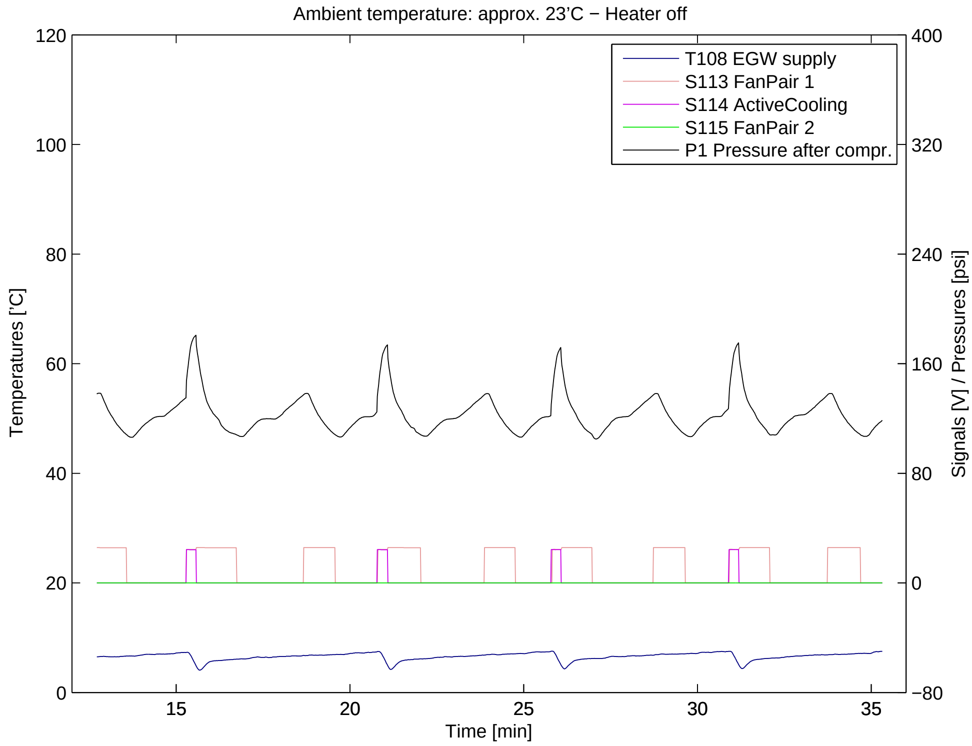

Figure 26: *Graph of the temperatures, pressures and signals measured in the LCU, at* $23°C$ *without the heater. The vertical axis on the left side shows the temperature levels and the vertical axis on the right side shows the pressure and signal levels.*

Heater Off In figure 26 the behaviour of the control system from the first test can be seen. When the refrigerant is bypassed (S114 = 0 V), the EGW supply temperature (T108) is gradually increasing (there is no Active Cooling of the EGW) and the pressure after the compressor (P1) increases as well. As the EGW supply temperature reaches $7.25°C$ it goes into Active Cooling (S114 = 27 V). At the same time as the switch from Bypass to Active Cooling occurs the pressure, P1, reaches 138 psi triggering the dual pressure switch and the fan pair 1 is turned on (S113 = 27 V), see signal S113 in figure 26. As the refrigerant begins to go through the evaporator heat exchanger, see figure 13, it is heated by the returning EGW, thus the pressure increases rapidly. The rate of this dynamic is shown in figure 26.

The Active Cooling cools the EGW supply temperature in 0.3 minutes and switches back into Bypass when it is cooled enough, reaching $4.7°C$. Thus one EGW supply temperature cycle is when the temperature increases from $4°C$ to $7.5°C$ and then decreases to $4°C$, see figure 27. For the unloaded case the cycle time is around 5.2 minutes. When the refrigerant switches to Bypass the pressure, P1, has not decreased enough for the fans to go off. The fan pair 1 can also switch on and off while in bypass, due to the pressure increase in the refrigerant loop.

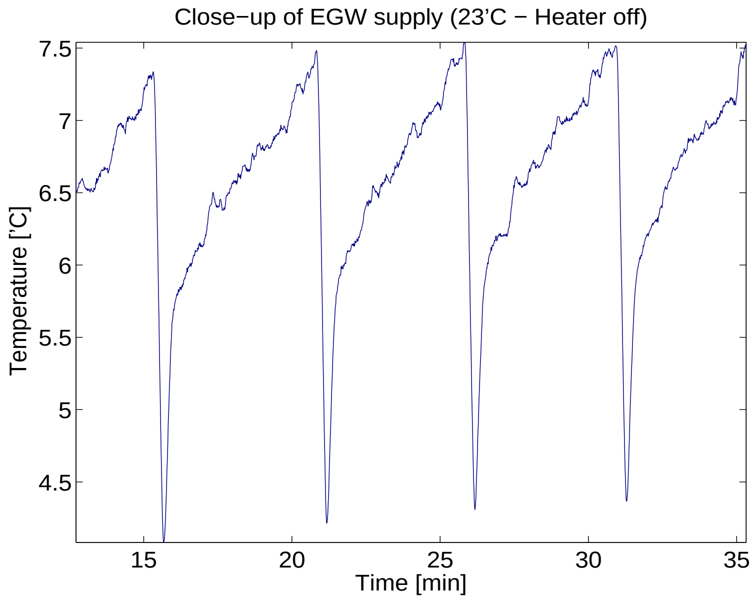

Figure 27: *Graph of the EGW supply temperatures in the LCU, at* $23°C$ *with the heater on.*

The cycles of the pressure, P1, has a clear characteristic, see figure 26. In the Bypass cycle the pressure reaches a peak of 138 psi and due to the maintenance of the appropriate pressure at the compressor outlet the fans are switched on and off. In the next EGW supply temperature cycle the system goes into Active Cooling and the pressure reaches a higher peak of 174 psi. The time between two Active Cooling cycles is around 5.5 minutes. The cycle time for Active Cooling is 0.3 minutes and the time cycle for Bypass is 5.2 minutes. In the figure 26 it looks as though fan pair 1 is turned on at different pressure levels if the refrigerant is in Active Cooling or Bypass.

The reason for this is due to the high increase rate of the pressure P1 when entering Active Cooling and the signal for the fan pair 1 tries to follow.

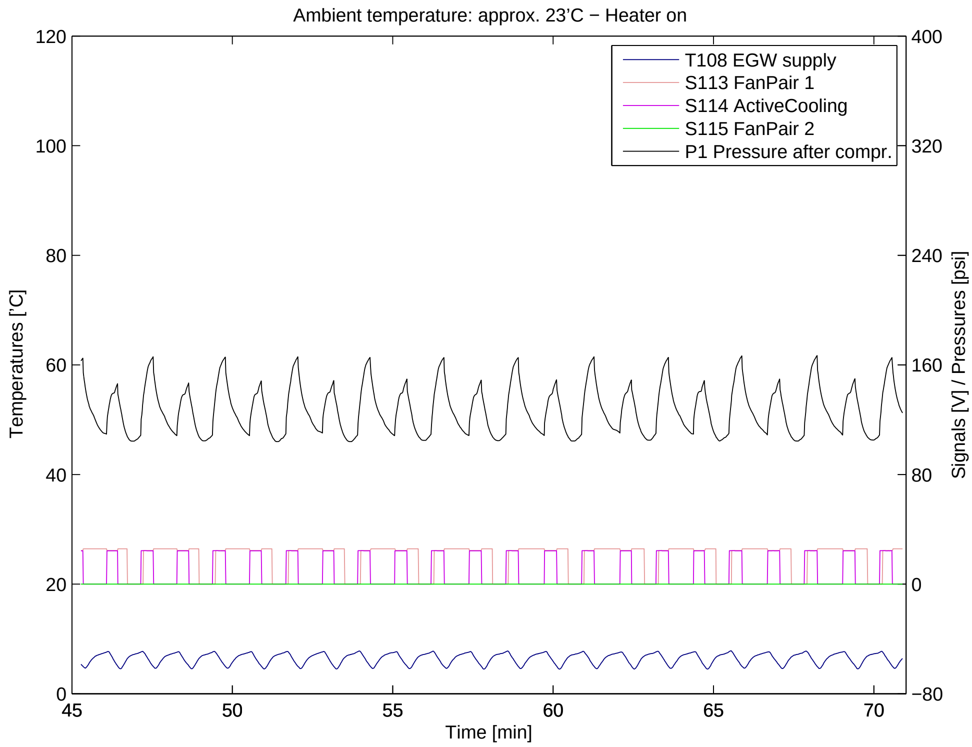

Figure 28: *Graph of the temperatures, pressures and signals measured in the LCU, at 23°C with the heater on. The vertical axis on the left side shows the temperature levels and the vertical axis on the right side shows the pressure and signal levels.*

Heater On The behaviour of the system with a heater on can be seen in figure 28. The EGW supply temperature levels controlling the switching between Active Cooling and Bypass are the same as for the case without the heater. The time between the Active Cooling cycles is around 1 minute, since the EGW supply temperature is heated and its temperature increases 5.5 times faster than without the heater, compare figure 26 with figure 28. The duration of the Active Cooling cycle is 0.4 minutes and that of Bypass is 0.6 minutes.

The dynamic of the pressure, P1, looks different compared to its behaviour without the heater. When the refrigerant enters Active Cooling and fan pair 1 is switched on, the pressure increases and reaches a level of 166 psi. When the T108 is cold enough and the refrigerant enter Bypass, the pressure has not decreased enough to switch off fan pair 1. As it enters the second Active Cooling cycle, the pressure reaches a lower peak of 148 psi. In the next Bypass cycle the pressure decreases below 108 psi and fan pair 1 is turned off. Thus the fan pair 1 is turned off every other Bypass cycle, see signals S113 and S114 in figure 28. When the heater is off fan pair 1 is turned off twice in one Bypass cycle.

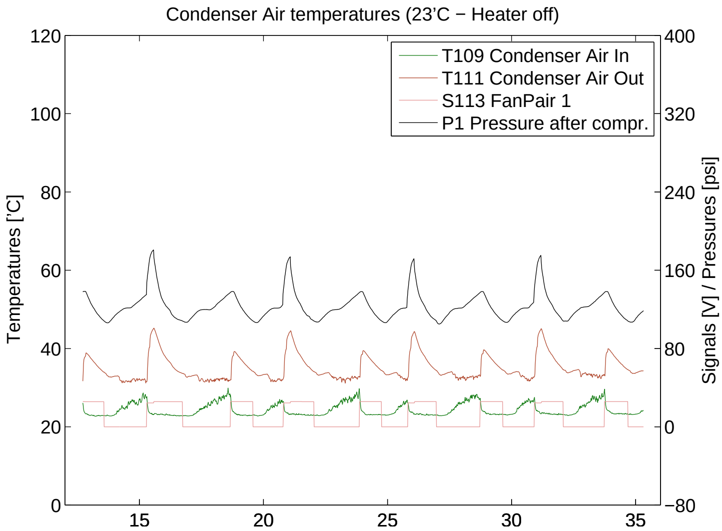

Figure 29: *Graph of the inlet and outlet temperatures of the condenser in the LCU, at 23°C with the heater off.*

Heat exchange due to fan activity The temperature at the outlet of the compressor has a clear relation to the pressure/temperature of the refrigerant after the compressor, see figure 29. When the refrigerant is in Active Cooling ($S114 = 27$ V) the pressure, P1, increases almost instantly to a maximum peak values. The temperature of the refrigerant increases with the pressure increase. As the fans are turned on ($S113 = 27$ V) they remove the heated temperature from the refrigerant in the condenser unit. This heat removal is in relation to the outlet air temperature of the condenser unit, this relation can be seen in figure 29. As the the pressure increases and the fans are turned on, the outlet air temperature increases. When the pressure has reached its peak and starts to decrease the outlet air temperature also follows this behaviour and decreases. From the figure 29 it can also be seen how the styrofoam walls of the experimental setup affects the tests. The inlet temperature slowly increases when the fans are off, the reason for this is the lack of airflow caused by the isolation of the setup. As the pressure slowly increases when in Bypass, it also heat up the still air which is close to the condenser.

4.2.2 Surrounding air temperature at approximately $30°C$

To reach the desired temperature level, the sauna aggregate was turned on as well as the heater. When the temperature level of $30°C$ was stable in the styrofoam sauna (both air and fluids), the LCU was turned on. Firstly, the measurements when the heater was on (its steady state is shown in the graph in figure 30) were carried out. Secondly the heater was turned off, to carry out the measurements without any load (its steady state is shown in the graph in figure 31). At this temperature only fan pair 1 was turned on.

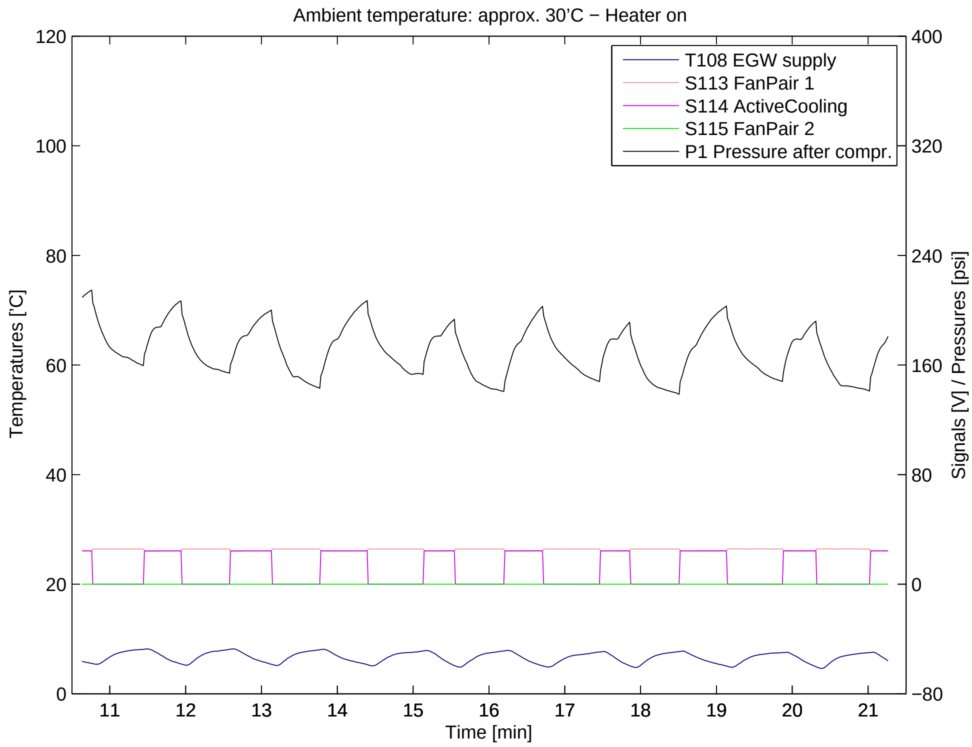

Figure 30: *Graph of the temperatures, pressures and signals measured in the LCU, at $30°C$ with the heater on. The vertical axis on the left side shows the temperature levels and the vertical axis on the right side shows the pressure and signal levels.*

Heater On The behaviour of the system with a heater on can be seen in figure 30. The EGW supply temperature levels controlling the switching between Active Cooling and Bypass are the same as for the previous tests. The EGW supply temperature cycle has a time duration of 1.2 minutes (5 cycles in 6 minutes). Thus the refrigerant switches to Active Cooling every 1.2 minutes and the Active Cooling cycle lasts for 0.4 or 0.5 minutes (every other cycle lasts for 0.4 minutes and the others lasts for 0.5 minutes), see figure 30. The pressure, P1, increases more while in the longest Active Cooling cycle, up to 205 psi. While in the shortest Active Cooling cycle the pressure, P1, reaches a peak of 190 psi instead. The lowest pressure that will be reached after the compressor, P1, is 140 psi, thus fan pair 1 is never switched off, see figure 30. The Bypass cycle time is around 0.8 minutes.

Heater Off The behaviour of the system with a heater off can be seen in figure 31. The EGW supply temperature levels controlling the switching between Active Cooling and Bypass are the same as for the previous tests. With the heater off the EGW supply temperature cycle time is around 4.3 minutes, 3.6 times longer than with the heater on. The Active Cooling cycle time is around 0.3 minutes, which is 1.67 times faster compared to when the heater was on, see figure 31. The refrigerant goes into Active Cooling every 4.3 minutes, same as the cycle time of the EGW supply temperature. The pressure, P1, increases up to 168 psi when in Active Cooling, and when in Bypass it decreases to a minimum level of 121 psi, thus fan pair 1 is never switched off here neither. The Bypass cycle time duration is approximately 4 minutes.

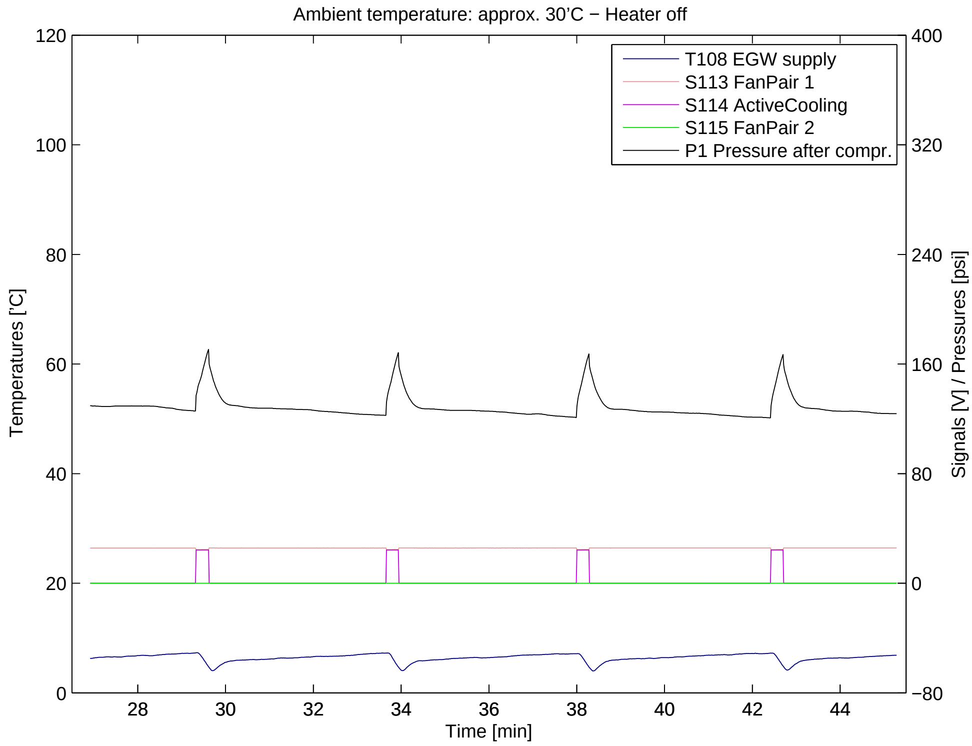

Figure 31: *Graph of the temperatures, pressures and signals measured in the LCU, at 30°C without the heater. The vertical axis on the left side shows the temperature levels and the vertical axis on the right side shows the pressure and signal levels.*

4.2.3 Surrounding air temperature at approximately $40°C$

Both the sauna aggregate and the heater where used during the heating process to reach the desired temperature. Two tests were preformed, one with the heater on (its steady state is shown in the graph in figure 30) and one without the heater on (its steady state is shown in the graph in figure 33). It was necessary to put a particle board above the outlet of the condenser, seen in figure 25. This was done to inhibit the heated air within the styrofoam sauna to be mixed with the lower temperature on the outside, thus reach the desired temperature level and keep it stable. Both fan pairs were on in the beginning of the measurements.

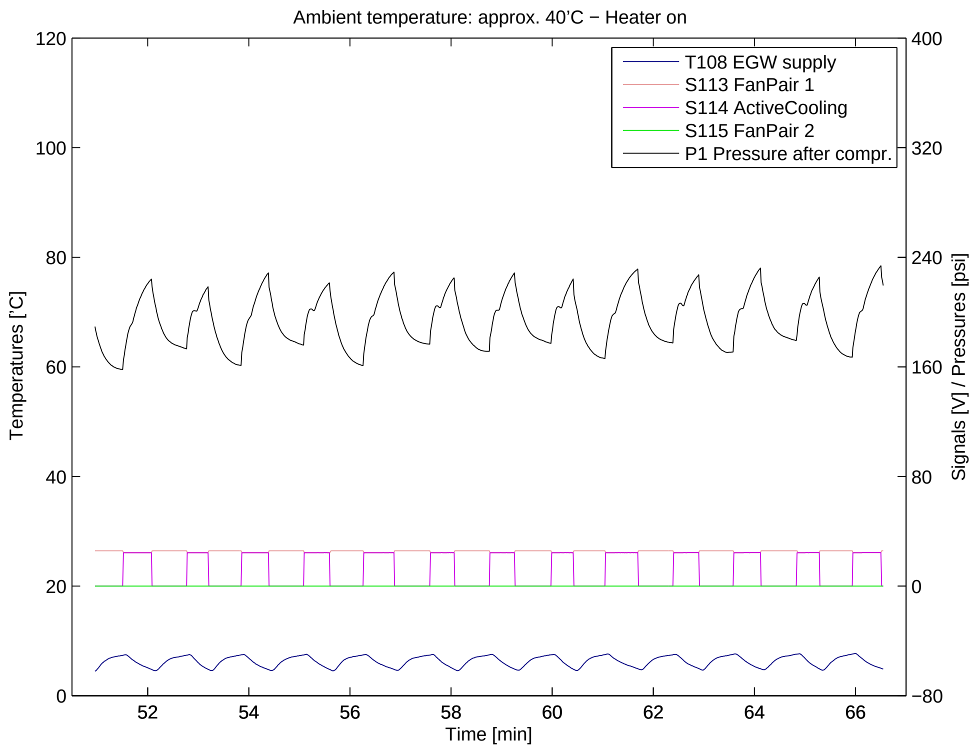

Figure 32: *Graph of the temperatures, pressures and signals measured in the LCU, at $40°C$ with the heater on. The vertical axis on the left side shows the temperature levels and the vertical axis on the right side shows the pressure and signal levels.*

Heater On The behaviour of the system with a heater on can be seen in figure 32. The EGW supply temperature levels controlling the switching between Active Cooling and Bypass are the same as for the previous tests. With the heater on the EGW supply temperature cycle has a time duration of 1.2 minutes. Thus the refrigerant switches to Active Cooling every 1.2

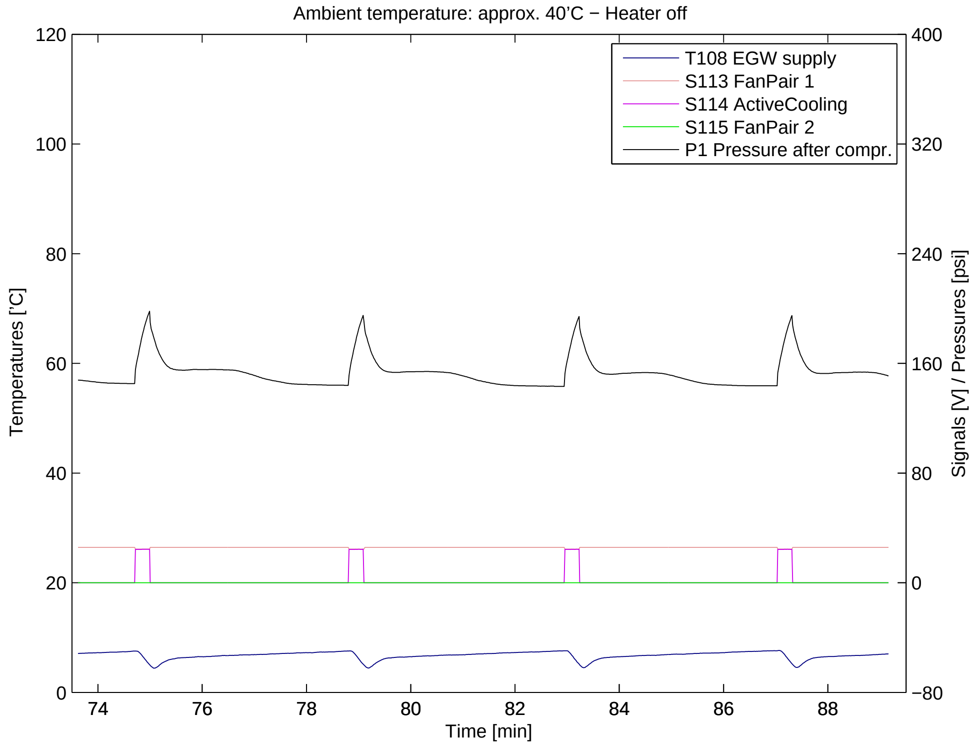

Figure 33: *Graph of the temperatures, pressures and signals measured in the LCU, at* 40°*C without the heater. The vertical axis on the left side shows the temperature levels and the vertical axis on the right side shows the pressure and signal levels.*

minutes and the Active Cooling cycle lasts for 0.6 or 0.7 minutes (every other cycles lasts for 0.6 minutes and the others lasts for 0.7 minutes), see figure 32. The pressure, P1, increases more while in the longest Active Cooling cycle, up to 230 psi. While the refrigerant is in the shortest Active Cooling cycle the pressure, P1, reaches a peak of 225 psi. The lowest pressure P1 will reach is 170 psi, thus fan pair 1 is never switched off, see figure 32. The Bypass cycle time is around 0.6 minutes.

Heater Off The behaviour of the system with a heater on can be seen in figure 33. The EGW supply temperature levels controlling the switching between Active Cooling and Bypass are the same as for the previous tests. Without any load the EGW supply temperature cycle time is around 4 minutes, 3.3 times longer than with the heater on. The Active Cooling cycle time is around 0.3 minutes, which is twice as fast compared to when the heater was on, see figure 33. The refrigerant goes into Active Cooling every 4 minutes. The pressure, P1, increases up to 195 psi when in Active Cooling, and when in Bypass it decreases to a minimum level of 140 psi, thus fan pair 1 is never switched off here neither. The Bypass cycle time duration is approximately 3.7 minutes.

4.2.4 Summary of measurements on present control

As a mean of comparison for the different test cases, table 9 presents the cycle times for the EGW supply temperature, Active Cooling and Bypass for the different test case situations.

Table 9: *Comparison of the different cycle times (c.t.) for the EGW supply temperature, Active Cooling and Bypass for the different experimental test scenarios. All times are in minutes.*

	23 - on	23 - off	30 - on	30 - off	40 - on	40 - off
EGW c.t.	1.0	5.5	1.2	4.3	1.2	4.0
Active C. c.t.	0.4	0.3	0.5	0.3	0.6	0.3
Bypass c.t.	0.6	5.2	0.7	4.0	0.6	3.7

4.3 Noise Level Measurements

The noise level measurements were carried out in two different ways. Firstly, the fans were controlled through the NFO to work at different frequencies. Secondly, only fan pair 1 was on, at their constant frequency of 50 Hz. This is the frequency at which both fan pairs normally work in the LCU.

4.3.1 All fans on at different frequencies

There are three test cycles performed, Test1, Test 2 and Test 3. For each test cycle the fans work in different frequencies, from 0 Hz increasing by 5 Hz up to 50 Hz, thus 11 levels. At each frequency level the noise was measured for 1 minute and the average noise level was the registered result. Since the frequency converter itself make noise, due to its own fan, there were measurements carried out when the condenser fans where off. This was done to have in mind the level of the background noise, which has an influence on all the noise measurements.

The outcome of these measurements can be seen in table 10.

Table 10: *Three noise level tests with all fans on at different frequencies.*

Frequency (Hz)	Noise (dB)		
	Test 1	Test 2	Test 3
0	66.4	64.0	63.5
5	67.0	64.0	63.7
10	66.9	64.0	64.1
15	69.3	68.1	68.0
20	74.4	74.0	74.2
25	80.1	79.8	79.6
30	83.3	83.1	83.1
35	86.2	86.3	86.7
40	89.3	89.5	89.3
45	92.2	92.1	92.7
50	94.8	95.6	95.42

4.3.2 Fan pair 1 on at a constant frequency of 50 Hz

The noise level measurements of all fans working at 50 Hz has already been carried out (the same as in the present control when both fan pairs are on), it was only of interest to have measurements on the noise level when fan pair 1 was working at 50 Hz (normal control conditions of LCU). Three measurement cycles á 1 minutes were carried out. The frequency converter was used as well to have similar background noise for both noise level test setups. The outcome from these test cycles can be seen in table 11.

Table 11: *Three noise level tests with two fans on at 50 Hz.*

Frequency (Hz)	Noise (dB)		
	Test 1	Test 2	Test 3
50	89.7	89.5	89.4

4.3.3 Comparison of noise level tests

The results from both of the noise level test setups are illustrated for comparison in figure 34. From this figure it can be seen that the noise level maintain a constant level up to 10 Hz. The reason for this behaviour could be due to the noise from the frequency converters fan. At 0 Hz the condenser fans are off, the noise level does not seem to increase until they reach 10 Hz, thus the frequency converter has a higher noise level than the condenser fans when they work at less than 10 Hz. Between 10 Hz and 25 Hz the relation between the frequency and the noise level is linear. Its relation continues to be linear from 25 Hz up to 50 Hz, but the slope is much steeper, thus the noise level increases more slowly as the frequency increases. It is interesting to highlight the fact that fan pair 1 have the same noise level as all 4 fans have at 40 Hz.

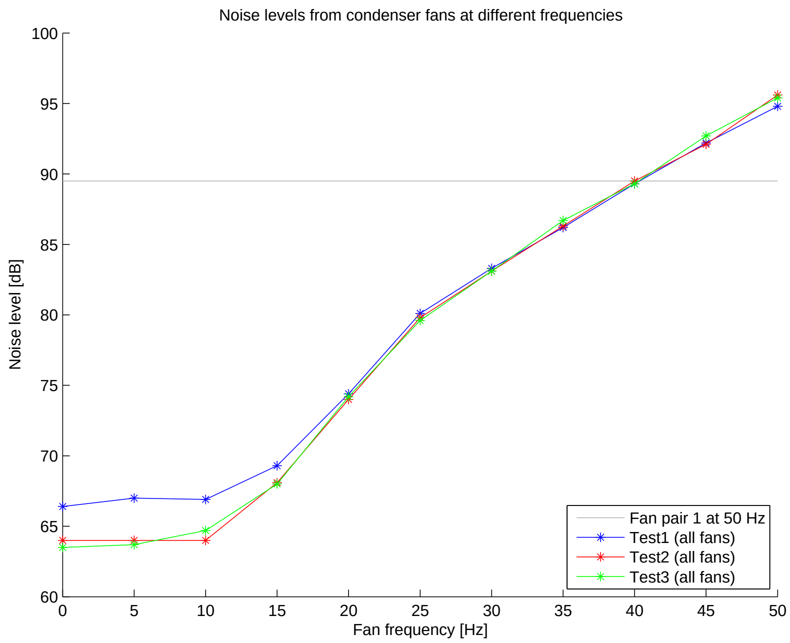

Figure 34: *Graph over the results obtained from the noise level tests. One test setup when all 4 fans are frequency controlled from 0 Hz, increasing by 5 Hz, up to 50 Hz and one test setup with fan pair 1 at 50 Hz.*

4.4 Pressure Setpoint Measurements

The pressure setpoint measurements were carried out by connecting the pressure sensor at the high pressure side, P1, with the frequency converter. Then a pressure setpoint (of 8.6 bar = 125 psi) was set in the frequency converter, as well as PI parameters and other settings described in table 8. The PI control settings were chosen to act slowly to changes in an attempt to reduce risk of the fan speed increasing more than necessary, thus keeping the noise level down.

The display of the frequency converter showed the frequency supplied to the fans and read manually to give an indication of how the frequency varied due to pressure changes. The results from the measurement can be seen in figure 35.

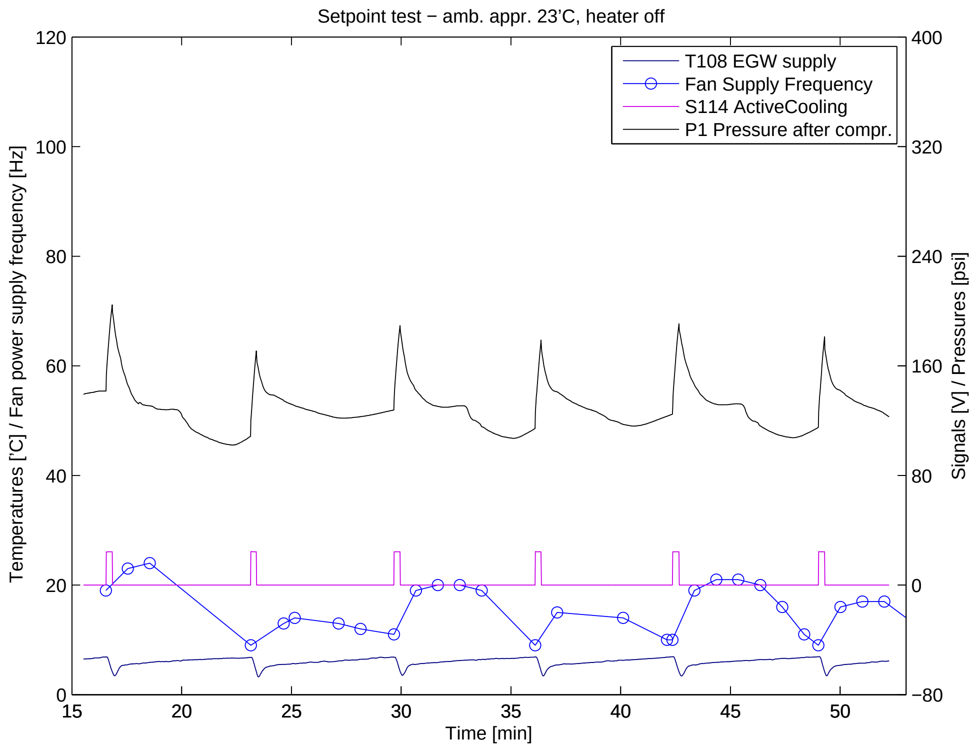

Figure 35: *Graph over the results obtained from the pressure setpoint test at an ambient temperature of 23°C, without the heater. The vertical axis on the left side shows the temperature levels together with the fan power supply frequency and the vertical axis on the right side shows the pressure and signal levels.*

From figure 35 the relation between the pressure after the compressor and the frequency of the condenser fans is illustrated. When the refrigerant is in Active Cooling ($S114 = 27$V) the pressure increases rapidly and the frequency increases as well, but at a much slower rate. As the pressure reaches its peak and decreases, the fan frequency also decreases. The cycle time of Active Cooling is 0.3 minutes, same as in the case with fixed frequency in section 4.2.1.

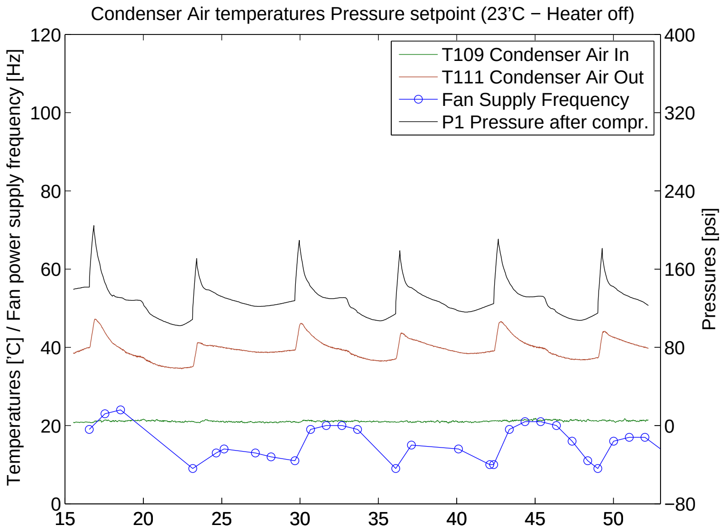

Figure 36: *Graph of the inlet and outlet temperatures of the condenser in the LCU, at $23°C$ without the heater. The vertical axis on the left side shows the temperature levels together with the fan power supply frequency.*

Figure 36 shows the relation between the pressure/temperature of the refrigerant after the compressor and the outlet air temperature of the condenser unit. This relation is the same as the one described in section 4.2.1, between the pressure and the outlet temperature at a surrounding temperature of $23°C$ without a heater. As the pressure of the refrigerant increases as well as its temperature due to Active Cooling the frequency of the fans increases. Thus the outlet air temperature increases due to the heat removal by the fans of the refrigerant. This heat removal causes a decrease in both the pressure and the temperature of the refrigerant, which then causes a decrease in the fan frequency and in the temperature outlet.

An interesting result between these measurements and the ones carried out in section 4.2.1 is the difference between the inlet air temperature for both tests. In the one carried out it section 4.2.1 the inlet temperature fluctuated with the refrigerant temperature, due to lack of airflow when the fans where off. The inlet air temperature from these measurements are constant instead. The explanation for this could be that due to the frequency control of the fans, as they are never switched off. This causes a continuous airflow through the condenser unit, thus the air is never still enough to get warmed up.

4.5 Discussion

The experimental tests carried out on the system does not represent all the different working situation that the LCU normally endure. The different situations simulated by the use of the heater and the styrofoam sauna are considered to be enough for this project as a ground for evaluation of the LCU system.

When temperatures have been measured, the thermocouples have been placed on tubes of the EGW line and the refrigerant line, with an adhesive tape. It is obvious that the exact temperature is hard to measure this way, but it was not possible to enter the tubes to measure the two substances. The measuring accuracy is considered enough, since the exact temperature is not as important as the relation between the temperatures, pressures and fan activity.

The isolated area, caused by the styrofoam walls, is rather small and the walls inhibit a natural air flow through the condenser unit. The styrofoam wall 1, see figure 17, was very close to the air inlet of the condenser unit, thus inhibiting a normal air inflow to the unit. The effects of this inhibition was clear in the comparison between the measurements of the inlet air temperature of the condenser carried out in section 4.2.1 and the ones carried out in section 4.4. When the fans are off, the inlet temperature increases slowly but when the fans were frequency controlled and on all the time the inlet air temperature was constant. This was not considered to be important since the relation between the refrigerant pressure and the fan work was of higher interest.

The outlet of the condenser unit was blocked by wall 4, see figure 17, thus the wall had to be moved to enable the outlet air to leave and also to avoid it to return directly to the air inlet. A special arrangement has been done with particle boards and a plastic bag to screen off the inlet and the outlet airflow of the condenser. This arrangement does not resemble the real airflow behaviour but it was the best that could be done with the available equipment.

In the case of the noise measurements, the noise from the fans of the frequency converter was dominant during the measurements with the frequency controlled condenser fans. Not until the condenser fans reached a frequency of 10 Hz they had an impact on the noise level. Therefore from the noise level results, both with the frequency controlled tests and those carried out at normal working frequency, the offset caused by the noise from the frequency converter has to be kept in mind when comparing to other test results than the ones carried out here. When comparing the test carried out for this project, the offset caused by the frequency converter is irrelevant, since both measurements have the same offset.

The pressure setpoint measurement was only carried out at an ambient temperature of $23°C$ without a heater. This was because there were not enough power supplies to have the frequency converter connected at the same time as the heater and since there was no free space left to have the frequency converter inside the styrofoam sauna wall 4 could not be used to heat up the inside air. These results are therefore only applicable for a special case and can only be compared to the measurements carried out with the same ambient temperature, no heater and with fan pair 1 at 50 Hz.

The frequency activity of the fans was read manually since the information was only available through the display of the frequency. Thus the data about the fans frequency is less continuous and the number of registered measurement are significantly less compared to the pressure and temperature measurements. There was no easier way to register the frequencies than manually and this strategy was enough to see the fan behaviour when frequency controlled and to see the relationship to pressure and outlet air temperature.

5 Analysis of Present Control

The LCU is dimensioned to function in a temperature range of −40 °C up to 55 °C, and with a load ranging between 0 and 11kW. This fact resulted in a refrigeration system that is oversized for many of the situations. The compressor is dimensioned to handle the worst case scenario, due to this it is oversized for the low load/low temperature cases.

To deal with the vast variations of cooling demand and to avoid frequent start-ups and shut-downs of the compressor, the bypass function was included. It is a method that permits the system to adjust rapidly to new load requirements, but from an energy efficiency point of view it is wasteful.

Even though the limits that regulate the interchanging between Active cooling and Bypass are clearly stated in the documentation, the limits on a real specimen of the LCU differ significantly from those stated in documentation.

5.1 Primary and Secondary Control

The main control to keep the EGW temperature at an acceptable and relatively constant level is the interchanging of Active cooling (normal vapour compression refrigeration cycle) and Bypass. This control is entirely based on the temperature of the supply EGW.

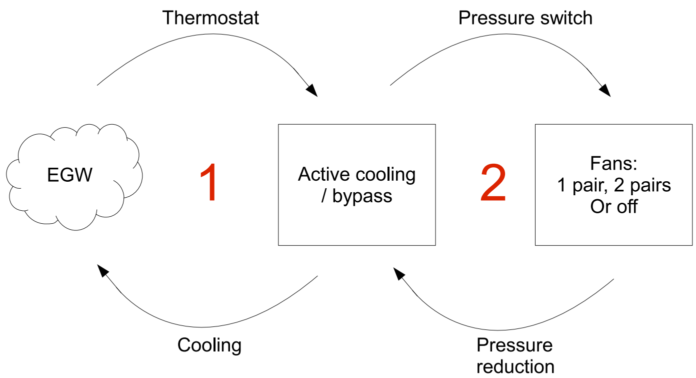

Figure 37: *The division of the interesting control cycles in the LCU and their correlation. The primary control is marked as 1 in the figure, and the secondary control is marked as 2.*

As a secondary control, the condenser fans are regulated based on the refrigerant pressure as it leaves the compressor. This control is entirely based on the refrigerant pressure, and not on the EGW temperature, as shown in figure 37. Thus, the fans can be on as the system is in bypass mode. Since unneeded work is added to the system, the pressure increases and the fans are activated. This is unnecessary fan work if one bear in mind that the original intention of the refrigeration system is to cool the EGW. An illustration of this division is shown in figure 37.

As can be seen in the results presented in section 4.2, the system cools the EGW rapidly each time it is in Active Cooling mode. To compensate for the rapid cooling, the system needs to alternate between Active Cooling and Bypass quite frequently. It is not hard to see that a more continuous capacity control (less of an on/off switching) could lead to a smoother EGW temperature.

5.2 Statistics

The tests that were performed do not fully describe the range of situations that the LCU need to handle. As previously mentioned, it is supposed to handle a load of 11 kW even at a temperature of 55 °C. In the experimental tests described in section 4.2 the maximum temperature achieved was 40 °C, and the maximum load was 4 kW. It is important to bear in mind that the results presented therein show the behaviour of the LCU at different degrees of part load. An illustration of what region that lie as a base for this analysis is shown in figure 38.

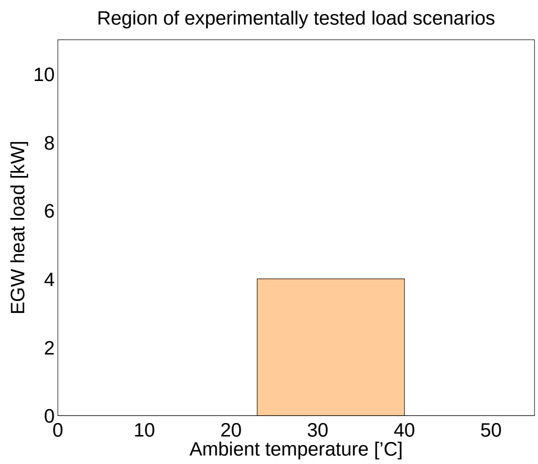

Figure 38: *The coloured region represents the part of the LCU load cases that has been experimentally tested compared to the whole region of load cases that the LCU should handle.*

Table 12: *Comparison of the different mean pressure levels, and the percentage of Active cooling for the different experimental test scenarios.*

	23 - on	23 - off	30 - on	30 - off	40 - on	40 - off
mean P1 [psi]	127	123	170	128	194	153
Active C. [%]	33	5.1	42	6.2	44	7.3

As a mean of comparison for how much of the cooling capacity that is actually used for the different scenarios that were experimentally tested, the percentage of time that the system was in its Active Cooling state was calculated. This comparison is shown in table 12 together with the mean pressure level after the compressor for each of the scenarios.

This percentage can be used when discussing the actual refrigeration output compared to the work input, see section 2.1. The theoretical Carnot COP can be calculated, according to equation 3, for the part of the tests where actual cooling has occurred. The time percentage when the system has been in Bypass no actual cooling is achieved, and thus the $COP = 0$. A comparison is presented in table 13. The calculations are based on an EGW temperature of 5 °C, thus $T_L = 5 + 273\,K$.

Table 13: *Comparison of the different Carnot COPs for the different experimental test scenarios. Also lists the time percentage of Active Cooling times the Carnot COP.*

	23 - on	23 - off	30 - on	30 - off	40 - on	40 - off
T_H [K]	296	296	303	303	313	313
Carnot COP	15.58	15.58	11.19	11.19	7.98	7.98
Active C. * Carnot COP	5.14	0.79	4.70	0.69	3.51	0.58

As pointed out in section 2.1, the Carnot COP is a theoretical COP, valid only for the case of totally reversible processes. This means that it points out the maximum theoretical COP for those two temperatures, T_H and T_L.

The calculations relating Carnot COP and time percentage of Active Cooling has been included as a reference to illustrate how much energy that is actually wasted in normal part load cases for the LCU. Of course a standard refrigeration cycle operating between these two temperature levels would not yield such a high COP, but the “loss of efficiency” due to the Bypass operation mode is clear.

To this should also be added the fact that the fans are constantly held as a mean of controlling the compressor pressure at a suitable level, even though the momentary COP is null, and the refrigerant cycle is not used for cooling.

5.3 About pressure levels

For the present control system, the mean pressure level is different for the tested load scenarios, as seen in table 12. The different pressure levels are not as strongly dependent on the ambient temperature differences as on the different heat loads for the tested scenarios. This could be since a higher load requires more refrigerant flow through the evaporator and thus a higher refrigerant flow in the entire refrigerant cycle. Since the flow is induced by the pressure increase in the compressor, the pressure level after the compressor also increases to supply a higher refrigerant flow.

6 Modelling and Validation

Modelling in MATLAB/Simulink has been done for the Ambient mode case. In this model, the EGW is cooled directly by the surrounding air temperature and by the fans in the condenser heat exchanger. A heating has been included in the model to simulate heat added to the EGW while not inside the LCU. For more information about heat transfer see [3].

When the system is in Ambient mode, the EGW mixture is passed through the Ambient heat exchanger to be directly cooled by the ambient air, as described in section 3.2. The fans help in the process by forcing the air through the condenser heat exchanger at a higher speed. Test data from the old control of the system have been used to approximate how much the fans affect the cooling of the EGW.

6.1 Modelling

The heat transfer in the condenser unit, from EGW to ambient air, was assumed to depend linearly on the difference between ambient temperature and EGW temperature, see equation 6. With this assumption made, the coefficients for the cases with and without active fans were derived from test data of the present system. The test data included air inlet temperature, air outlet temperature, air flow rate, EGW supply temperature and EGW return temperature. One coefficient, k_{nofans} was derived regarding the heat transfer while the fans are inactive and one for the case when the first pair is operating, k_{fans}. No test data were found where all four of the fans were active in Ambient mode.

$$\dot{Q} = k\Delta T = k(T_{egw,return} - T_{ambient}) \tag{6}$$

The coefficients were derived based on the assumption that the rate of heat transfer in the Ambient heat exchanger equals the rate of heat loss for the EGW. Since the properties for the EGW are known, equation 7 could be used [3].

$$\dot{Q}_{egw} = C_{p,egw} q_{egw} (T_{egw,return} - T_{egw,supply}) \tag{7}$$

$$k = C_{p,egw} q_{egw} \frac{T_{egw,return} - T_{egw,supply}}{T_{ambient} - T_{egw,return}} \tag{8}$$

By assuming that equation 6 and 7 are equal, the coefficient calculation is shown in equation 8. The EGW flow rate, q_{egw}, was taken from earlier test data. The specific heat capacity value, $C_{p,egw}$, was included in the Simulink model as a look-up table. The data used for this table originated from [16], and Simulink interpolates for values inbetween the specified data. The look-up table has two inputs and one output; the values of the EGW return and the percentage of ethylene glycol in the mixture are the inputs and the output is the specific heat capacity for the EGW.

Since there were no data on how the fans cooled when working at part speed, the fans effect on the heat transfer was assumed to be linear between the on and off state. When there are no fans the fan effect is 0 and when there is 1 fan pair the fan effect is 0,5. Fan effect 1 was saved for the case when all four fans were operating, but as explained previously in this section, no such case was found in the available test data.

The simulated heat load also used the relationship given in equation 7 to calculate the EGW return temperature as a function of the given EGW supply temperature, specific heat capacity, EGW flow rate and a specified heat load.

6.2 Results

The results from the simulation models of both the present control and a new one are here presented. A comparison between the simulated behaviour of the present control and real test data is also included. The complete Simulink model can be found in Appendix B.

6.2.1 Results from the simulated model with the present control

In figure 39 the results from a simulation of variable heat load is presented. At time 160 minutes the heating of the EGW is increased with an increase in fan activity as a result. The cooling cycles are clearly occurring more frequently and the cycle time for the fans is longer. The increased heat load has a duration of 20 minutes.

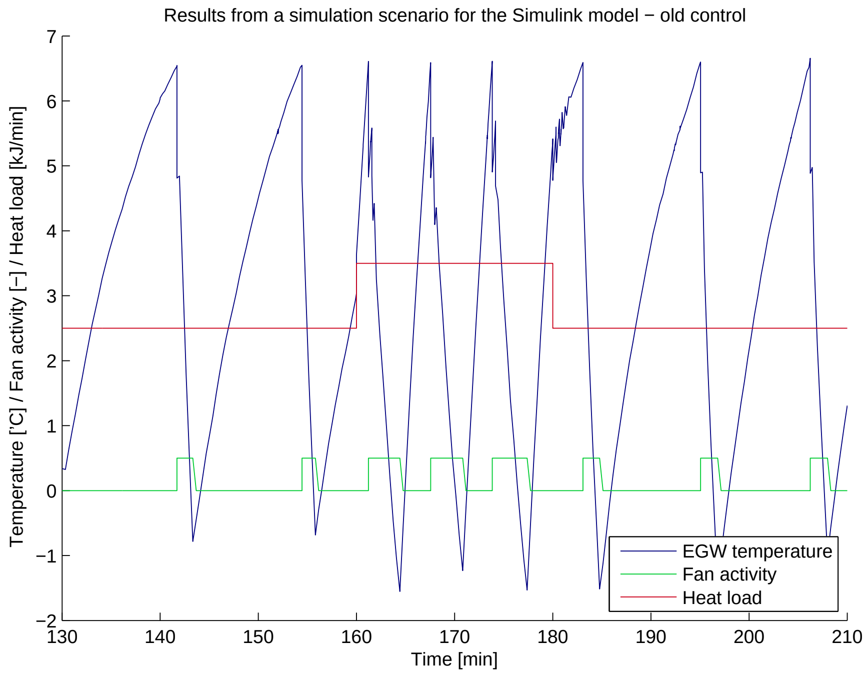

Figure 39: *Graph of the measured EGW supply temperature, heat load and fan activity from the simulation model.*

6.2.2 Model validation

In figure 40 a comparison between the test data of the system and the simulated behaviour can be seen. The simulated response can be adjusted to correlate to the measured one by adjusting the simulated heat load. The real heat load during the validation data test was not known.

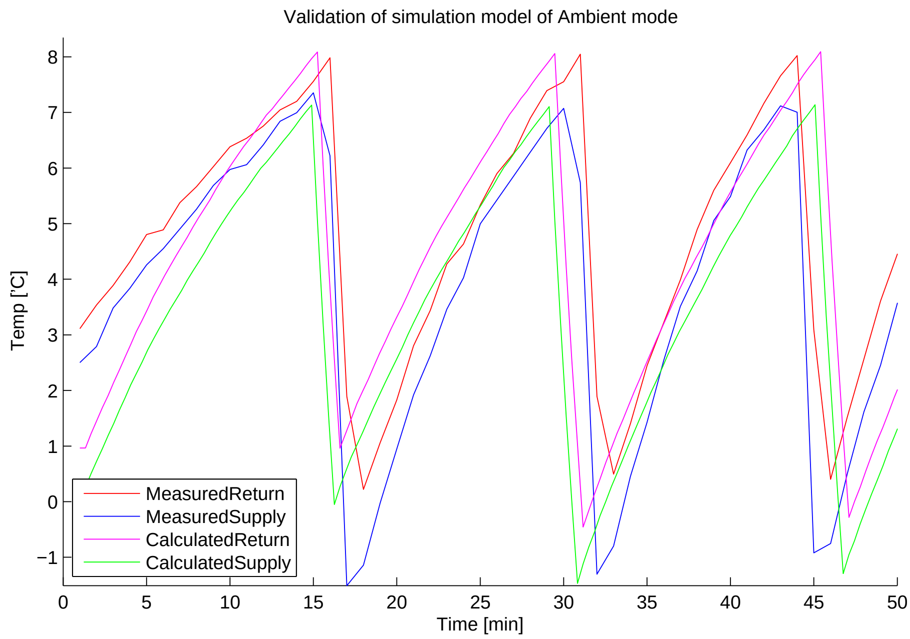

Figure 40: *Graph of the measured EGW return and supply temperatures and the calculated EGW return and supply temperatures from the simulation model.*

6.2.3 Results from the simulated model with a new control

A new control was tested on the model to see if the performance of the system could be improved. A PI control was chosen to be sufficient since the dynamics did not require a control apt to fast changes. The PI parameters used for the result presented in figure 41 are $P = 0.2$ and $I = 0.1$.

The same heat load case was chosen as in the simulations of the present control system to allow for easier comparison. As can be seen the EGW supply temperature follows the setpoint of 6 °C quite well. The fan activity is held low at all times, although a high increase is seen for the time period of increased heat load. It is still not as high as for the old control.

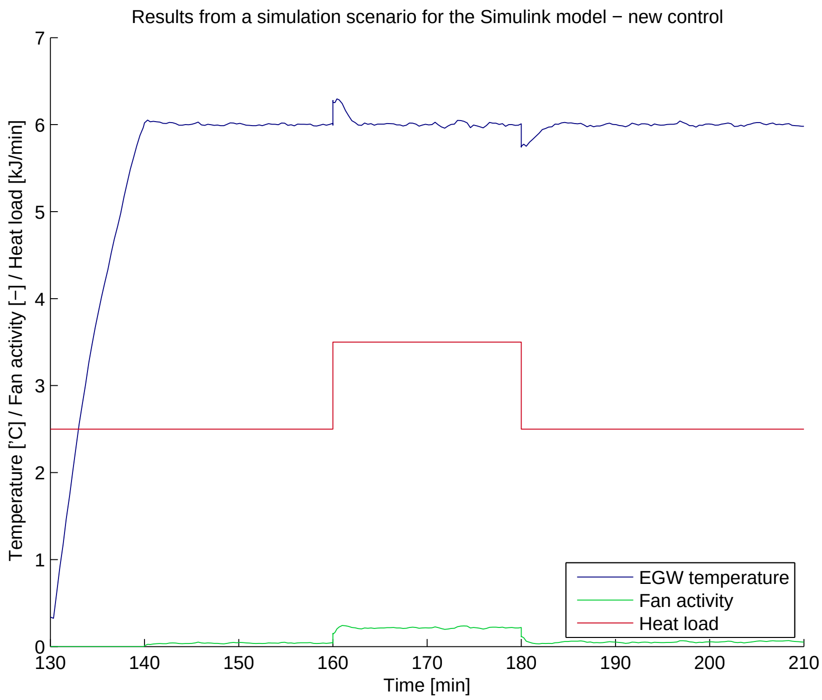

Figure 41: *Graph of the measured EGW supply temperature, heat load and fan activity from the simulation with a new control.*

6.3 Discussion

As with all models, the question must be raised of whether or not the assumptions were reasonable to make. As for the assumption of two heat transfer coefficients, one for fan pair 1 being on and the other for all fans being off, it seems valid based on the validation data available. It is still unknown whether or not the model behaves well at other ambient temperatures.

Another aspect that needs to be considered is the fact that the new control is based on a linear relationship regarding the heat transfer coefficients. This is by no means validated since no data for heat transfer between these two points could be obtained.

It is also important to note that the "frequency control" of the fans for this simulation is based on fan effect from *one* fan pair and not all four as was the case in section 4.4.

7 Concluding Remarks

In this section the results obtained through analysis of the present control system are discussed. Possible changes that can be carried out and suggestions for future work are also presented.

7.1 Discussion

There are a few key points that have been in focus during the evaluation of this LCU, and these are discussed in this section.

7.1.1 Frequency control of fans

Considering the results from the noise level tests (section 4.3) and having in mind the results from the pressure setpoint measurements (section 4.4) it can be seen that the noise level of the fans can be decreased by implementing frequency control.

During the pressure setpoint tests the highest frequency reached by the fans was 24 Hz and the lowest frequency achieved was 10 Hz. Thus the highest noise level was around 80 dB and the lowest was around 64 dB (a difference of 16 dB). This can be compared to the noise level when fan pair 1 work at 50 Hz, which is around 89 dB. In the case with a constant fan frequency the fans were switched on and off, this causes a noise level difference of about 24 dB. It is of importance to have in mind that this comparison is only valid in the case of no heat load and a surrounding temperature of $23\,°C$. Though it is not unreasonable that the frequency control of the fans should have some kind of noise reduction in other temperatures and heat load situations.

7.1.2 Reduce unnecessary fan work

With the objective to keep the noise level down, the division between primary and secondary control addressed in section 5.1 can be questioned. As pointed out, this division leads to the fans having to work to keep the pressure down even when no actual cooling is going on.

Playing with the idea of disregarding that intermediate step, the idea of letting the EGW supply temperature control the fan work could come up, still keeping the primary control as it is today. The fan control would then be similar to what the setting is in Ambient mode, but probably with a different setpoint. The down side with this approach is that the refrigeration system is not in Ambient mode, and the refrigerant cycle needs to be controlled as well. It is clear from the results presented in section 4.2 that the pressure increases rapidly each time the system switches from Bypass into Active Cooling. But this is also based on the same EGW temperature, so one could consider this as an option. This would however require a more complex control than a standard PI or PID control, since the signal that the EGW temperature has succeeded the approved level would require the fans to take action fast even though the EGW temperature starts to decrease. Should the fans not increase in speed as the mode changes from Bypass to Active Cooling, the risk of a too high pressure is high, at least for higher ambient temperatures.

7.1.3 Altering existing capacity control

To reduce the need of fan work, one could instead address the problem that originates extra fan work, i.e. the wasteful compressor work. As pointed out in section 2.2.1, there are several ways of controlling the capacity of a refrigeration system by reducing compressor work. One could consider replacing the existing compressor with two or three smaller ones. This could lead to a less wasteful capacity control, but would probably also require a bit more space than the present compressor needs, which could be a problem.

A more realistic idea could be to replace the existing compressor with a new one that is constructed for frequency control. This would have the ability to reduce the refrigeration capacity significantly for cases of lesser load. Even though it cannot be adjusted down to the level of no refrigeration, it could improve the performance of today's system. Together with the existing bypass function, the compressor could run at it's lowest speed for very low load scenarios, still interchanging between Active cooling and Bypass but in a less energy wasteful manner.

The idea of combining one variable speed compressor with the bypass function would also have the benefit of not needing to start and stop the compressor frequently. This has been of interest for the company when designing the present control system. Combining the two methods would lead to a similar, yet less wasteful control.

Even though energy saving was not the main focus while evaluating the control system, it should surely be regarded beneficial, regardless of the associated fan activity. A mobile system consuming a less energy would last a bit longer before needing to refill fuel.

7.1.4 Ambient model

The model of Ambient mode show that a frequency control leads to a less fluctuating EGW temperature, even when the heat load changes temporarily. Furthermore, the fan activity is held at a more constant and lower level compared to with the present control. Even though the model is quite simple in its structure, the experimental results of Active mode shows the same kind of improvement regarding fan activity. Thus, it is reasonable to assume that frequency control for Ambient mode also would result in a lower noise level than with the present control strategy.

7.1.5 BEA multiplex system to implement new control systems

The use of the BEA system for the implementation of a new control system in the LCU would eliminate the need of the PCB, since the BEA system can manage to control all signals to the different motors, the fault indicators, timers etc. The inflexibility of the PCB would be replaced by the flexibility of the BEA system due to the reprogrammable hardware modules, webserver, "Plug and Play" function and fault diagnostics.

The reprogrammable hardware modules with the webserver feature enables developers to work on the same project. The "Plug and Play" feature simplify the service of modules if they are out of order. The MasterID handles the update of the software to the new module, thus no extra equipment is needed for service. The fault diagnostics simplify the error search, and together with the webserver the fault search can be done without being close to the system. The enabling of flexibility by the BEA system could make it possible to implement different control solutions for evaluation and also increase the possibility for customization.

The sensors used in the present LCU system should be replaced in order to frequency control the compressor and the condenser fans. The BEA modules need voltage as input, thus the pressure sensors purchased for the experimental testing can be used if resistors are used when connecting them to the modules.

7.2 Conclusions

There are many conclusions about the control of the LCU that can be drawn based on the results of this thesis. For one, the experimental noise level tests show that the noise level from fan pair 1 at 50 Hz is equivalent to all fans working at 40 Hz. This implies that the noise level is decreased by having all four fans on at lower frequencies compared to two fans at full speed. From the experimental noise tests and pressure setpoint tests carried out on the system it is clear that for the no heat load case at a surrounding temperature of $23°C$ frequency control of the fans lowers the noise level significantly. When implementing frequency control of the fans they work at a lower frequency, thus lowering the noise level. The Simulink simulation indicates that frequency control of the fans probably would reduce the noise level in Ambient mode as well.

Another conclusion that can be drawn from these results is that the compressor is bypassed a large amount of the time. This is due to the fact that the system is dimensioned to handle the worst case scenario of load and surrounding temperatures. This bypass method wastes energy which is undesirable in a mobile vehicle. It also leads to more than necessary fan work.

As a mean for improvement, a frequency control of the compressor could be introduced. However, this cannot be done with the present control. If frequency control i desired to implement, the present control needs to be replaced by one that is designed for frequency control.

Regarding the implementation of a new control for the LCU, the present technique of relay logic is not the only solution. The BEA multiplex system has the necessary features as communication and control development tools to be implemented with the LCU. It also has the advantage that it is flexible for changes, and a control system can easily be modified even after implementation.

7.3 Future work

To continue the work of improving the LCU control performance, the following issues are recommended by the authors:

- Implement frequency control of the all four fans to reduce the noise level caused by the on/off behaviour.
- Complement the bypass loop with frequency control of the compressor. For this the present compressor in the LCU needs to be replaced.
- Implement the BEA multiplex system to control the LCU in a more flexible way.

- Complement with additional tests for different temperatures and load scenarios than the ones tested in this thesis work.

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A Additional Background Theory

A.1 Sensors

This section describes what sensors are, what they are used for and how different types of sensors work. Focus will be on the different temperature and pressure sensors that exist. A general introduction to sensors can be found in [17] and [18], and these sources have also been used as a basis for this section.

When systems are being monitored and/or controlled, physical parameters of the system are to be measured. The sensors register a change in the measured parameter, converts its value and sends a signal for the system to process. The parameters that usually are measured in control systems are force, pressure, flow rate, temperature, position and humidity.

There are three steps in all sensor operations:

1. The change detected in the measured physical parameter is converted to an equivalent property change in the sensor.
2. The property in the sensor is transformed into an electrical signal, either in the form of current or voltage.
3. The signal from the sensor is amplified, conditioned and transmitted to a device for processing.

In step 1 the property change in the sensor is different depending on the measured parameter, and there are also several measurement techniques for each parameter. The sensor can be designed to change its induced current, induced voltage, resistance, capacitance or inductance. The components of a sensor can be seen in figure 42.

A.1.1 Temperature sensors

The measuring and monitoring of temperature is very important in many technical systems. Temperature sensors are used in engines, computers, air-conditioning systems, chemical process plants, buildings and vehicles. There are three different classes of temperature sensors, they can change physical dimension, they can change resistance due to temperature changes or they can work based on thermoelectric phenomena.

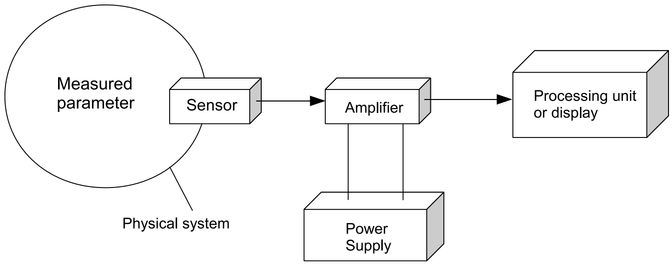

Figure 42: *The sensor components, sensor head, amplifier, power supply and processing unit or display.*

Sensors based on dimensional change The majority of liquids and metals change their dimension due to temperature changes. Both mercury and alcohol are used in *liquid-in-gas thermometers*, since they expand and contract proportionally when the temperature changes. The accuracy of this sensor is $\pm 0.5°C$. There are some limitations when using this kind of temperature sensor since readings are made visually, the accuracy is quite poor and the possibility of a continuous and automatic registration of temperature changes is impossible.

Another sensor device which changes dimension as a function of temperature is a *bimetallic strip*. A bimetallic strip is assembled of two (or more) metal layers, which have different thermal expansion coefficients. Due to this difference, their structure will change when the temperature changes, and the deflection is used to relate to the temperature of the strip.

Sensors based on resistance There are two different types of temperature sensors which are based on changes of resistance, *Resistance Temperature Detectors (RTD)* and *thermistors*.

The *RTD* act on the transduction principle, that the resistance on its material changes with the temperature, i.e. the resistance increases with the temperature. The change of resistance is translated, using a Wheatstone bridge, to a proportional voltage. The signal flow is

$$T \Rightarrow R \Rightarrow V$$

Its linear relation to temperature and its accuracy, $\pm 0.005°C$, are two of the main advantages for this sensor. Another advantage is that its drift over time is very small, less than $0.1°C$/year, thus it do not need frequent calibration. Some disadvantages about the RTD is that its dynamical response is slow compared to other temperature sensors, and also that it cannot be used for measuring high-frequency transient temperature variations.

Thermistors are based on semiconductor materials. Its resistance decreases exponentially with the temperature. The signal flow is

$$T \Rightarrow R \Rightarrow V$$

Its accuracy is $0.01°C$ or higher, which is better in comparison to an RTD sensor. A disadvantage of the thermistor is that its operating range is much smaller than that of an RTD.

Based on thermoelectric phenomena *Thermocouples* work based on the Seebeck effect. When two distinct electrical metal conductors are in contact, they form a thermoelectric connection that produces a voltage. This voltage (V) is proportional to the temperature difference (ΔT) of the connection and this is the Seebeck effect, this signal flow is

$$\Delta T \Rightarrow V$$

The connection between the two metals is shown in figure 43.

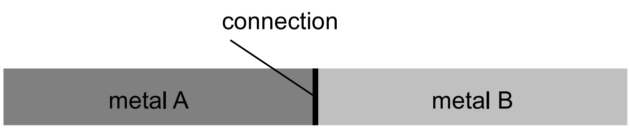

Figure 43: *The thermoelectric connection between two metals.*

The circuit of the thermocouple can be seen in figure 44.

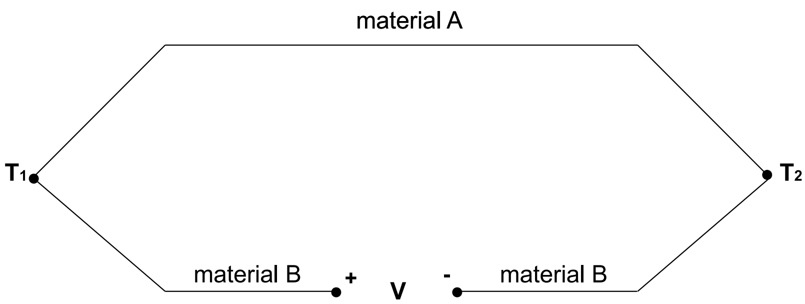

Figure 44: *The circuit of the thermocouple.*

Thermocouples are commonly used since they are user-friendly and cheap. One disadvantage of the thermocouples is that their accuracy is around $\pm 1 - 2°C$.

A.1.2 Pressure sensors

Pressure sensors are used in many technical systems, such as traffic cameras, combustion engines, chemical process plants etc. These sensors measure the pressure of a liquid or a gas. Pressure can be either absolute pressure or local atmospheric pressure. Absolute pressure is the pressure measured relative to a total vacuum or zero pressure. The local atmospheric pressure is the pressure against a surface caused by the weight of the above air. This pressure varies at different locations due to the height difference from sea level.

There are 4 different types of measuring pressure techniques: based on displacement, based on piezoresistive strain-gauge, based on piezoelectric effect, or based on capacitance.

Sensors based on displacement The principle of this sensor is to first convert the pressure change, $\Delta P = p_1 - p_2$, into a displacement, Δx, which is proportional and then convert the displacement to a proportional voltage, V_{out}. Combining these two relations the signal flow is

$$\Delta P \Rightarrow \Delta x \Rightarrow V_{out}$$

Sensors based on Piezoresistive Strain-Gauge These sensors uses the piezoresistive effect, the change in the resistance of the material due to mechanical stress, of strain gauges (a device which measure the strain of an object) is used to determine strain caused by an applied pressure.

The principle of the relation between displacement and pressure work, is shown figure 45. The strain (Δx) on the diaphragm is proportional to the applied pressure ($\Delta P = p_1 - p_2$). The resistance (R) of the strain gauge changes in proportion to its strain (Δx). A Wheatstone bridge circuit can be used to get a proportional output voltage (V_{out}) from the strain gauge. The signal flow for this sensor is

$$\Delta P \Rightarrow \Delta x \Rightarrow R \Rightarrow V_{out}$$

Sensors which are Piezoelectric-Based This is the most versatile pressure sensor type. The pressure of the diaphragm is transformed into a force

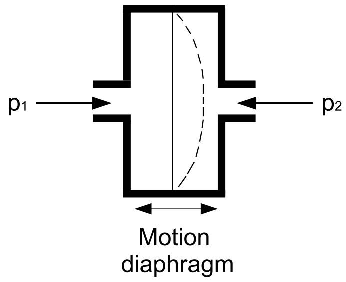

Figure 45: *Illustration of the principle of how strain-gauge pressure sensors work.*

(F) which acts on the piezoelectric element. This can be seen in figure 46. A proportional voltage (V_{out}) to this force will be generated by the piezoelectric element, which then is proportional to the pressure (P), the signal flow is

$$P \Rightarrow F \Rightarrow V_{out}$$

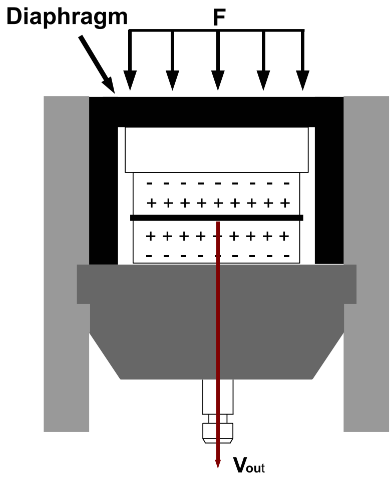

Figure 46: *Illustration of a piezoelectric pressure sensor and its usage of force.*

Sensors based on Capacitance The principle of sensing the diaphragm pressure can also be used to change the capacitance between two charged plates located in a sensor. The change in the diaphragm (x), due to the pressure (ΔP), is proportional to the change in capacitance (C). To get a voltage output (V_{out}) from the capacitance change an operational amplifier, a reference voltage and a reference capacitor can be used. The signal flow is

$$\Delta P \Rightarrow x \Rightarrow C \Rightarrow V_{out}$$

A.2 Frequency Inverters

This section is based on information from [19] and [20], and the interested reader is referred to these sources for more details.

Speed control of induction machines rely on a control of supply frequency, and therefore the common way is to use a frequency inverter to supply the motor. This usually contains a AC to DC rectifier, DC to AC inverter, a tachometer to measure the position of the motor and some control circuits to control the process [19]. Although all standard induction machines can be operated by use of frequency inverters, not all are designed for low speed usage. This can lead to problems with overheating in case of long term usage in low speed, since the cooling of the motors can be dependent on speed.

NFO The NFO (Natural Field Orientation) is a special type of frequency inverter which uses a patented method to alter frequencies. The method leads to a smooth sinus wave output, and thus reduces the electromagnetic interference associated with fast changes due to for instance fast choppers [20].

B Simulink Block Diagrams

This Appendix includes all the simulink block diagrams used for the model of Ambient mode. In figure 47 the model structure is presented. The subsystems presented in that figure, *Condenser heat exchanger*, *Heating EGW* and *Controller*, are presented in figures 48, 50 and 53, respectively. These subsystems have subsystems of their own and these are presented in the figures 49, 51, 52, 54 and 55.

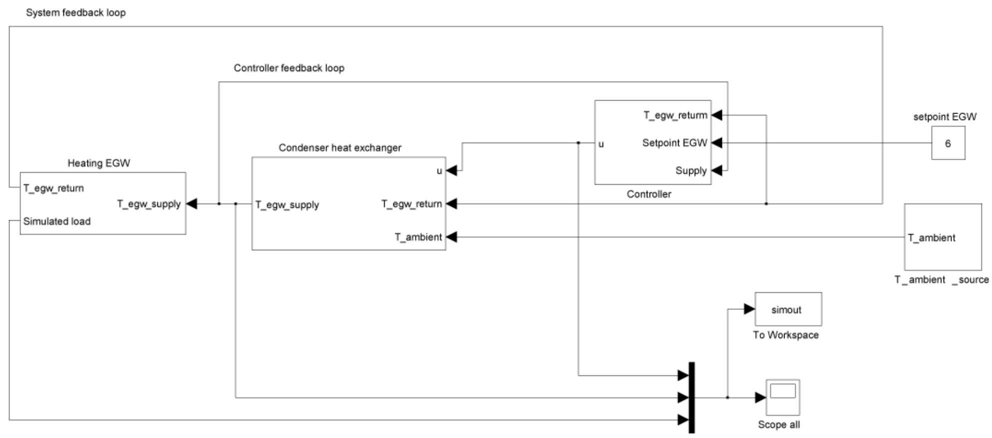

Figure 47: *Simulink model of Ambient mode. The subsystems are shown in figures 48, 50 and 53.*

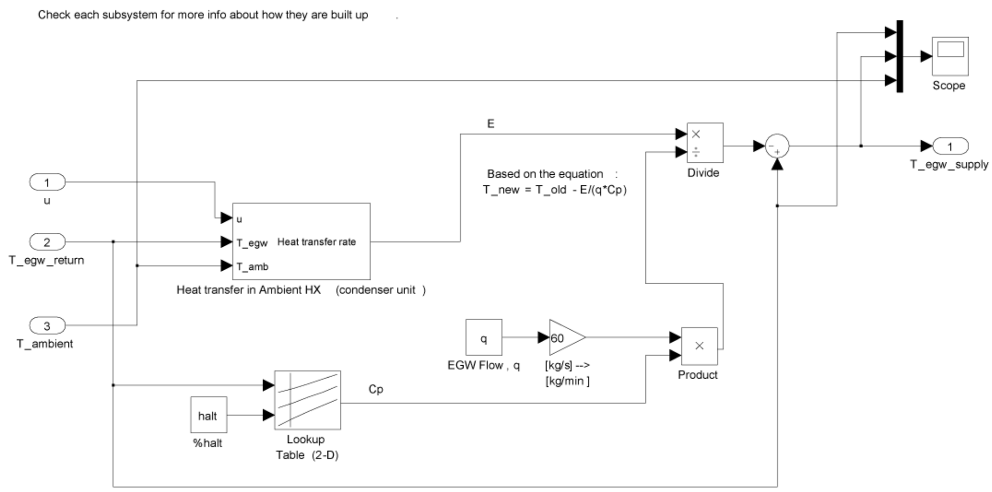

Figure 48: *Simulink model of Ambient mode; subsystem "Condenser heat exchanger".*

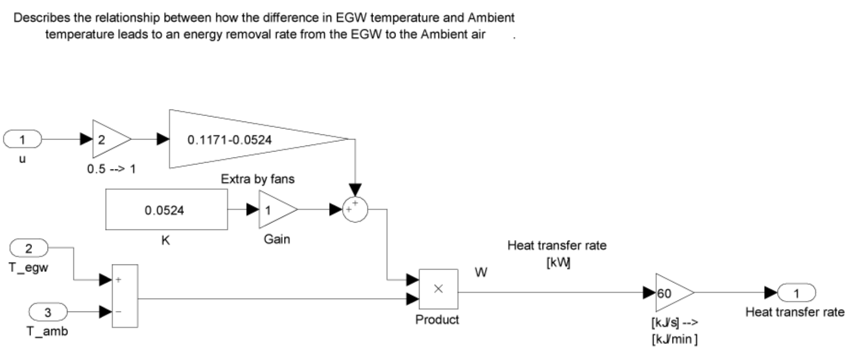

Figure 49: *Simulink model of Ambient mode; subsystem "Heat transfer in Ambient HX (condenser unit)".*

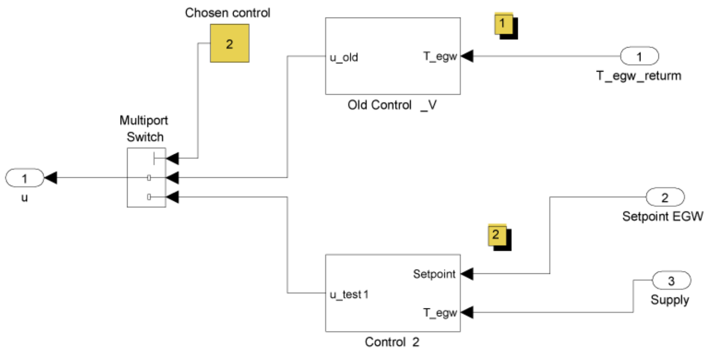

Figure 50: *Simulink model of Ambient mode; subsystem "Controller".*

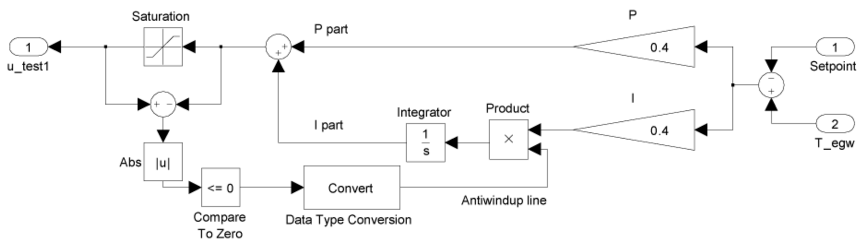

Figure 51: *Simulink model of Ambient mode; subsystem "Control 2".*

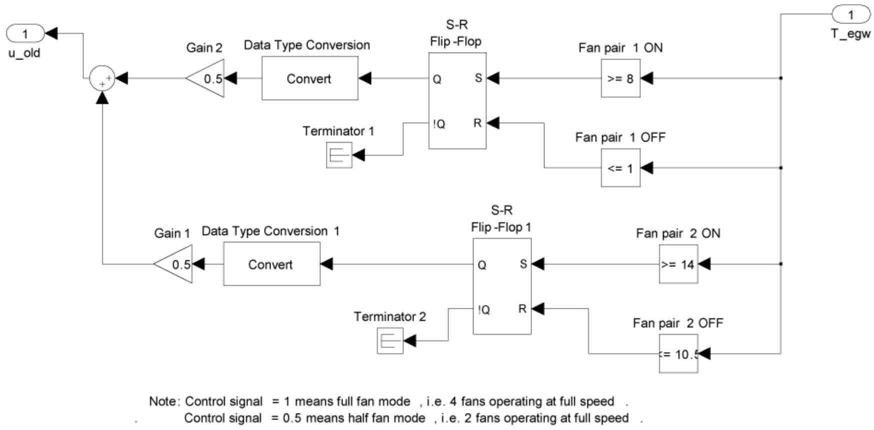

Figure 52: *Simulink model of Ambient mode; subsystem "Old Control _V".*

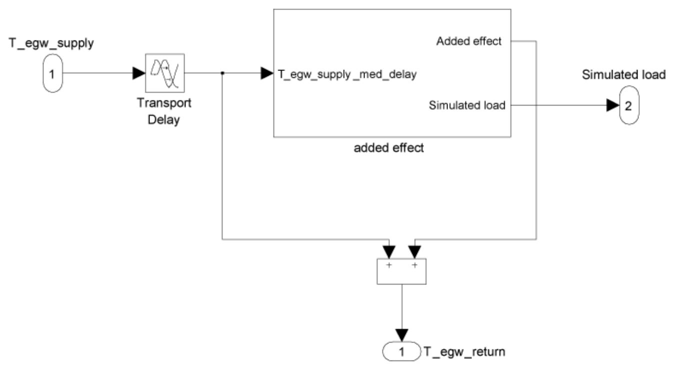

Figure 53: *Simulink model of Ambient mode; subsystem "Heating EGW".*

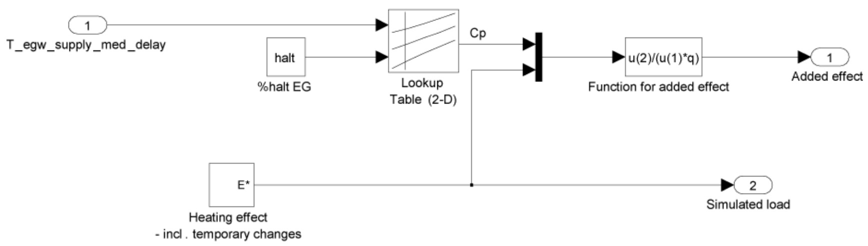

Figure 54: *Simulink model of Ambient mode; subsystem "added effect".*

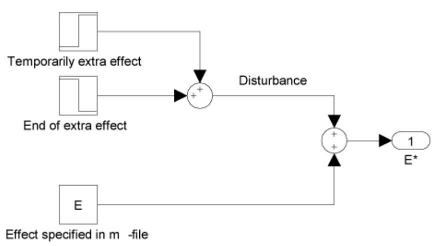

Figure 55: *Simulink model of Ambient mode; subsystem "Heating effect - incl. temporary changes".*