# Weather predictive control of power restricted two phase cooling system

# Weather Predictive Control

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Master's thesis

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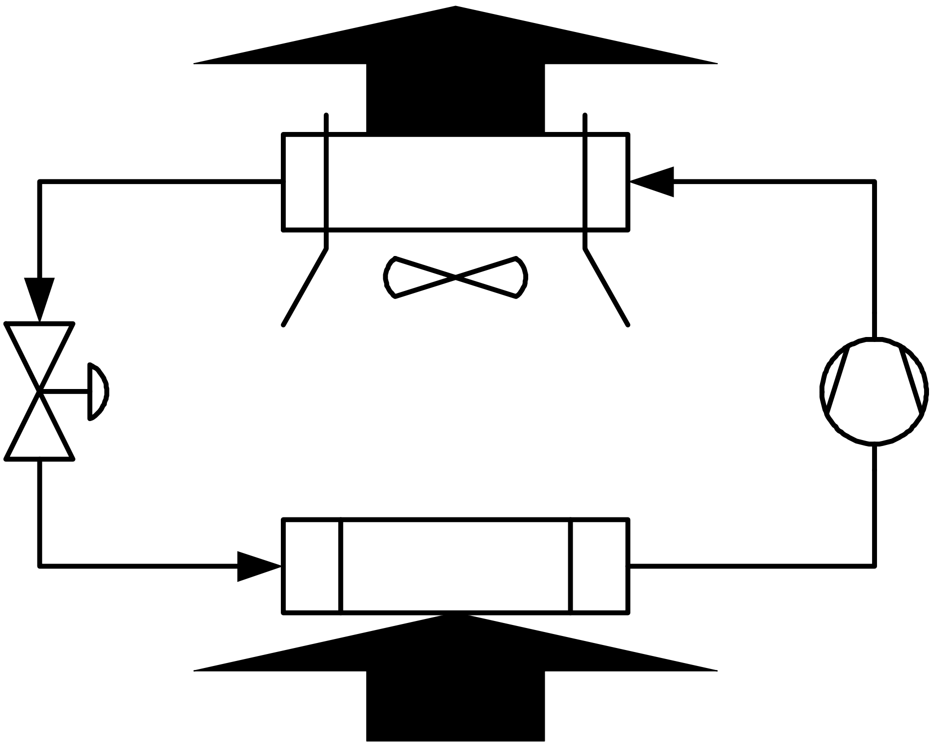


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**ABSTRACT:**

The goal of this project is to limit the maximum power consumption used and keeping the maximum temperature below a given temperature on the refrigerator system from Danfoss, which is setup in Aalborg University. High outside temperature causes high power consumption.

It is intended to use weather prediction to be able to keep both the temperature constraints and power consumption constraints. The prediction is intended to lower the temperature before the outside temperature is causing the power consumption to reach the limit.

To solve this control problem a Model Predictive Controller (MPC) is implemented. To implement this, models of the cooling system have to be derived and a Performance function has to be tuned.

It is expected that implementing MPC will ensure all constraints to be kept. Furthermore the average power consumption is expected to be lowered.

Simulation shows that all constraints are kept, but it also shows that the average power consumption is not lowered. Experiments support the simulated results.

Overall MPC is very useful to implement on cooling systems that have power consumption and temperature constraints. There will be improvements using MPC compared to a power restricted controller without prediction.

# Preface

This report is a Master thesis, authored by group 06gr1036B at Aalborg University in the period from February 1st to June 1st 2006, under supervision of Henrik Rasmusen and external supervisor from Danfoss, Lars Finn Sloth Larsen.

The report is addressed to Danfoss, students and engineers in the field of control, interested in MPC of cooling systems and implementation of MPC in general.

*Aalborg University, June 1th 2006.*


Philippe Zillinger Molenaar

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# Part I

# Introduction

# 1 Introduction

In our part of the world cooling is used in numerous places, for instance in buildings where air condition is installed or in refrigerator systems in stores etc. The basics for all cooling systems is that energy in the form of heat is moved from a cold to warmer place. This of course can not be done by just moving the air, instead cooling systems use boiling point and pressure differences, of the used cooling fluid to move the energy. Figure 1.1 is an illustration of a simplified cooling system.

A cooling system is build up of four main parts: compressor, condenser, expansion valve and evaporator. The evaporator is in the cold room that needs to be cooled and the condensor is placed outside. The principle of how the cooling system works will be explained in detail later.

When talking about large scale cooling systems as air condition installations in buildings or large cooling storages in factories or large shops, there are some issues that are not handled yet. One of the issues is related to the price of the electricity. When factories, shopping centers etc. apply to buy electricity they pay for the amount they use, but also for the maximum load used. This maximum load is expensive, which means that the companies that have a high peak load of electricity pay a lot of money for it. This is the reason that some companies have a diesel generator that they can switch on in situations of high peak loads, but this is also an expensive solution. In some places the peak load is also constrained by fuses that

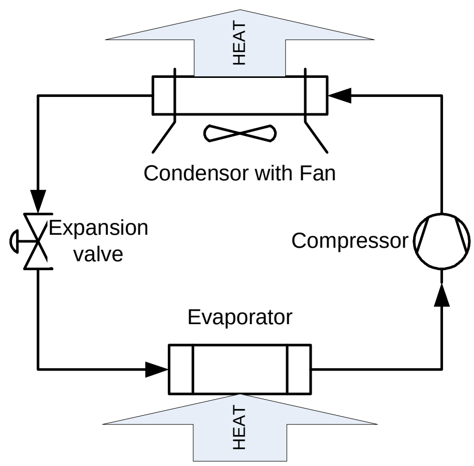


**Figure 1.1:** Illustration of a simplified cooling system

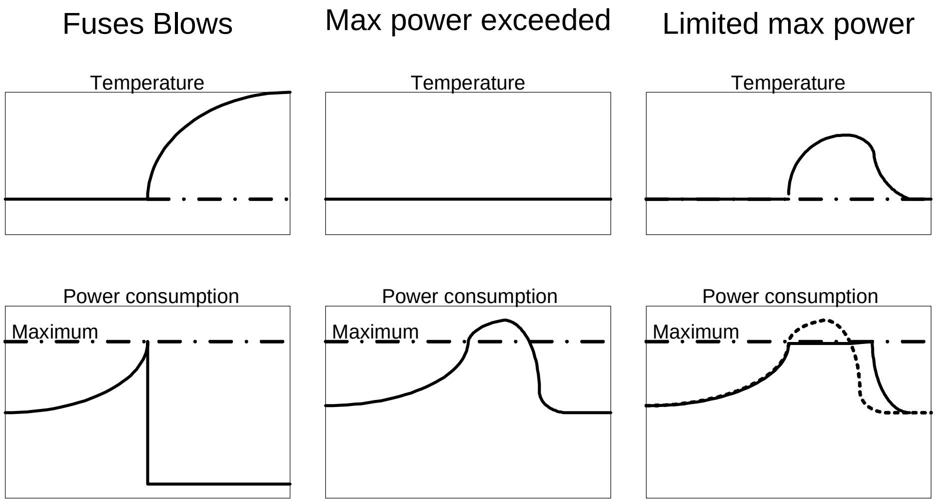


**Figure 1.2:** Illustration of temperature in peak loads situations

can blow and this makes the cooling system stop, after which there is no cooling available. Instead it will be useful to have a solution that can control the power consumption of the cooling system, in a way that the system will not use more power than a predefined limit. This means that the load can be set not to be larger than the specified peak load and the fuses will stay intact, thereby the cooling system will still provide cooling even if this cooling is not enough to hold the preferred temperature.

Three scenarios are illustrated in Figure 1.2. It is clear that the first scenario where the fuses blow is unwanted because the temperature is going to rise until it reaches the outdoor temperature, if the fuses are not replaced. The second scenario is unwanted because the power consumption is higher than the maximum allowed, although the required cooling will be obtained. In the third scenario the system does not use more power than the maximum load allowed and as a result of this the temperature will be higher than required. When the power consumption constrains allow it, the temperature in the cooling system is lowered to the desired level.

Because it is not desired to have the temperature in the cooling system to be above the reference temperature, a control system will be designed. The idea is to predict when the power consumption in the system will have to be limited because of the peak level. Before this situation occurs under-cooling the system in advance by lowering the reference temperature can be applied. This is illustrated in Figure 1.3 on the facing page. This causes the system to be at the desired cooling temperature at the present moment.

The goal for this project ensure that the temperature is newer above the reference temperature by making a weather prediction based controller handle this topic. In this control design it will be tried to keep the cooling efficiency during the (entire) operation time as high as possible, this is presented as the Coefficient Of Performance (COP). The power consumption must in the (entire) operation time be restricted to be below a given maximum limit.

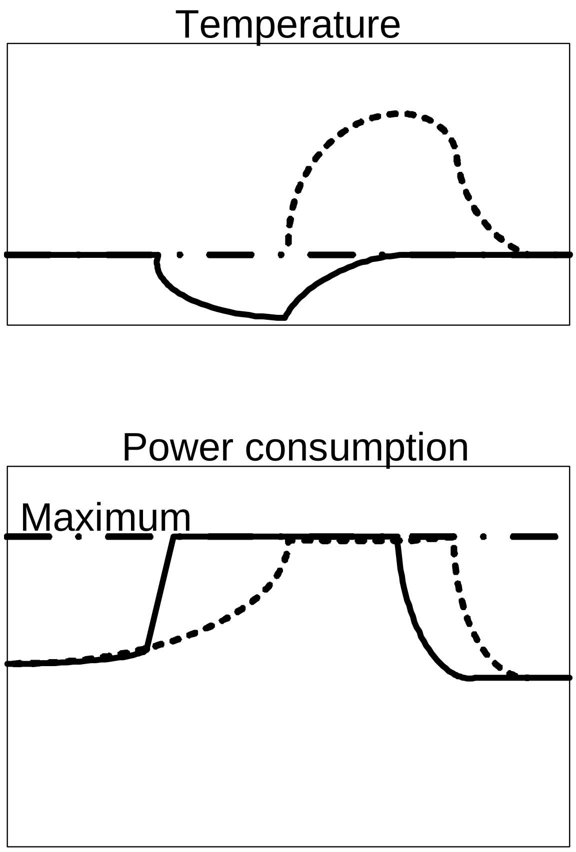


**Figure 1.3:** Illustration of how the deviation from the reference temperature can be minimized

# Model Description

# 2

## Contents



*The models explained in this chapter are not being used further on in the design, they are used to be able to understand how a cooling system works and thereby knowledge to design a controller. This chapter describes how the system is connected and what influences each other. This meaning is explaining the energy relations, how COP is calculated and what influences this. A energy relation is put up for the energy going in to the cooling room from the surroundings, to be able to simulate a real cooling room by using the water heater as the load. Finally a weather model is found to be able to simulate weather influence on the system.*

## 2.1 The Cooling System

As described in the introduction cooling systems utilize that the boiling point for a fluid changes for a different pressure, the higher the pressure, the higher the boiling point.

For this cooling system a cooling fluid named R134a (HFC-134a) is used, the molecular formula is $F_3CCH_2F$ [Airliquide 2006]. R134a has a boiling point at:

| Pressure | Boiling point |
|---|---|
| 1.013 bar | -26.6 °C |
| 3.5 bar | 5 °C |
| 4.9 bar | 15 °C |
| 5.7 bar | 20 °C |
| 13.2 bar | 50 °C |

This makes it possible to make a cycle where energy, in the form of heat, is moved from a room at -18 °C to the outdoor temperature at 20 °C. This calls for an example:

*The cooling system is build up as illustrated in Figure 2.1. The outdoor temperature is 20 °C and the temperature in the cooling room is -18 °C as mentioned above. To be able to move energy from the cold room to the outdoor air the pressure in the evaporator must be near 1.013 bar that gives a boiling point at -26.6 °C and thereby it is possible to move energy from the -18 °C cold room to the evaporator. The compressor can then compress the heated vapor from 1.013 bar to lets say 13.2 bar this gives the vapor a boiling point at 50 °C and because the outdoor temperature is 20 °C energy is transferred from the vapor inside the condensor to the air. By this the vapor is transformed into liquid with a temperature above 20 °C and below 50 °C, the temperature depends on how much air is blown trough the condenser and thereby how much energy is taken out of the cooling fluid. The expansion valve is inserted to return to the wanted low pressure. The low pressure must be low enough for the cooling fluid to be transformed into vapor before entering the compressor.*

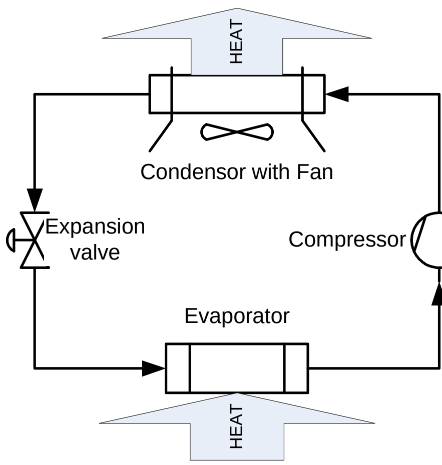


**Figure 2.1:** Illustration of a simplified cooling system

In order to be more specific a more fundamental understanding of thermodynamics is needed and to be able to make calculations some models of the different parts are needed too. This means energy models of the evaporator, the condenser and the compressor. The valve does not need an energy model as it doesn't apply or take energy out of the system. All this is included in the next section.

## 2.2 Thermodynamics

To give a good description of the energy transferred in the different phases of the cooling cycle of the system in Figure 2.3 on the following page, a pressure-enthalpy diagram is used. A illustration of a Pressure-Enthalpy diagram is shown in Figure 2.2. The steps in the diagram are described here:

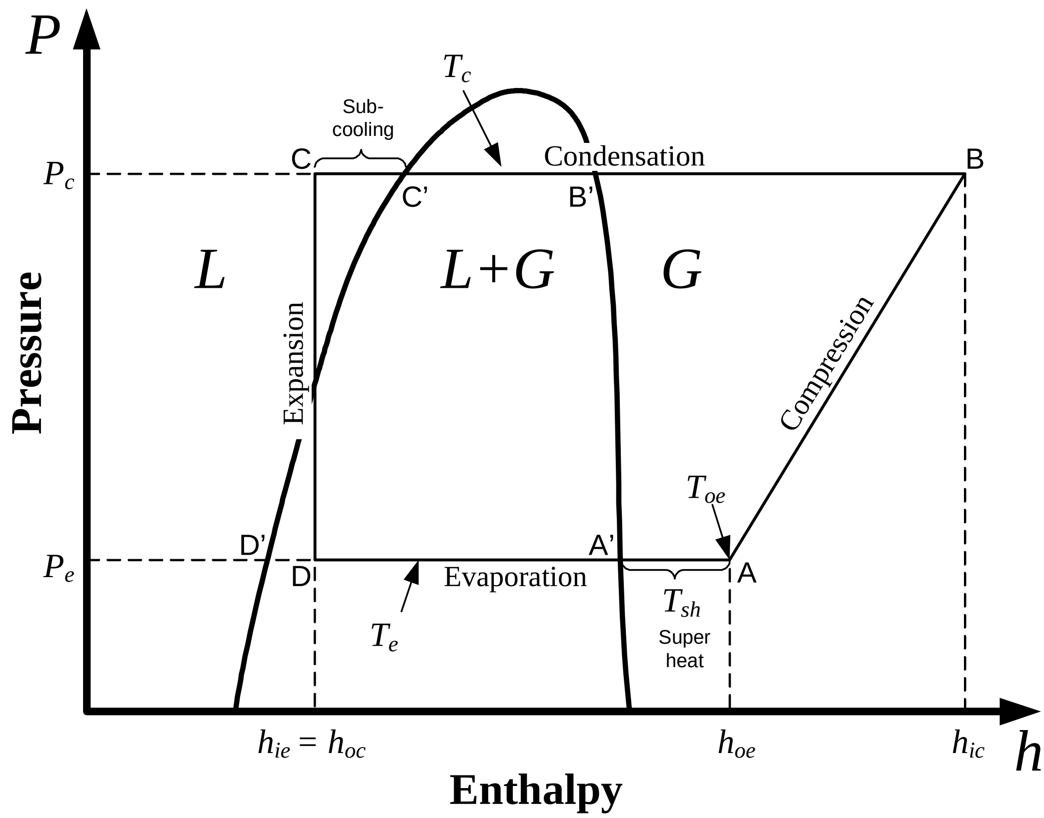


**Figure 2.2:** Pressure-Enthalpy diagram of the refrigerator system

**A to B**

The superheated vapor coming from the evaporator is compressed, and work is put into the system. The vapor pressure and enthalpy both increase. The temperature is increased to $T_c$, which is above the temperature of the medium used for rejection (here the ambient air). The work done by the compressor is given in equation 2.7 on page 18

**B to C**

The high pressure vapor condenses and is totally condensed into a liquid, reaching the point on the saturated liquid line C′. Further cooling to point C beyond this line, called subcooling, assures no vapor is left at the end of the condensing phase.

**C to D**

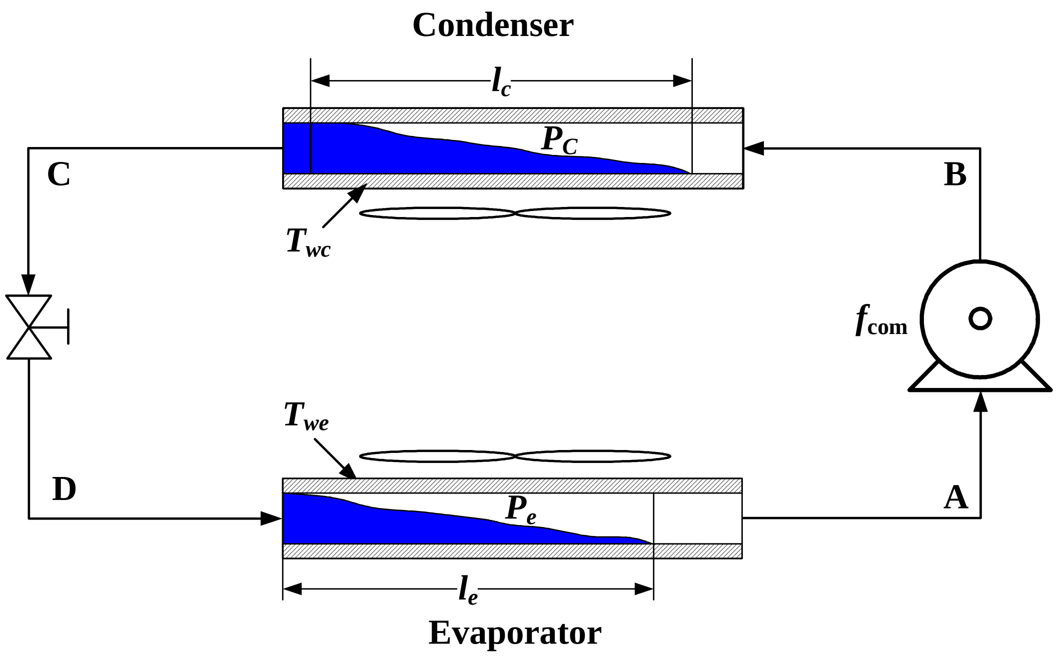


**Figure 2.3:** Scheme of refrigerator system

Before the liquid enters the evaporator it is led through a thermostatic expansion valve, which lowers its pressure and therefore its boiling point. For the concerned refrigeration plant the thermostatic expansion valve is utilized because of the necessity of maintaining a certain amount of superheat in the coolant.

**D to A**

Because the refrigerant is at a temperature below the medium to be cooled, it absorbs heat from the medium and boils. It changes phase from a liquid to a gas and reaches the saturated vapor line at point A'. In order for the refrigerant to change state, it must take in heat energy. During this transfer of heat energy, the refrigerant remains at a constant temperature. Any additional heat applied at constant pressure causes the refrigerant to enter the superheat region at point A' . The superheat is important, because this prevents liquid condensation, in the tube to the compressor. Liquid will damage the compressor. Increase in superheat from the evaporation phase has a corresponding increase in the total heat of rejection at the condenser and results in the compressor operating at a higher temperature.

## 2.3 Energy Equation

The total energy rate in the cooling system equals zero due to the law of energy conservation. The energy rates entering the system are the thermal energy absorbed by the coolant in the evaporator $\dot{Q}_{DA}$, and the energy applied by the compressor $\dot{W}_{AB}$, $\dot{Q}_{BC}$ is the energy rate leaving the system in the condenser.

The following for the conservation for the energy in the system holds:

$$0 = \dot{Q}_{DA} + \dot{W}_{AB} - \dot{Q}_{BC} \tag{2.1}$$

### 2.3.1 Evaporator

The evaporator is divided into a two-phase section and a superheated vapor section, see Figure 2.3 on the preceding page. The equation for the time rate change of the length of the two phase section $l_e$ is [He et al. 1998]:

$$\rho_{le} h_{lge} A_e (1 - \bar{\gamma_e}) \frac{dl_e}{dt} = -\dot{m}_{AB}(h_{A'} - h_D) - \alpha_{ie} \pi D_{ie} l_e (T_{we} - T_{A'}) \tag{2.2}$$

, in which $\rho_{le}$ is the mass density of refrigerator liquid, $\bar{\gamma_e}$ the void fraction in the evaporator, $h_{lge}$ the specific latent for going from liquid to gas, $A_e$ the size of the cross section of the evaporator.

The first term on the right-hand side of equation 2.2 corresponds to the energy storage rate due to the refrigerant flow with $\dot{m}_{AB}$ the refrigerant mass flow rate, the refrigerant enters the two phase section with enthalpy $h_D$ and exits with $h_{A'}$. The second term on the right represents the heat transfer rate from the tube wall to the two-phase refrigerant. $\alpha_{ie}$ is the heat transfer coefficient between refrigerant and inside of the evaporator, with $T_{we}$ and $T_{A'}$ the temperatures of the wall and refrigerant. $D_{ie}$ is the inside diameter of the evaporator.

The derivations of the equations in the next sections for the evaporator, compressor and condensor are taken from [Kallager et al. 2005], Chapter 4 Modeling .

Because the superheat is kept constant, the value of $l_e$ is kept constant eliminating the dynamical aspect of it. This makes the left side of the equation zero, giving:

$$T_{we} = \frac{\dot{m}_{AB}(h_{A'} - h_D)}{\alpha_{ie} \pi D_{ie} l_e} + T_{A'} \tag{2.3}$$

Considering the heat load circuit the following energy balance is given:

$$\frac{dE}{dt} = \dot{m}_{hlc} C_{hlc} (T_{o,hlc} - T_{i,hlc}) - \dot{Q}_{we} \tag{2.4}$$

,the inlet temperature of the water in the heat load circuit is $T_{i,hlc}$ and $T_{o,hlc}$ the outlet temperature. $\dot{m}_{hlc}$ and $C_{hlc}$ are the mass flow rate and specific heat of the water. $\dot{Q}_{we}$ is the heat transfer rate to the evaporator.

The overall energy conservation implies that energy from the water will be absorbed by the coolant in the evaporator along the evaporation length:

$$\dot{m}_{hlc} C_{hlc} (T_{o,hlc} - T_{i,hlc}) = \dot{m}_{AB}(h_A - h_D) \tag{2.5}$$

extending the steady state expression 2.5 gives the equation for the calculation of temperature of the refrigerant $T_{A'}$:

$$\frac{dT_{A'}}{dt} = \frac{\dot{m}_{hcl} C_{hlc} (T_{o,hlc} - T_{i,hlc}) + \dot{m}_{AB} (h_A - h_D)}{m_e C_e} \tag{2.6}$$

, $m_e$ is the mass of the refrigerant in the evaporator and $C_e$ the specific heat.

### 2.3.2 Compressor

The work done in the compression step from point A to B in Figure 2.2 on page 15 is given by:

$$W_{AB} = (P_c - P_e) V_{com} f_{com} \tag{2.7}$$

, in which $P_c$ and $P_e$ are the condensation and evaporation pressure. $V_{com}$ and $f_{com}$ are the compressor volume and frequency.

The mass flow through the compressor is calculated by:

$$\dot{m}_{AB} = \alpha_1 P_A f_{com} \tag{2.8}$$

, $\alpha_1$ is a constant which is estimated in [Kallager et al. 2005], p.137.

### 2.3.3 Condensor Model

The condenser is modeled according to the evaporator. The formula for the wall temperature of the condenser $T_{w,con}$ is given by:

$$T_{w,con} = T_C - \frac{\dot{m}_{AB}(h_B - h_C)}{\alpha_{i,con} \pi D_{i,con} l_{con}} \tag{2.9}$$

, in which the heat transfer coefficient from condenser to the ambient air is $\alpha_{i,con}$, $D_{i,con}$ the inside diameter of the condenser tubes with length $l_{con}$ .

The heat is transferred from the coolant to the ambient air using a fan. The following holds:

$$\dot{m}_{AB}(h_B - h_C) = \dot{m}_{fan} C_{air} (T_{o,fan} - T_{i,fan}) \tag{2.10}$$

, the airflow generated by the condenser fan is $\dot{m}_{fan}$, cooling the condenser tube walls. The inlet temperature of the fan is $T_{i,fan}$, the outlet temperature of the cooling side is $T_{o,fan}$. $C_{air}$ is the specific heat of air

The following can be derived in the same way as the evaporator:

$$\frac{dT_C}{dt} = \frac{\dot{m}_A(h_B - h_C) - \dot{m}_{fan} C_{air} (T_{o,fan} - T_{i,fan})}{m_{con} C_{con}} \tag{2.11}$$

### 2.3.4 Expansion Valve

The expansion valve is a mechanical control valve from Danfoss.

It is assumed that the pressure propagation so fast, that there will be no difference in the mass flow across the evaporator. Due to this assumption, the coolant mass flow can be considered constant throughout the system, and equal to the flow through the compressor:

$$\dot{m}_{AB} = \dot{m}_C = \dot{m}_D = \dot{m}_A = \dot{m}_B \tag{2.12}$$

## 2.4 COP and System Parameters

The objective for any cooling system is to minimize the energy consumption of the cooling system, without compromising the cooling performance of the system.

The Coefficient of performance (COP) is used as a measure of the energy efficiency of the cooling system. Equation 2.13 shows how COP is calculated.

$$COP = \frac{\text{Energy moved}}{\text{Power consumption}} = \frac{Q}{E} = \frac{Q}{W_{AB} \cdot \eta} \tag{2.13}$$

,$\eta$ is the efficiency of the compressor. $Q$ and $W_{AB}$ are given in Figure 2.4 on the next page, which is a pressure-enthalpy diagram of the cooling fluid in the system in which a refrigeration cycle is added. In general there are some different parameters that can be adjusted in the cooling system, Mass flow, Evaporator pressure, Condensor pressure, Super heat temperature and the Sub-cooling temperature. To help explaining these parameters the pressure-enthalpy diagram is used again.

If neglecting the fan on the condensor then the compressor is the only component that needs electrical power. The compressor power consumption is, as described in Section 2.3.2, depending of the pressure differences ($P_C$-$P_E$) and the mass flow ($\dot{m}$). This gives three differen parameters that can be adjusted. Two more parameters are Sub-cooling($T_{SC}$) and Superheat ($T_{SH}$). That gives a total of 5 parameters to change. In order to describe the effect of each parameter, all parameters are kept constant except the one described, this is to illustrate the principle:

**$P_C$:** A way to make the cooling system more effective is to lower the condensor pressure ($P_C$) and keep the evaporator pressure constant, thereby Q is constant and $W_{AB} \cdot \eta$ is smaller. To do this the the fan on the condensor must move more air trough the condensor. The physical limit when running at maximum speed with the the fan.

**$P_E$:** If this pressure is increased the evaporator will not transfer enough energy from the cold room to the cooling fluid as the increase in $P_E$ will increase the temperature on the evaporator thereby lowering the cooling capacity.

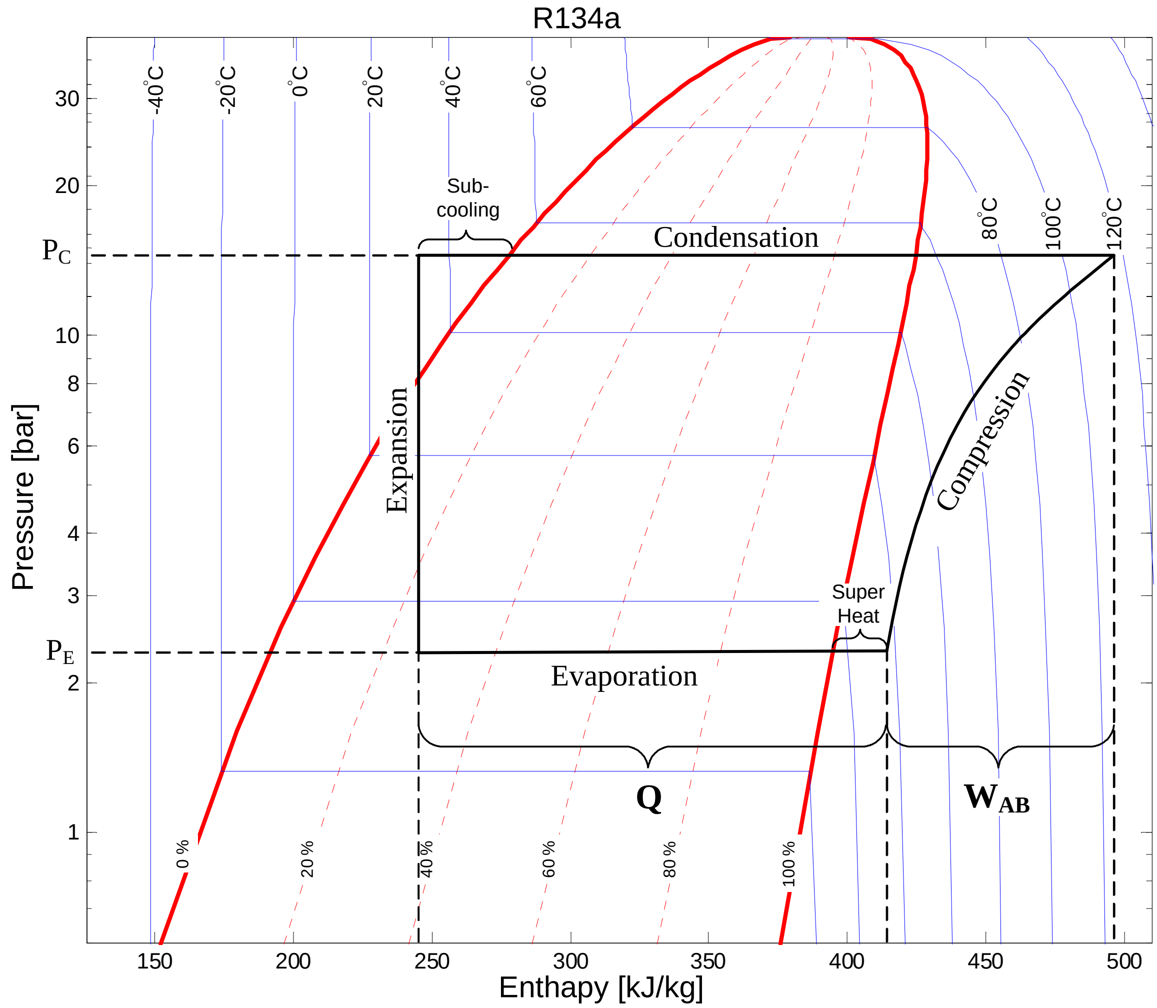


**Figure 2.4:** Pressure-Enthalpy

**$\dot{m}$:** By increasing the mass flow the cooling capacity will also increase, but to increase the mass flow the compressor must run faster and thereby use more energy.

**$T_{SC}$:** Sub-cooling can be used to transfer more energy without increasing the compressor speed, instead the condensor fan must run faster or an extra cendensor must be added to the system to cool the cooling fluid more.

**$T_{SH}$:** It is essential that there is a super-heat margin to guarantee complete vapor, because the compressor cannot compress fluid and will break down. On the other hand the larger this margin becomes, when the same amount of cooling is required, the lower the COP gets because the temperature of the vapor increases and thereby also the volume according to Equation 2.14 on the facing page, the equation is only valid for ideal gas as in the super heat region. Increase in volume means that the compressor must run at higher speed to move the same amount of cooling fluid, this result in higher energy consumption.

$$P \cdot V = n \cdot r \cdot T \tag{2.14}$$

There are three actuators in the cooling system, The compressor, the condensor fan and the expansion valve. A normal control of this system is to have a controller for the expansion valve to control the super heat ($T_{SH}$). Another controller for the condensor fan to control the condensor pressure($P_C$) and the last controller for the compressor to control the evaporation pressure ($P_E$). How these controllers are implemented is described in Chapter 3 on page 27

To implement load on the cooling system a water heater is connecter, this is described in the next section.

## 2.5 The Water Heater

The water heater is connected to the cooling system as a heat load, meaning that the water heater is put in to simulate the cold stored room. This means that the water heater has to behave like a normal cold stored room. Doing this it has to give energy to the cooling fluid as described in the subsection 2.6.2 on page 23. It must be possible to lower the temperature in the water heater when doing this the energy but in to the tank must increase according to Equation 2.17 on page 23.

The water heater is build up by a tank for water and an electric heat element it is assumed that the tank is isolated and thereby goes all the energy to heat the water and it is thereby also assumed that no energy goes from the water to the surroundings.

This gives an energy equation as shown in Equation 2.16

$$Q_{IN} = C_{water} \cdot m_{water} \cdot \Delta T \tag{2.15}$$

$$\frac{dT}{dt} = \frac{Q_{COOL} - Q_{IN}}{C_{water} \cdot m_{water}} \tag{2.16}$$

The weather does influence the cooling system this is described in the following section.

## 2.6 Weather Model

To be able to simulate the weather influence on the cooling system in the research laboratory a model of the weather is needed and understanding how this will affect the system to make a simulation of this weather.

### 2.6.1 The Weather Influence

Any cooling system is limited one way or another so it can only deliver a given amount of cooling. For a normal cooling system the limitation is in the amount of cooling fluid that can be circulated in the system and the pressure that the compressor can deliver and of course also the pressure that the system can withstand. As described earlier these limitations give a maximum of energy transferred from the cold stored room to the surrounding environment (Outdoor), this means that if it takes a certain amount of energy to hold the cold stored room at -18 °C when the outdoor temperature is 20 °C, then it is not possible to hold the stored room at -18 °C if the outdoor temperature increases without using more energy.

This means that the surrounding temperature influence on how much energy that is needed to keep the stored room cool. The temperature over 24 hours does vary significantly and even if looking in differences between midsummer temperature and temperature changes in September, there are some differences. Figure 2.5 illustrates how the temperature curve could looks in the midsummer and in September, both have a maximum temperature at 23 °C but it is clear that in the midsummer the temperature is higher than 20 °C over a longer period than in September.

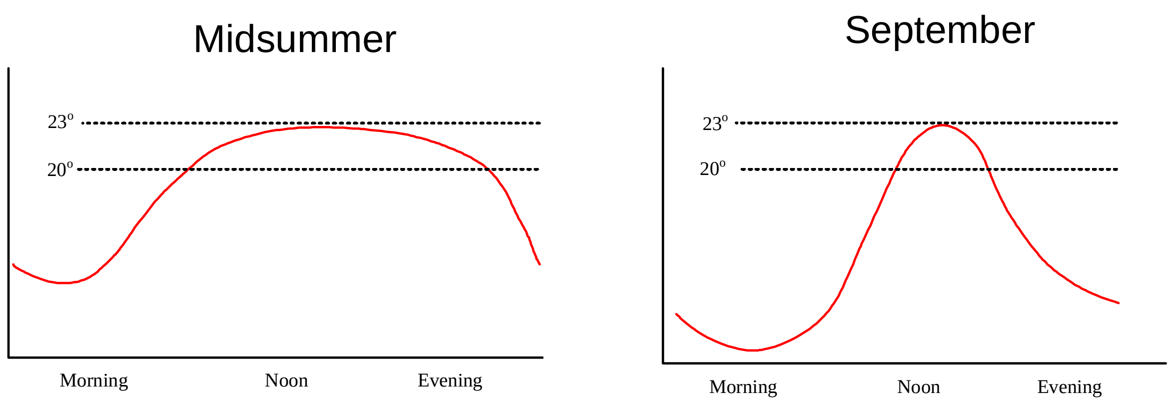


**Figure 2.5:** Illustration graph of temperature over 1 day.

Lets say that a specific cooling system has a maximum cooling capacity to hold the temperature at -18 °C in the cool stored room when the outdoor temperature is 20 °C. This can be limited by a maximum power consumption or the system itself. It is clear that this system can not hold the temperature at the preferred temperature in either of the two days illustrated in Figure 2.5 and by looking at the figures it is also clear that there will be a larger deviation from the preferred temperature in the midsummer example, due to the outdoor temperature is above 20 °C in a longer period.

To be able to say more about the temperature changes in the cold stored room it will be necessary to get more information about the cooling system combined with the stored room.

### 2.6.2 The Stored Room

The laboratory system used in this project, does not have a real cooling room but it is possible to simulate a stored room by increasing or decreasing the load made by the water heater.

Lets assume that the temperature in the cold stored room must be -18 °C and the temperature outside the cold stored room is 20 °C that makes a temperature difference on 38 °C. The energy going from the surroundings in to the cold stored room can be calculated from Equation 2.17

$$Q_{IN} = C_{wall} \cdot A \cdot \Delta t = C_{wall} \cdot A \cdot (T_2 - T_1) \tag{2.17}$$

$Q_{IN}$ : Energy flow from the surroundings to the stored room [W]
$C_{wall}$ : Specific thermal conductivity $[W/(m^2 \cdot \Delta t)]$
$A$ : Surface area, including wall, floor and ceiling, $[m^2]$
$T_2$ : Surrounding temperature $[^\circ C]$
$T_1$ : Temperature inside the stored room $[^\circ C]$

We will assume that the energy coming from the surrounding environment is 1500 W when the surrounding temperature is 20 °C and the temperature in the cold stored room is -18 °C. This means that $C \cdot A = 39,6 \approx 40$. Equation 2.18 shows the model of the energy going from the surroundings to the cold storage room.

$$Q_{IN} = E_{surrounding} = 40 \cdot \Delta t = 40 \cdot (T_2 - T_1) \tag{2.18}$$

How fast it is possible to change the temperature in the stored room depends on how much energy is taken out by the cooling system, the amount of energy coming in to the room from the environment and the room specific heat capacity, including the products stored in it, assuming that the temperature of the products is the same as the temperature of the cooling room. This is written in Equation 2.19

$$\begin{aligned} Q_{product} &= Q_{IN} - E_{COOL} \\ C_{product} \cdot m_{product} \cdot (\Delta T_{product}) &= Q_{IN} - Q_{COOL} \\ &\Updownarrow \\ \Delta T_{product} &= \frac{Q_{IN} - Q_{COOL}}{C_{product} \cdot m_{product}} = \Delta T_{ROOM} \end{aligned} \tag{2.19}$$

$C_{product}$ : Specific heat capacity of the product [kJ/(kg $\cdot \Delta t$)]
$m_{product}$ : Mass of the product [kg]
$\Delta T_{product}$ : Temperature change of the product or the cooling room $[^\circ C]$

### 2.6.3 Weather

In order to get an overview of how the temperature changes in a month a temperature graph over the maximum and minimum temperature are plotted for July, [DMI 2005].

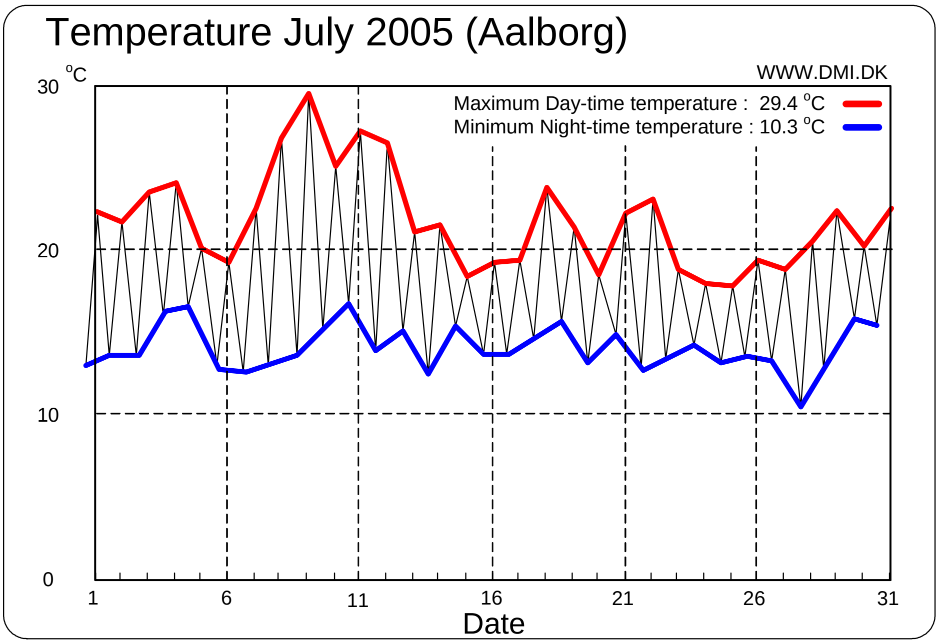


**Figure 2.6:** Graph over July temperatures in Aalborg.

It is assumed that the minimum temperature is in the night and the maximum is at midday, as illustrated in the 24 hour graphs in Figure 2.5 on page 22.

As we all know the temperature changes over the year this can also be seen in Figure 2.7 on the next page where the average temperature from 1961 to 1990 is plotted.

#### Model of Weather

If a simple model of the temperature must be made, it could be an idea to simplify the temperature changes to be sinus waves. One sinus wave for illustrating the slow changes over the year and a faster sinus wave to illustrate day and night temperatures.

The sinus that represents the year curve must have an average of approximately 7.7 °C and a peak amplitude at approximately 15.4 °C, found in Figure 2.7. But because this project is trying to handle peak load situations the year curve is neglected due to the fact that there are no significant peak loads in the winter period only the midsummer is of interest to this project.

The day to day temperature cycle could be a sinus with the peak to peak amplitude in the temperature of approx. 5 °C in the winter and approx. 8 °C in the summer, found on

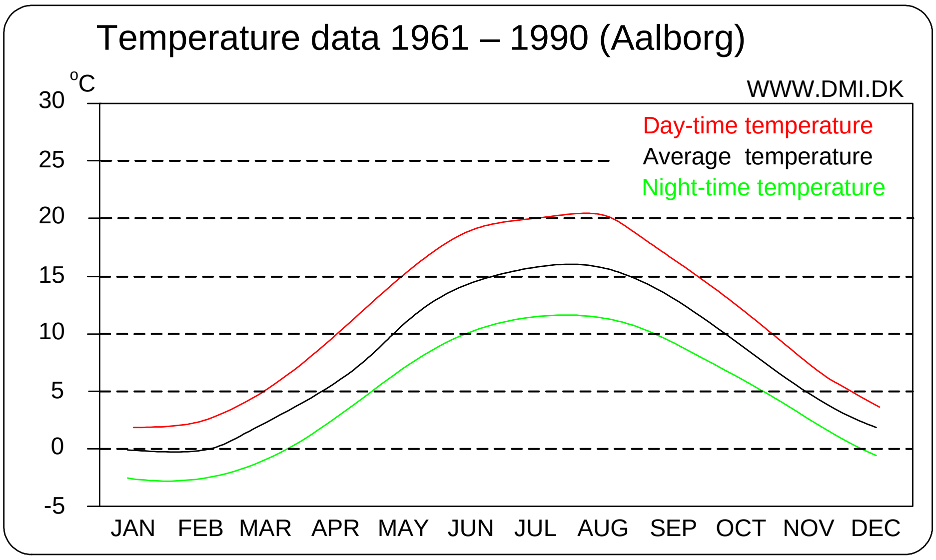


**Figure 2.7:** Average graph of temperature from 1961 to 1990,[DMI 2005]

Figure 2.7. This will mean that the temperature change in the day cycle is some how related to the average temperature in the month. This is described mathematically in Equation 2.20, where the Average temperature is approx. 17 °C found on Figure 2.7 for the midsummer.

$$\begin{aligned} \text{Daycycle} &= \left(5 + \frac{\text{Average temp.}}{5}\right) \cdot sin(1/24 \cdot t) + \text{Average temp.} \\ &= 8.4 \cdot sin(1/24 \cdot t) + 17 \qquad (2.20) \\ t &: \text{Time in hours} \end{aligned}$$

This is not the final solution because there are some unknown factors that has to be applied. In a sunny day the temperature is usually higher than i a cloudy day also the wind and the precipitation changes the temperature as well. Tease unknown factors will result in a larger or a smaller temperature difference between day and night, but it will also be able to change the mean temperature by some offset. When this is applied to the equation the weather model has the structure as shown in Equation 2.21.

$$\begin{aligned} \text{Weather model} &= \xi_1 \cdot 8.4 \cdot sin(1/24 \cdot t) + 17 + \xi_2 \qquad (2.21) \\ t &: \text{Time in hours} \\ \xi_1 &: \text{Stochastic change in temperature amplitude} \\ \xi_2 &: \text{Stochastic change in temperature offset} \end{aligned}$$

# Existing System

3

## Contents



*This chapter contains information of the controllers used in the inner loops of the cooling system. All three controllers are briefly descried. The controllers in this chapter are designed by the 7. semester group and described in [Kallager et al. 2005].*

As explained earlier three controllers are needed to control the cooling system. A Fan controller, a Compressor controller and a Super heat controller. Figure 3.1 illustrates how the controllers are connected to the cooling system. They are described in the following.

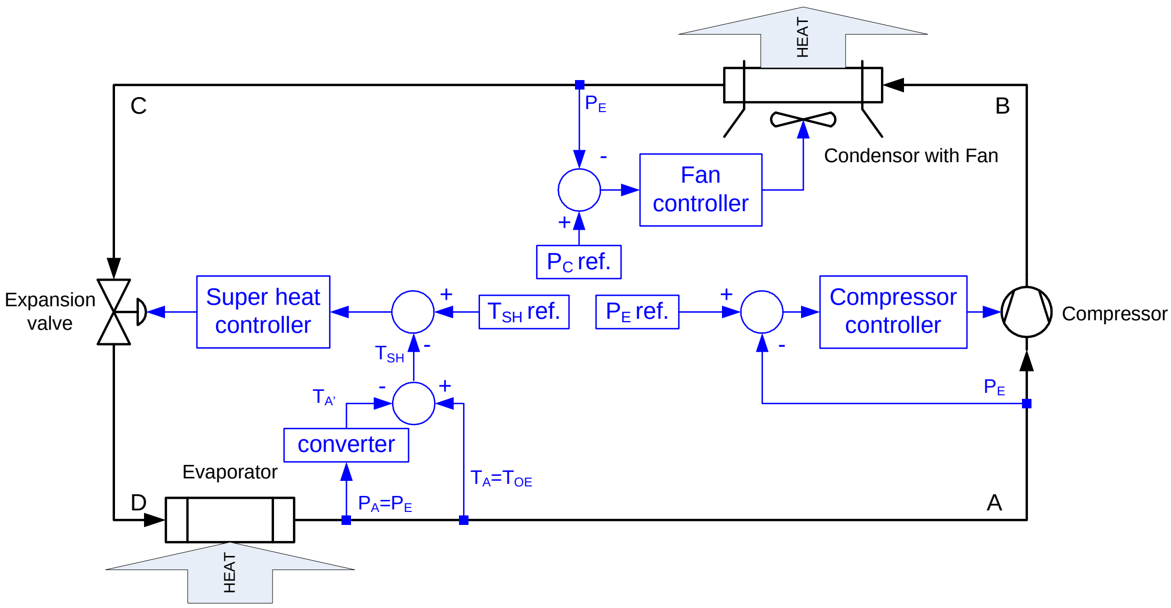


**Figure 3.1:** Fig: Illustration of the three controllers on the system

## 3.1 Super Heat Controller

The super heat controller is a mechanical controller made by Danfoss A/S. The principle is the same as shown in Figure 3.1 on the previous page but some in a mechanical way instead if making a electrical controller.

The valve is build up as illustrated in Figure 3.2, where $T_A$ is the temperature measurement on the outlet of the evaporator. This temperature measurement is made converted to pressure directly by the container drawn by dotted lines. This container has the same gas inside as the cooling system and will there for generate the same pressure as the evaporator for the same temperature. Because the measurement is made on the outlet of the evaporator the temperature is higher and thereby will the pressure in the container also be higher. If this force from the temperature measurement is higher than the pressure in the evaporator plus the force coming from the spring inside the valve, the the valve will open more and visa versa. This will thereby keep the correct super heat temperature.

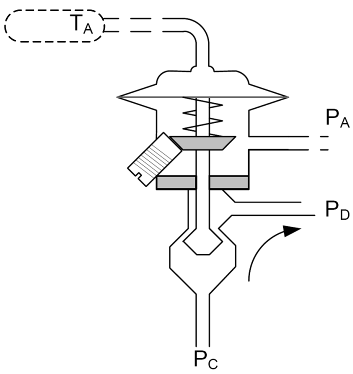


**Figure 3.2:** Basic principle behind a thermostatic expansion valve.

As the expansion valve controls the superheat inside the evaporator, it is assumed to be a process within the evaporator. The pressure wave is assumed to be so fast, that there will be no difference in pressure trough the evaporator and the condensor and thereby will the mass flow trough the system be the same. Due to this assumption, the coolant mass flow can be considered constant throughout the system, and equal to the flow through the compressor.

## 3.2 Fan and Compressor Controller

The condensor fan controller controls the fan speed signal and is set up to control the outlet pressure on the condensor and thereby the reference for this controller is chosen to be the condensation pressure $P_C$. This is proportional to the condensation temperature $T_C$. The compressor controller on its turn controls the compressor speed signal and is set up to control the evaporation temperature $T_e$, this yields for the evaporation pressure $P_e$ to be set as reference, as this is proportional to the evaporation temperature. The condensor model and evaporator model used are described in section 2.3.

A PI-type controller is chosen for both controllers since it is commonly used and eliminates steady-state errors. The fan and compressor speed signal both have an upper and lower limit, which is modeled with a non-linear saturation block in the feedback loop with controller. When the fan voltage does not saturate the system behaves as a closed loop. The feedback loop will however be broken when the actuator saturates because the output of the saturating element is then not influenced by its input. The unstable mode in the controller may then drift to very large values. When the actuator is not saturated any more, it may then take a long time for the system to recover. It may also happen that the actuator bounces several times between high and low values before the system recovers.

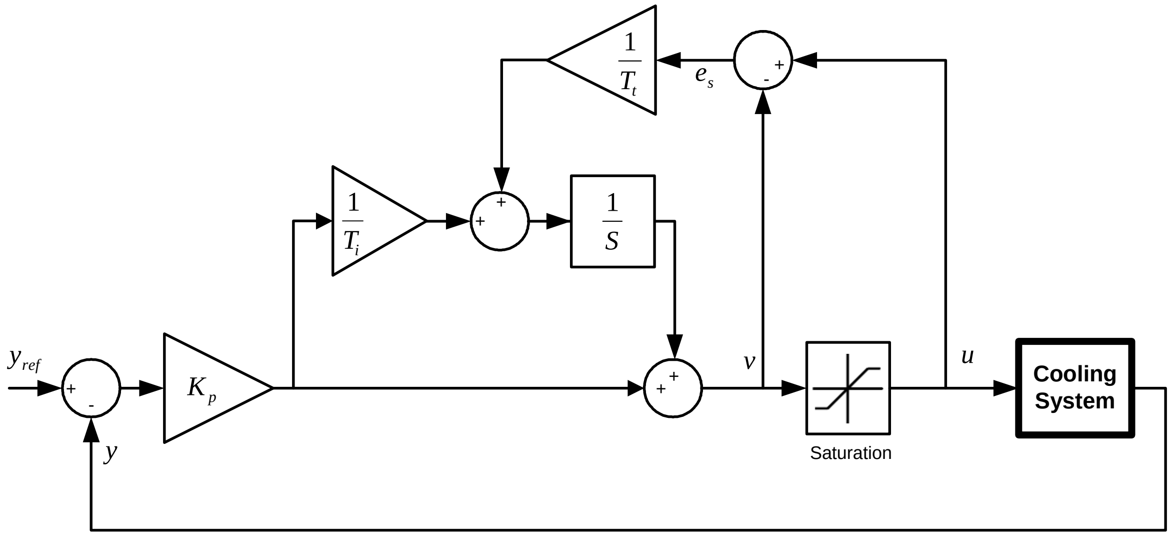


**Figure 3.3:** PI fan controller with Anti Integrator Windup

The remedy for the integration windup effect, called Anti Integrator Windup (AIW) chosen here is adding an internal controller feedback acting on $u - v = e_s$, see Figure 3.3. The signal $e_s$ is fed to the input of the integrator through gain $\frac{1}{T_t}$. The signal is zero when there is no saturation. Under these circumstances it will not have any effect on the integrator. When the actuator saturates, the signal is different from zero and it will try to drive the integrator output to a value such that the signal $v$ is close to the saturation limit.

The tuning of the fan-controller parameters will be done using the Ziegler-Nichols frequency response method, where a process is brought to oscillate using relay control. This method finds the ultimate gain and ultimate period of the given process. Obtaining the best possible speed is not of significant importance as other parts of the system are much slower than the fan.

The tuning of the compressor controller is based on information from a step test on the system and then by approximating the step response to a first order characteristic, giving an approximation of the system which is to be controlled. The integration action time $T_i$ is set to be equal to the time constant of the first order approximation, compensating for the most significant pole of the system and using Routh's stability criterion the gain $K_p$ is determined. Furthermore a relay is introduced in the controller, that has the purpose of

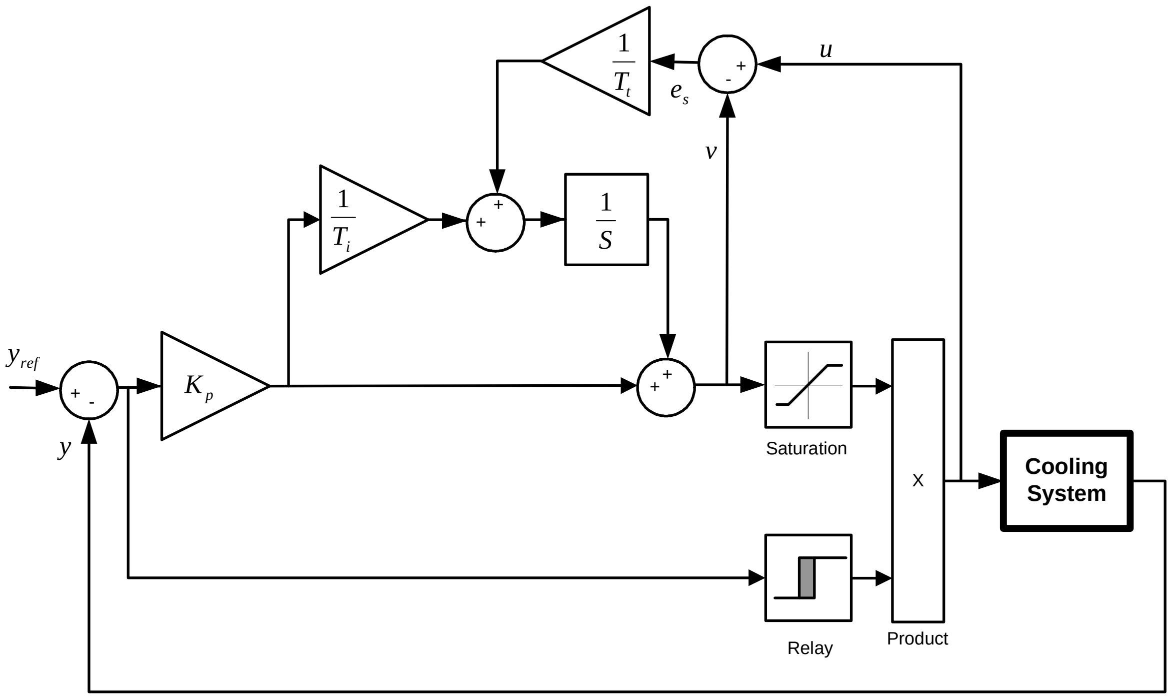


**Figure 3.4:** PI compressor controller with Anti Integrator Windup including relay

disabling the controller output, whenever there is an overshoot of more than 0.1 bar. This is done due to the fact, that if more cooling has been provided to the system than necessary, it isn't energy efficient to keep the compressor running when the pressure increases again afterwards. Instead the compressor signal will be zero and the ambient temperature will do the work of increasing the pressure as fast as possible, since the compressor will not counteract this. The modified controller is illustrated in Figure 3.4.

## 3.3 Conclusion

It is concluded that the controllers made by the 7. semester group more or less can be used in the system. But because they can not directly be implemented there are no need for validation at this point. Both controllers will be changed further on in the report and when the change has been implemented a validation will be made. The super heat controller is the mechanical controller and will only be verified through other test and as long it is able to keep a super heat level between 0 and 14 $^{\circ}C$ and the steady state is near 8 $^{\circ}C$ no further validation will be made.

# System Performance 4

## Contents



*This chapter starts by expressing what will be evaluated on the system, thereby giving the performance objectives. Next is establishing a connection between a real cooling system where the load is a cooling room with e.g. frozen meat and relating this to the laboratory cooling system with a water heater generating the load. After this connection is established the working range of the laboratory is found and a operating point is found. Finally a connection between this operating point and a real cooling system is described.*

As stated in the main introduction the main objective is to keep the power consumption of the system below a given maximum load, since power use above it is assumed to be very expensive. At the same time it is desired to not go above a certain cooling reference temperature, meaning that independent of the weather, a sufficient low cooling temperature in the cooling room is always guaranteed. Implementation of a weather predictive controller, which will lower the cooling room temperature in advance before the outdoor temperature increases, is expected to be able to keep these limitations. Implementation of this predictive controller is also expected to increase the Coefficient Of Performance to the case without weather prediction. Next to this it is desired to totally have a lower power consumption, since works has to be paid for. This is generally expected as a result on the constraints on the maximum level of power consumption and because the COP is assumed to be higher, due to prediction.

Main Objectives:

1. Maximum limit on power consumption used by the compressor on the cooling system, i.e. smaller than the maximum load.
2. Maximum limit on the temperature in the cooling room, which must be kept, under all weather conditions.

3. Higher COP, compared to system without weather prediction, due to MPC controller.

4. Lower the total power consumption of the system due to prediction.

COP is one way to calculate how well a cooling system performs, instead of COP, it can in some cases be more relevant to calculate power consumption per degree (PCPD). Equation 4.1 shows how PCPD is calculated.

$$\begin{aligned} \text{PCPD} &= \frac{W}{T} \qquad (4.1) \\ W &= \text{Power consumption of the cooling system [W]} \\ T &= \text{Temperature in cooling room } [^\circ C] \end{aligned}$$

The following section will make the connection between a real cooling system and the cooling system setup in the laboratory.

## 4.1 Real World To Lab-Setup

The link between a real cooling system operating in a super market or a firm, is described in this section. It is intended to give the reader knowledge of how the laboratory cooling system can be compared to a cooling system with heat loads coming from products that need to be cooled and the energy in the form of heat going trough the walls into the the cooling room. Figure 4.1 on the facing page illustrates an imaginary cooling room inside a building.

In the figure it can be seen that energy is going into the cooling room through the walls from the inside working area, this energy is called $Q_{IN}$. $Q_{IN}$ can be calculated from Equation 4.2, where $C_{WALL}$ is the isolation coefficient for the wall and $A_{WALL}$ is the unified wall area. Assuming that the floor and the ceiling are included, the isolation coefficient is called $C_{ROOM}$.

$$\begin{aligned} Q_{IN} &= C_{WALL} \cdot A_{WALL} \cdot (T_{ROOM} - T_{COOL}) \qquad (4.2) \\ &= C_{ROOM} \cdot (T_{ROOM} - T_{COOL}) \end{aligned}$$

Assuming that the indoor temperature ($T_{ROOM}$) is not influenced by the outdoor temperature ($T_{OUT}$) due to the fact that the indoor temperature is controlled by an air-condition, the weather will not give any influence on $Q_{IN}$. Only the cooling room temperature $T_{COOL}$ will influence it, this if $T_{ROOM}$ is constant. Under normal conditions it must be -18$^o$C in the cooling room and +20$^o$C in the working area.

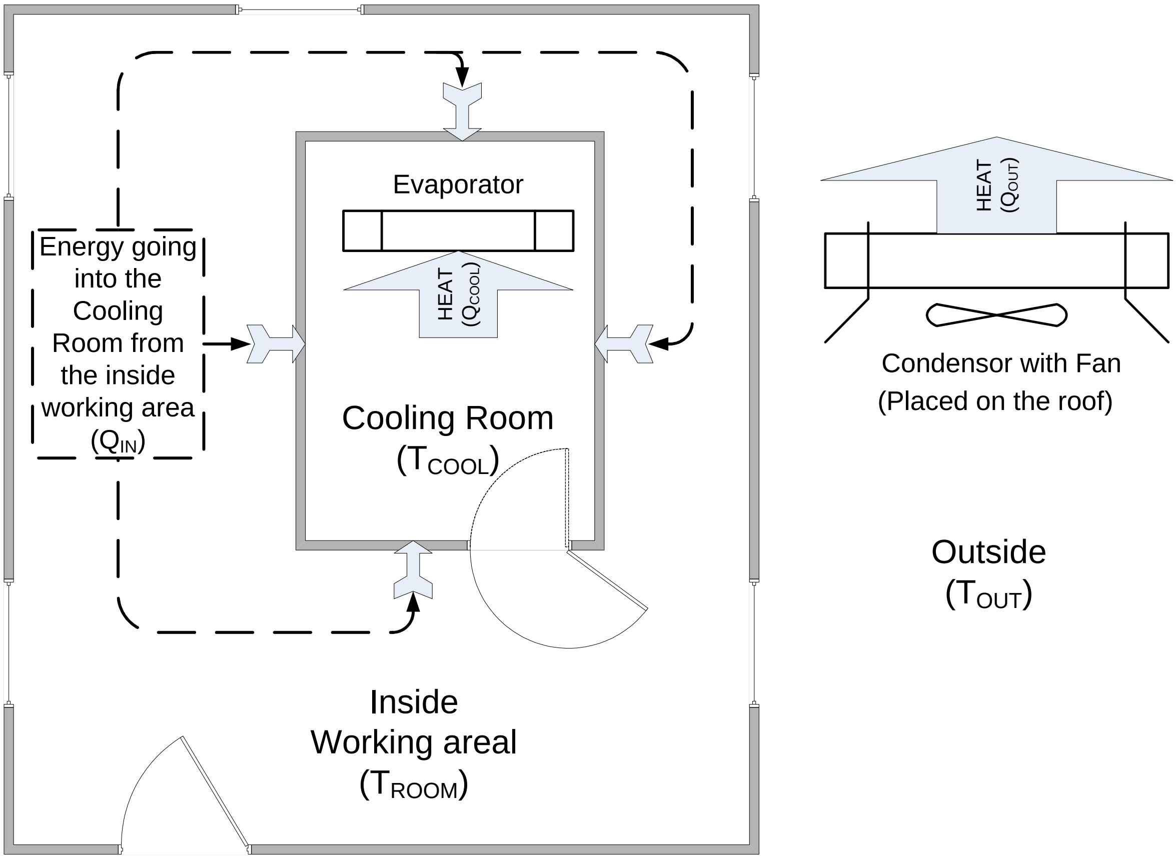


**Figure 4.1:** Illustration of a normal cooling room

### 4.1.1 The Weather Influence

As shown in Figure 4.1 the outside temperature T$_{OUT}$ only effects the condensor on the roof. In a real cooling system the energy taken out of the condensor is dependent on the amount of air the fan blows trough the condensor and the temperature difference over the fan, according to section 2.3.3 on page 18. Lets assume that the fan is always running at full speed to cool the condensor as much as possible. In this situation the air flow trough the condensor is constant and the heat taken out by the fan is only dependent on the outside temperature and condensor wall temperature T$_{CONDENSOR}$.

The energy that has to be taken out of the condensor by the fan, is equal to the energy taken out of the cooling room (Q$_{COOL}$) plus the work put in by the compressor (W$_{COMP}$). This is written in Equation 4.3, where $\xi$ is an air constant.

$$\begin{aligned} Q_{OUT} &= Q_{COOL} + W_{COMP} \\ Q_{OUT} &= f_{FAN} \cdot \xi \cdot (T_{CONDENSOR} - T_{OUT}) \end{aligned} \quad (4.3)$$

$W_{COMP}$ is assumed constant here and thus not influenced by the condensor while constant $Q_{COOL}$, thus considering a constant temperature in the cooling room T$_{COOL}$. This makes

$Q_{OUT}$ only influenced by $Q_{COOL}$, given in the second equation, resulting in a simple connection between the weather and the system. This means that the outside temperature $T_{OUT}$ is directly connected to the temperature of the condensor $T_{CONDENSOR}$, making the difference a constant value for constant $Q_{OUT}$ and $f_{FAN}$.

In order to implement the weather influence on the laboratory cooling system it is necessary to transform the temperature to the pressure of the condensor. The temperature difference between the outside temperature and the condenser temperature is set to be 15 $^oC$. This linearized transformation is calculated in Equation 4.4 from the data shown on figure 7.6 on page 60.

$$\begin{aligned} 40.7 &= x \cdot 12 + y \\ 33 &= x \cdot 8.5 + y \\ x = 2.2 \quad &and \quad y = 14.3 \\ T_{C\ OUT} &= 2.2 \cdot P_C + 14.3 \\ T_{C\ OUT} &= T_{OUT} + 15 \\ P_C &= \frac{(T_{OUT} + 15) - 14.3}{2.2} \end{aligned} \tag{4.4}$$

## 4.2 Working Range

To determine the range in which we will be able to control the temperature in the cooling room $T_{COOL}$, three tests are performed and described in detail below. The first one is for determining the operating range of the heat load given by the water heater. The second two are for determining the highest and lowest value of $T_{COOL}$.

### 4.2.1 Heat Load

The heat load in our system simulates the heat going from the surroundings into the cooling room. To determine the range of the heat load, the control signal for the heat load is increased in steps from 3 to 6 (which is the range in which change in the value gives a change in the heat load), corresponding with the minimum and maximum heat load that can be reached. The test results are plotted in Figure 4.2 on the next page and fit linear giving the following relationship between the heat load $Q_{hlc}$ and control signal $u$:

$$Q_{hlc} = 624.4 \cdot u - 267.4 \tag{4.5}$$

, giving the maximum and minimum value for the heat load: 3460 J/s and 1600 J/s.

### 4.2.2 Temperature in the Cooling Room

To determine the maximum cooling temperature $T_{COOL,max}$ that can be reached in the cooling room in steady state, the maximum evaporator pressure possible $P_{e,max}$ should be reached. This since it corresponds directly with the lowest cooling level possible. The condensor pressure is therefore set to be controlled at a maximum chosen here $P_{c,max}$= 12 bar and the compressor is running at the lowest speed possible, corresponding to a control value $u$ = 0. The heat load is at its minimum $Q_{hlc,min}$, simulating the lowest heat level going into the cooling room from the surroundings since $T_{COOL}$ is at its maximum.

The boundaries for the heat load used in the experiments are defined as: $Q_{hlc,max}$=3223 J/s and $Q_{hlc,min}$=2223 J/s. The first reason that they are different from the ones determined in section 4.2.1 on the facing page, is to give the limits some room for change in case desired later on. The second reason is that a higher limit for $Q_{hlc,min}$ and lower one for $Q_{hlc,max}$, than the ones which can be reached, results in a bigger temperature difference between $T_{COOL,min}$ and $T_{COOL,max}$, which gives a bigger working range. In other words this means that when the cooling room is at its lowest temperature, the maximum heat load going to the surroundings is lowered, resulting in a lower temperature in the cooling room which can be reached. Vice versa for the highest temperature.

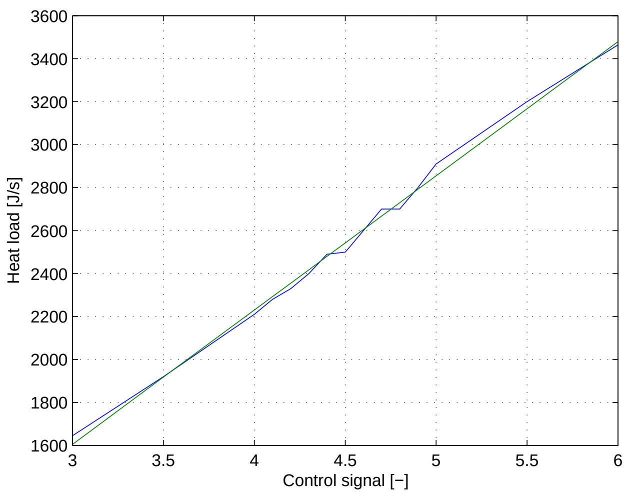


**Figure 4.2:** Relationship between control signal and heat load with linear fit

In Figure 4.3 on the next page the operating conditions for determining $T_{COOL,max}$ are given by the upper left point. Furthermore the considered working area when the system is in steady state is given, defined by $Q_{COOL} = Q_{IN}$.

The maximum $T_{COOL}$ that can be reached on the system, can be determined from the results in Figure 4.4 on page 37. The inlet temperature of the water entering the evaporator corresponding to $T_{COOL,max}$ is 13.64 °C in steady state.

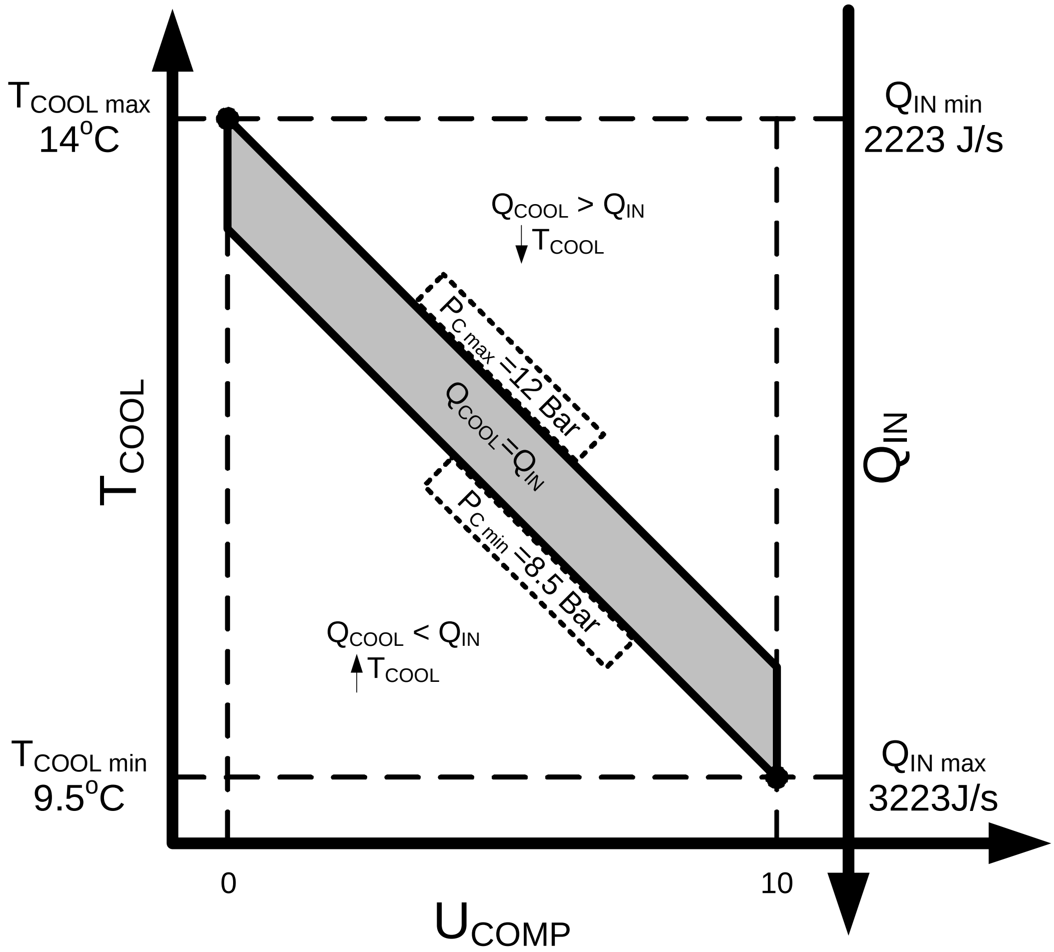


**Figure 4.3:** Working range of the temperature in the cooling room

For the determination of $T_{COOL,min}$ the condensor pressure is set to its lowest level chosen at 8.5 bar. The compressor is running at the highest speed possible, corresponding to a control value $u$ = 10. The heat load is at its maximum $Q_{hlc,max}$, simulating a maximum heat transfer from the surroundings into the cooling room, since the temperature in it is at its lowest level. In Figure 4.3 the operating conditions for finding $T_{COOL,min}$ are given. $T_{COOL,min}$ is determined from the results in Figure 4.5 on the facing page. This gives $T_{COOL,min}$ = 9.14 °C in steady state.

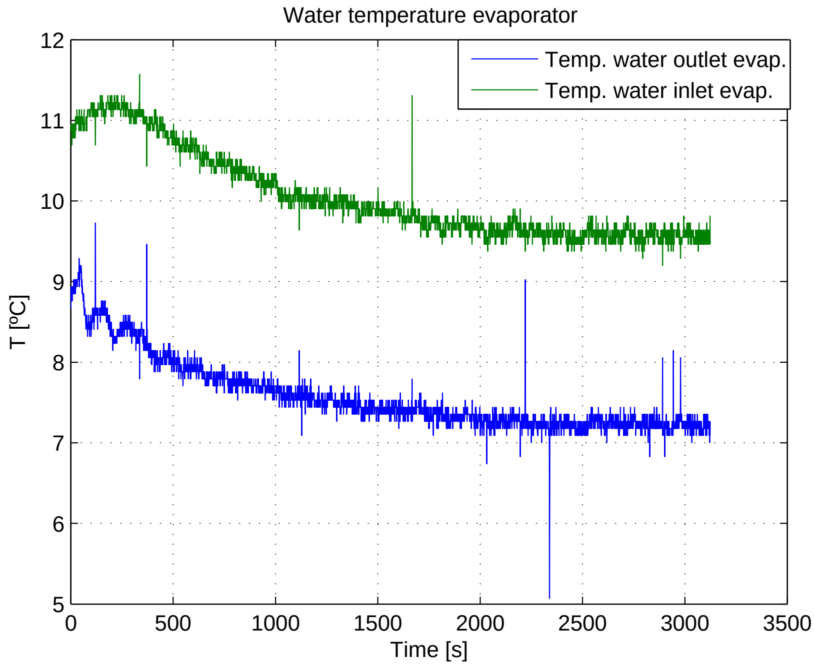


**Figure 4.4:** Inlet and outlet water temperature in the evaporator for determining $T_{cool,min}$

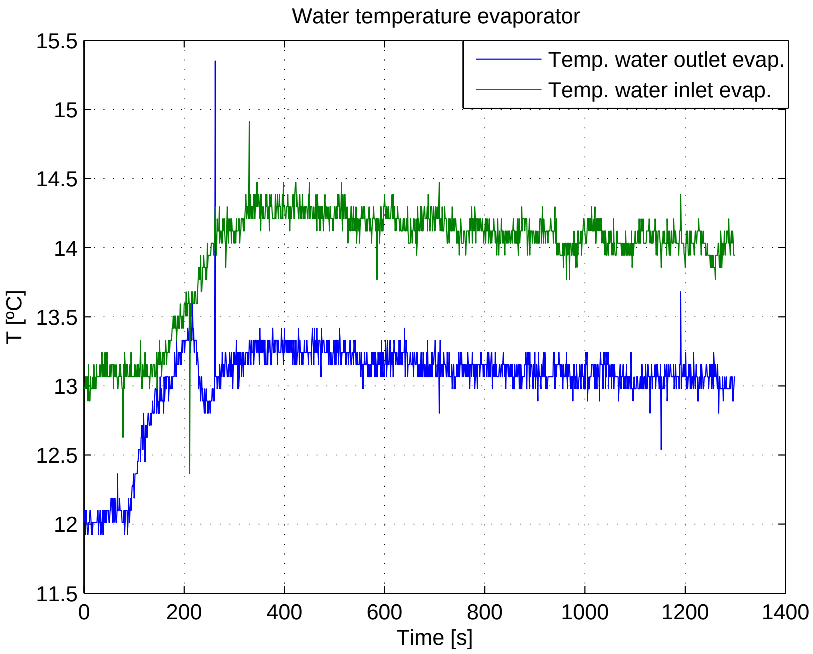


**Figure 4.5:** Inlet and outlet water temperature in the evaporator for determining $T_{cool,max}$

### 4.2.3 Operating Point

The working point chosen for controller tests in the second part is approximately in the middle of the working ranges of the following variables, giving:

- $T_{OUT}$ = 21.85°C$\Rightarrow P_C = 10.25$ Bar
- $Q_{hlc}$ = 2723 W
- $T_{COOL}$ = 11.39 °C

## 4.3 Connecting the Two Worlds

In order to get a better understanding of what the results of the test mean, this section will try to use the test results to calculate the different parameters for the equations in Section 4.1.

In the test the minimum $T_{COOL}$ was found to be 9.14 $^oC$ and the maximum $T_{COOL}$ was found to be 13.64 $^oC$. This gives a temperature difference of 4.5 $^oC$. Where the Heat load has a minimum of 2223 W and a maximum of 3223 W that gives a difference of 1000W from maximum to minimum.

Taking Equation 4.2 and putting these numbers in to it gives two equations with two unknowns. They are shown in Equation 4.6

$$\begin{aligned} 2223W &= C_{ROOM} \cdot (T_{ROOM} - 13.64^oC) \\ 3223W &= C_{ROOM} \cdot (T_{ROOM} - 9.14^oC) \\ T_{ROOM} &= 24.0035^oC \\ C_{ROOM} &= 222.\overline{2} \\ Q_{IN} &= 222.\overline{2} \cdot (24^oC - T_{COOL}) \end{aligned} \tag{4.6}$$

rewriting Equation 4.5 on page 34 gives Equation 4.7

$$\begin{aligned} Q_{IN} &= 624.6 \cdot u - 267.4 \\ u &= \frac{Q_{IN} + 267.4}{624.6} \end{aligned} \tag{4.7}$$

Figure 4.6 on the facing page illustrates in form of a block diagram, how the energy going from the working area to the cooling room is implemented. In Chapter 6 on page 47 the block ($Q_{IN}$ to u) is replaced with a close loop controller, because $Q_{IN}$ otherwise will be influenced by the voltage on the power supply net.

How does this relate to the real world? lets make an example:

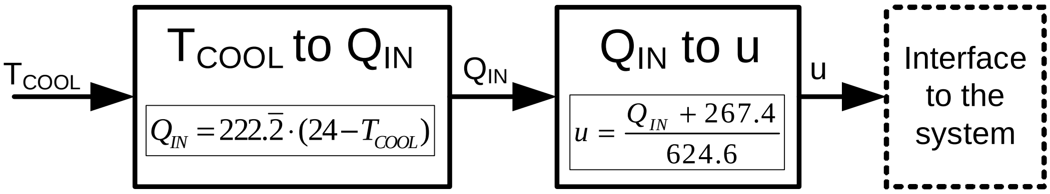


**Figure 4.6:** Block diagram of the $Q_{IN}$ implementation.

It is assumed that the real cooling room gets the same amount of energy from the surroundings as the lab. model. Minimum 2223W at the high temperature and 3223W at the lowest temperature. The difference is then the temperature in the cooling room and the temperature of the surroundings. Lets here assume that the surrounding temperature is $20^oC$ and the lowest possible temperature that the cooling system can reach is -$23^oC$. Hereby $C_{ROOM}$ is calculated to 74.95 according to Equation 4.6 on the facing page. This gives a maximum temperature at -$9.66^oC$ and a minimum temperature at -$23^oC$. If the cooling capacity for the systems are the same for both systems.

# Part II

# Model Predictive Control

# Introduction

# 5

## Contents



*This chapter will make an introduction to MPC, first a description of why MPC is chosen and how it is used here. Secondly a short introduction to MPC what it does and how it in general works.*

The outer loop controller chosen is a Model Predictive Controller (MPC). The main reason for this is the use of a predictive weather model, which predicts when the power consumption of the system has to be limited because of the allowed peak level.

For the control design a model of the cooling system that is to be controlled is required and in this case also a disturbance model, which models the weather influence. Based on prediction of the system and weather behavior, the goal of the controller is to minimize the deviation of the temperature above the reference cooling temperature and of the power consumption above the peak level.

In Figure 5.1 on the next page the cooling system with the predictive control scheme used is given. Here can be seen that the MPC controller consists of a control part, weather disturbance model and model of the cooling system. The input signals of the closed loop system are the cooling reference temperature $T_{COOL,ref}$ and a reference trajectory for the work applied to the system $W_{COMP,ref}$. The controller calculates the control signal $Q_{COOL}$. The outside temperature $T_{OUT}$ is put in as a know future trajectory, which is used by the disturbance model to predict the disturbance in the work done by the compressor. Furthermore the Model Predictive Controller requires operating constraints on the input signals and control signal. The MPC controller uses a performance function to calculate the optimal controller moves, this is described and linked to the cooling system in chapter 8 on page 65.

Before the controller can be designed and implemented, the following two transfers of the cooling system are required: $Q_{COOL} \rightarrow T_{COOL}$ and $Q_{COOL} \rightarrow W_{COMP}$, which are both considered independent of each other. For the weather disturbance model the transfer from $T_{OUT} \rightarrow W_{COMP}$ is needed. The models are derived in chapter 7 on page 53. First the evaporator pressure controller is redesigned and the condensor fan controller is linked to the weather influence, this is done in section 3 on page 27.

Before designing a MPC controller, A short introduction to what MPC is and how it works. This follows in the next section.

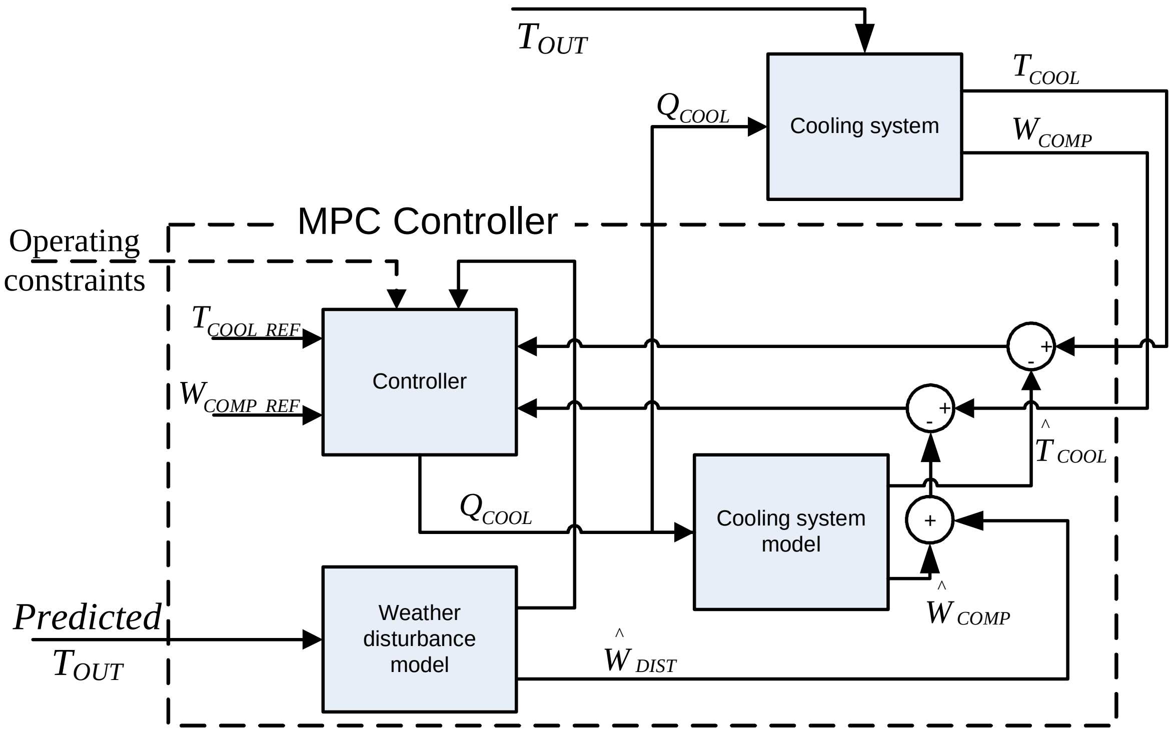


**Figure 5.1:** MPC control scheme of cooling system

## 5.1 General MPC

The benefits of MPC control compared to general PID controllers are it's ability to handle constraints on the control and output signals, there is no manual tuning of the controller needed and it's an effective means to deal with large multivariable systems. Which means no separate controller for every input-output combination has to be designed.

A general scheme of a Model Predictive Controller applied on a process can be seen in Figure 5.2 on the next page [Backx & van den Boom 2005]. The MPC controller requires a linear process model, and optional is a disturbance model for measured disturbances, which is also used here. Operating constraints on the control signal and the process output are also defined. The model predictive control system uses model based predictions of process outputs to manipulate the process inputs in such a way that deviations from specified values (setpoints) are minimized, subject to constraints on inputs and outputs. For unmeasured disturbances the controller provides feedback compensation (relatively slowly).

The tool used here to design a MPC controller is the MPC toolbox in Matlab [Bemporad et al. 2005]. An MPC Toolbox design generates a discrete-time controller, one that takes action at regularly-spaced discrete time instants. The sampling instants are the times at which the controller acts. The interval separating successive sampling instants is the sampling period $\Delta t$ (also called the control interval).

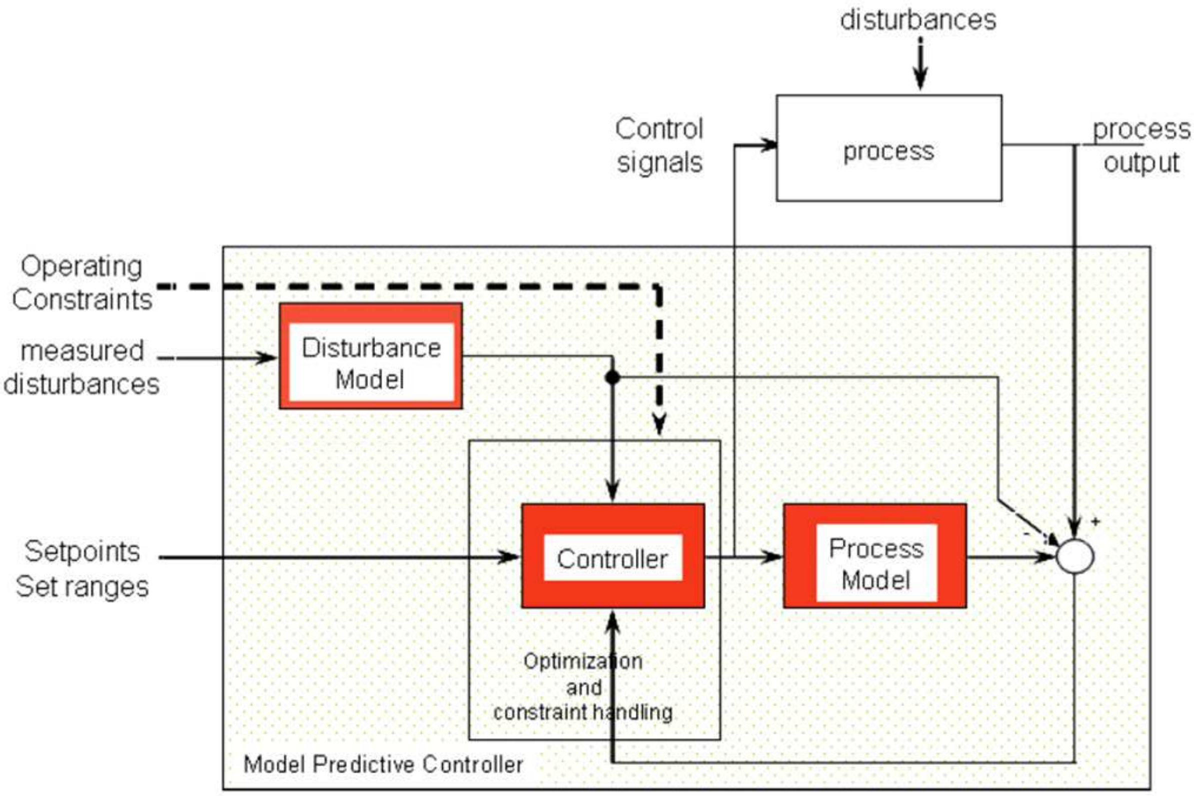


**Figure 5.2:** MPC control scheme

The latest measured output, $y_k$, and previous measurements, $y_{k-1}$, $y_{k-2}$, ..., are known and are the filled circles in Figure 5.3 on the following page(a). Figure 5.3 on the next page(b) shows the controller's previous moves, $u_{k-4}$, ..., $u_{k-1}$, as filled circles. To calculate its next move, $u_k$ the controller operates in two phases:

**Phase 1, Estimation:** In order to make an intelligent move, the controller needs to know the current state. This includes the true value of the process output ($y_k$) and any internal variables that influence the future trend, $y_{k+1}$,...,$y_{k+P}$). In which $P$ is the prediction horizon. To accomplish this, the controller uses all past and current measurements and the models of plant and measured disturbances.

**Phase 2, Optimization:** Values of setpoints, measured disturbances, and constraints are specified over a finite horizon of future sampling instants, $k+1$, $k+2$, ..., $k+P$. The controller computes $M$ moves $u_k$, $u_{k+1}$, ..., $u_{k+M-1}$, where $M$ ($\geq$ 1,$\leq$ P), see Figure 5.3 on the following page(b).

The optimization is done by minimizing a performance function, the general performance function that Matlab optimizes is: 5.1 on the next page

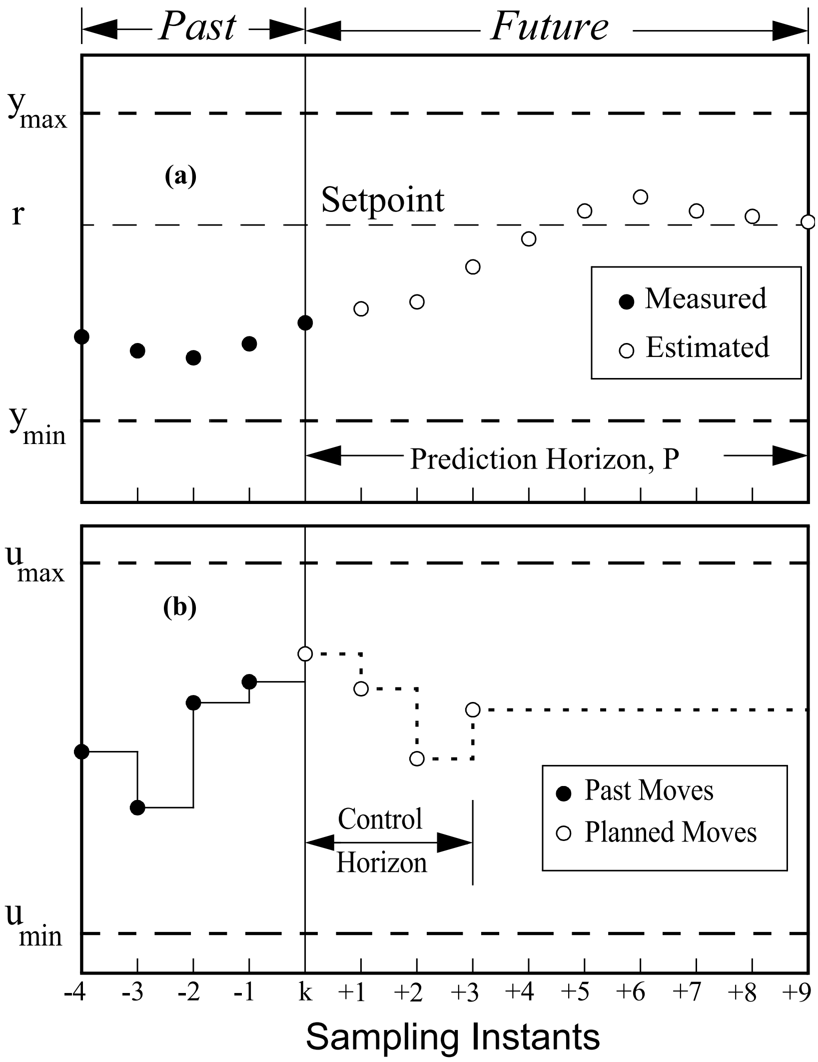


**Figure 5.3:** MPC horizon

$$J = \sum_{t=0}^{\infty}(u-u_{ref})^T\omega_u(u-u_{ref}) + (\Delta u)^T\omega_{\Delta u}(\Delta u) + (y-y_{ref})^T\omega_y(y-y_{ref}) + \beta_\xi \xi^2 \quad (5.1)$$

$u$: Is Input

$y$: Is Output

$\xi$: Is violations of the limits

# System Controller Redesign 6

## Contents



*This chapter describes how the inner loop controllers made by the 7.semester group [Kallager et al. 2005], has been redesigned and implemented on the cooling system. There are designed three controllers in this chapter, $Q_{COOL}$ controller, Fan controller with the weather disturbance implementation and a water heater controller to ensure correct heat load.*

## 6.1 $Q_{COOL}$ Controller

A way to make a more simple model of the system that must be controlled, is to redesign the Compressor controller to control the cooling that the system takes out of the cooling room. Figure 6.1 on the following page illustrates what the controller does control.

The redesign uses a PI-controller just as the old controller and for tuning the controller "Ziegler Nichols tuning method" for "open loop reaction rate" [Learncontrol 2006] is used. The PI-controller will be implemented as shown in Figure 6.6 on page 51. After implementation it is possible to fine tune the controller to improve the response in a more desired way.

Figure 6.3 on page 49 illustrates a step response of a given system. By using Equations 6.1 it is possible to calculate the three gains used in the controller.

$$\begin{aligned} K_p &= 0.9 \cdot \frac{K_1 \cdot T_2}{K_2 \cdot T_1} \\ K_i &= \frac{0.3}{T_1} \\ K_a &= 0.8 \cdot K_i \end{aligned} \tag{6.1}$$

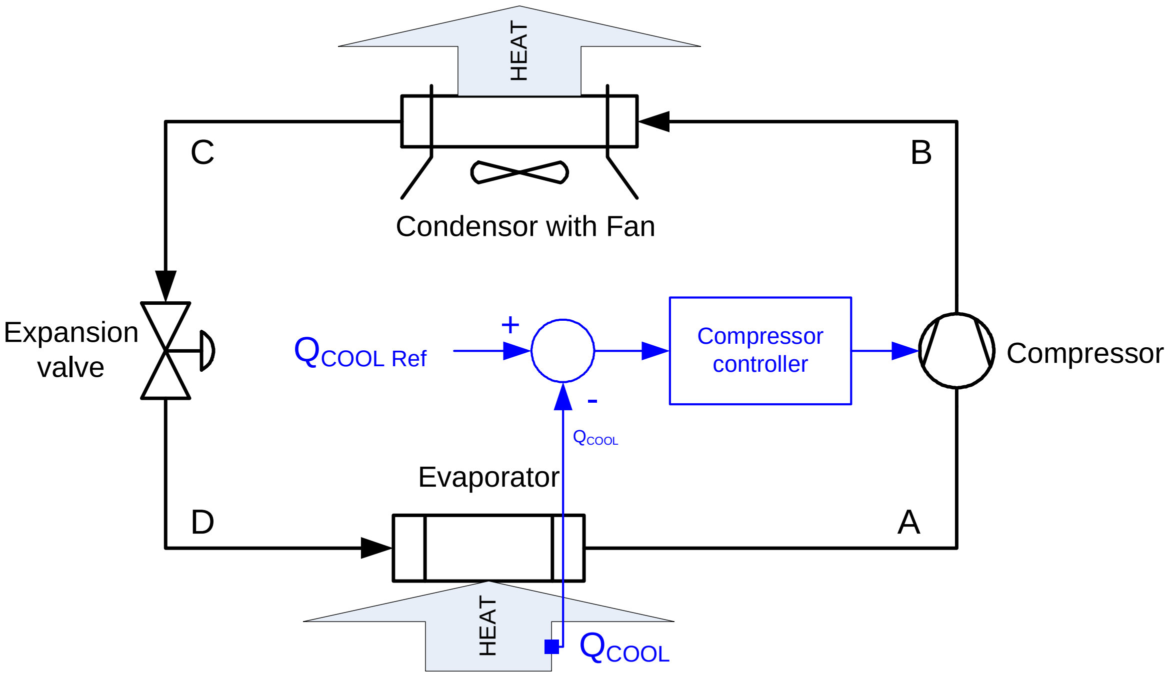


**Figure 6.1:** Fig: Illustration of the redesigned compressor controller

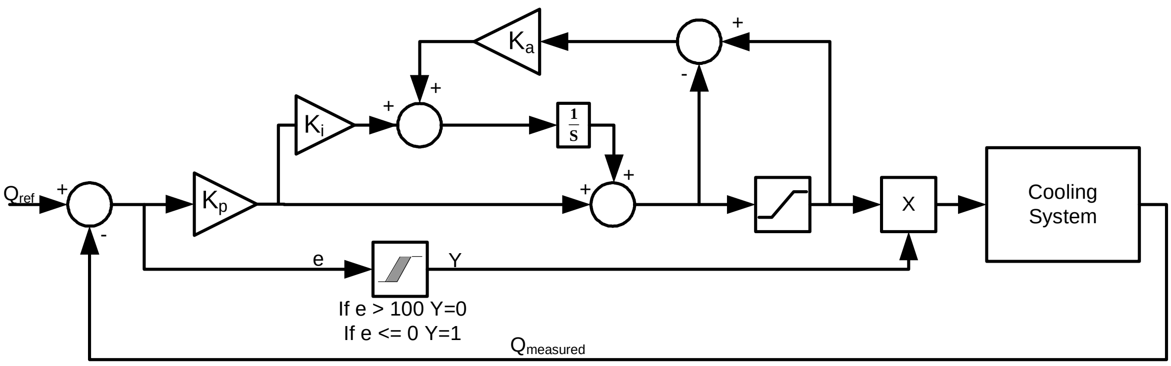


**Figure 6.2:** Fig: PI-Controller

### 6.1.1 Measuring $Q_{COOL}$

$Q_{COOL}$ is not measured directly but it can be calculated on two different ways.

- From the water in the heat load circuit on the secondary side of the evaporator.
- From the cooling fluid on the primary side of the evaporator.

To calculate the $Q_{COOL}$ from the cooling fluid it is necessary to use the change in enthalpy and the mass flow of the cooling fluid. The calculation of the enthalpy is not simple, the

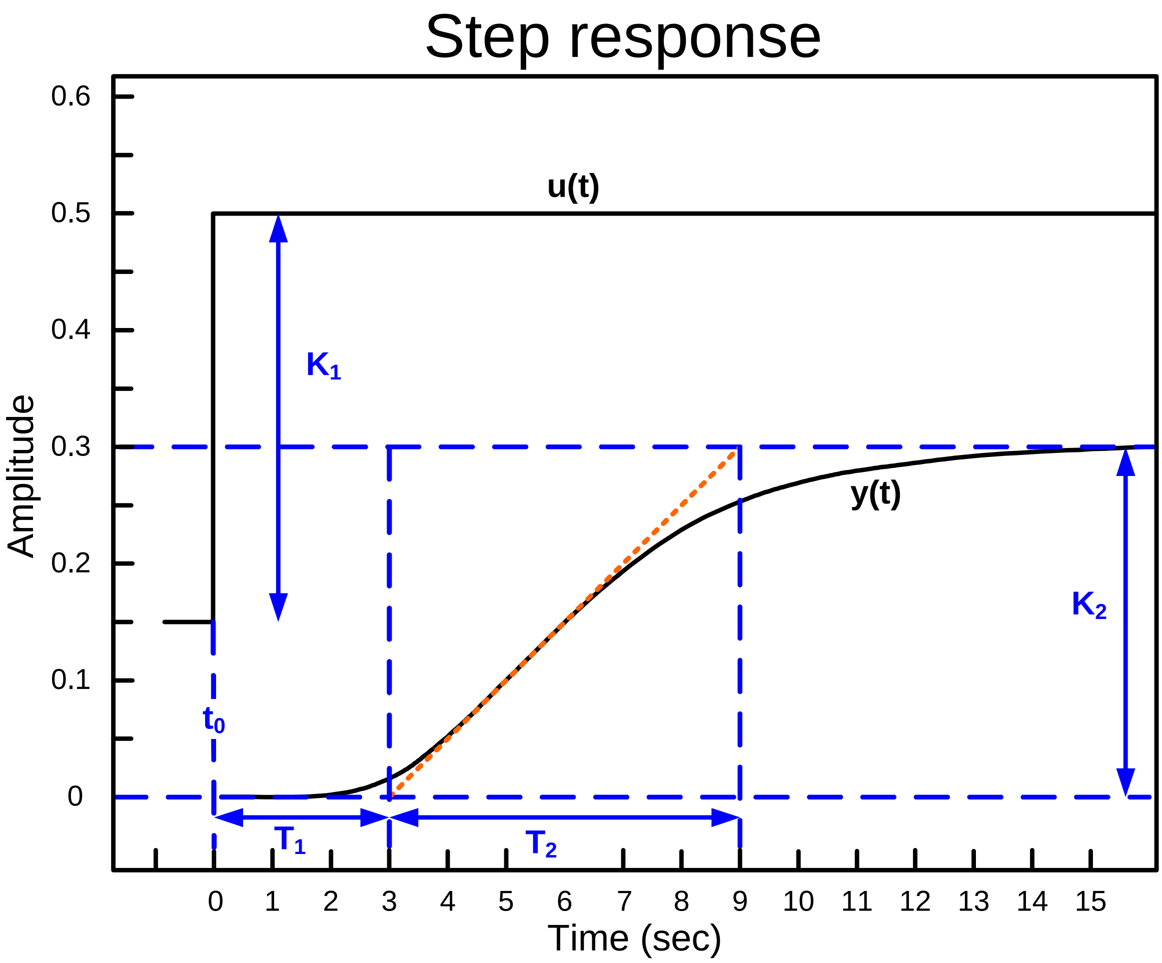


**Figure 6.3:** Step method and Ziegler-Nichols to calculate PI controller parameters

equation is highly nonlinear and has some of unknowns parameters given by the cooling fluid. Equation 6.2 shows how $Q_{COOL}$ is calculated from the primary side.

$$Q_{COOL} = \dot{m}_C \cdot (h_a - h_d) \tag{6.2}$$

The calculation of $Q_{COOL}$ calculated from water is more easy to derive. This can be calculated from the temperature difference from the inlet to the outlet of the secondary side of the evaporator ($T_{INLET} - T_{OUTLET}$), the specific heat capacity of water ($C_W$) and the mass flow of the water ($\dot{M}_W$). Equation 6.3 shows how $Q_{COOL}$ is calculated from the secondary side.

$$Q_{COOL} = C_W \cdot \dot{M}_W \cdot (T_{INLET} - T_{OUTLET}) \tag{6.3}$$

### 6.1.2 Tuning the PI-Controller

In the tuning procedure an open loop test is needed to calculate the first values for the controller and after this a fine tuning of the parameters is possible. The procedure is to apply a step to the input of the system when it is in steady-state and measure the output. Figure 6.4 shows a block diagram of the black box system with a single control input and a single measurement output.

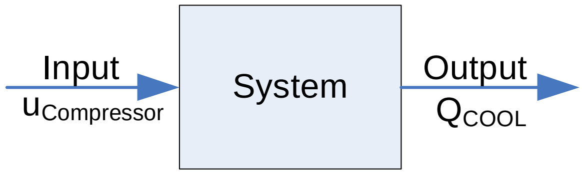


**Figure 6.4:** Fig: Block diagram of the test setup

The result from the step response test is shown in Figure 6.5. The lines and numbers used to calculate the first parameters to the PI-controller are included in the figure.

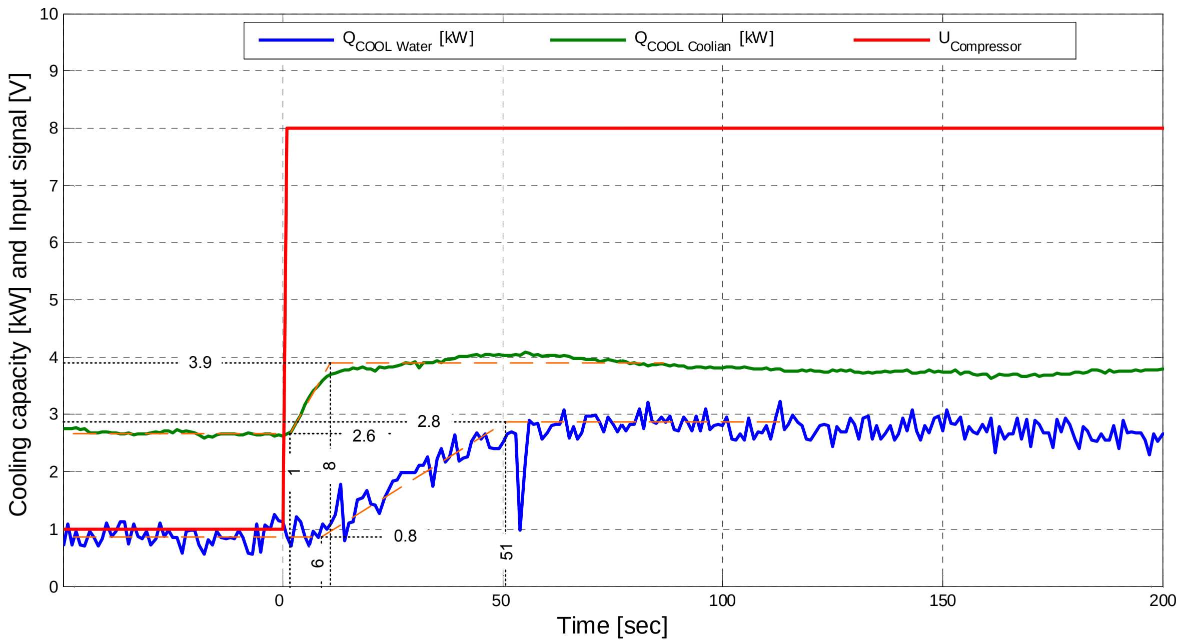


**Figure 6.5:** Fig: Graph of the step response test

Table 6.1 shows the calculated $K_p$, $K_i$ and $K_a$ only the controller on Coolant needed to be tuned what are the numbers after the arrows in the Table.

The designed controllers are implemented on the system and closed loop step tests are run to see how they perform. Figure 6.7 on the facing page shows a graph of these results of both controllers. It is chosen to use the controller using the cooling fluid to calculate the cooling capacity, $(Q_COOL)$, because the controller using the water has a longer settling time and it

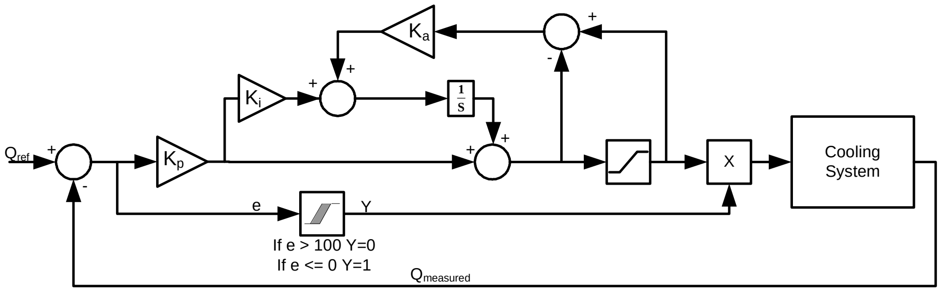


**Figure 6.6:** Fig: PI-Controller

| Water | Cooling fluid |
|---|---|
| $K_p = 0.0054$ | $K_p = 0.0339 \rightarrow 0.007$ |
| $K_i = \frac{1}{46.66}$ | $K_i = 0.3 \rightarrow 0.15$ |
| $K_a = 0.8 \cdot K_i$ | $K_a = 0.8 \cdot K_i$ |

**Table 6.1:** The calculated gains

has a bigger error when it is settled.

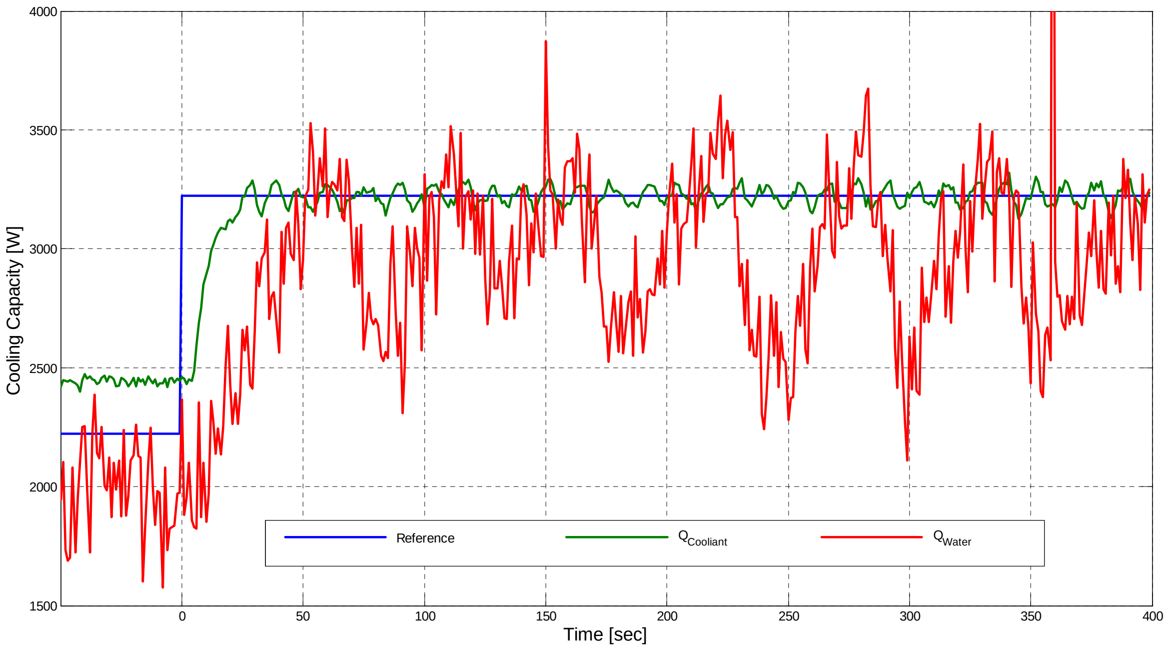


**Figure 6.7:** Fig: Step response of the calculated PI-Controller

## 6.2 Fan-controller and Weather connection

For the system used in the predictive model a direct connection between the outside temperature $T_{OUT}$ and $P_C$ is required, in which $P_C$ is controlled with the existing fan controller given in section 3.2 on page 28. As derived in section 4.1.1 on page 33, equation 4.4 gives the influence of the weather on the system. In the lab setup however the fanspeed $f_{FAN}$ will be changing to simulate the change in weather, however the difference between $T_{C,OUT}$ and $T_{OUT}$ will be considered constant.

Figure 6.8 shows a block diagram of how the outside temperature ($T_{OUT}$) is used to generate the reference pressure to the existing PI-controller that controls the pressure $P_C$ with the speed of the fan.

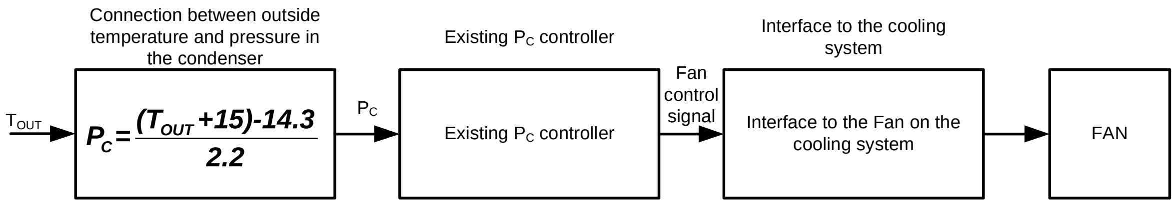


**Figure 6.8:** Block diagram of the extended fan controller.

# System models

# 7

## Contents



*This chapter describes all the model needed to implement the Model Predictive controller. First models related to the power consumption is derived, that being $Q_{COOL}$ to $W_{COMP}$ and the disturbance model $T_{OUT}$ to $W_{COMP}$ both model fitted using system identification. After this the model from $Q_{COOL}$ to $T_{COOL}$ is derived by using energy relations. Finally the system model is put on state space representation to make the implementation easier.*

## 7.1 Q to W and Weather Disturbance Model

For the model from the heatload to the work done by the compressor and the weather disturbance on this work, frequency response models in state space format are required for implementation in the MPC toolbox. To derive these models a system identification technique is used, for which the order of the system is required. An approximation of the order of the system is described here and afterwards the system identification results.

### 7.1.1 Q to W

To estimate the order of the transfer from $Q$ to $W$, equation 2.7 on page 18:

$W_{AB} = (P_c - P_e)V_{com}f_{com}$ is used, which gives the work done in the compression step. Assuming no weather influence, $P_c$ is considered constant. The evaporator pressure $P_e$ is assumed a linear function of the temperature $T_e$, which linearly depends on the heatload $Q_{hlc}$, making $P_e$ and $Q_{hlc}$ linearly dependent. The compressor frequency $f_{com}$ is also taken to be a linear function of $Q_{hlc}$. This gives the following relationship between the work and heatload, the last called $Q$ in this section:

$$W \sim Q^2 \tag{7.1}$$

This relationship holds for steady state, extending it to a dynamical model gives:

$$\frac{dW}{dt} \sim \frac{d(Q^2)}{dt} \tag{7.2}$$

This gives a non-linear system, therefore a linearization in point Q = 2723 J/s is chosen, which is in the middle of the operating range for the heat. This gives a first-order system to be modeled in this operating point. A frequency response model is required, since it is expected that this system will have a variety of frequencies as an input. In the frequency domain the system is given by:

$$\frac{W(s)}{Q(s)} = \frac{a \cdot s + b}{c \cdot s + d} \tag{7.3}$$

, in which a, b, c and d are the parameters to be determined by the system identification algorithm used.

### 7.1.2 Weather disturbance on W

For the influence of the weather on the work done by the compressor, the steady state equation 2.7 on page 18 is used again to approximate the order of the dynamical model of the outside temperature $T_{OUT}$ to the work $W$, giving the weather disturbance model.

A constant $T_{COOL}$ and therefore also $P_e$ is considered. The change in $P_c$ is taken to be a linear function of $Tout$. The compressor frequency $f_{com}$ is considered to be independent on $Tout$. This gives the following relationship:

$$W \sim T_{OUT} \tag{7.4}$$

extending this to the dynamical model required gives:

$$\frac{dW}{dt} \sim \frac{d(T_{OUT})}{dt} \tag{7.5}$$

This results in a first order model, similar to the structure of the model in equation 7.3.

### 7.1.3 System identification

To determine the parameters in the first frequency response model from $Q$ to $W$ given above of the form given in equation 7.3, the Matlab program Senstools [Knudsen 2004] is used. It requires the system model as an input, starting values of the parameters in the model and an input and output of the system measured in time domain.

To perform the system identification a working point is chosen according to section 4.2.3 on page 38.

For measuring the influence of $Q$ on $W$ the heat input signal is a step function sequence with Q going from 2723 J/s to 3223 J/s, the period is 1800 s. The reason for choosing a step function is that the resulting transfer calculated by Senstools will be valid for a wide variety of frequencies, which is required since the control input signal is expected to contain various frequency components. The range for the step input signal is chosen such that the controller for $Q$ can follow the signal fast enough. After measuring the work and fitting the data it gives the following transfer:

$$\frac{W(s)}{Q(s)} = \frac{1.11}{s+3.00} \tag{7.6}$$

, with a least squared error of the fit of 12.7 %. The bode diagram of the fit is given in Figure 7.2 on the next page.

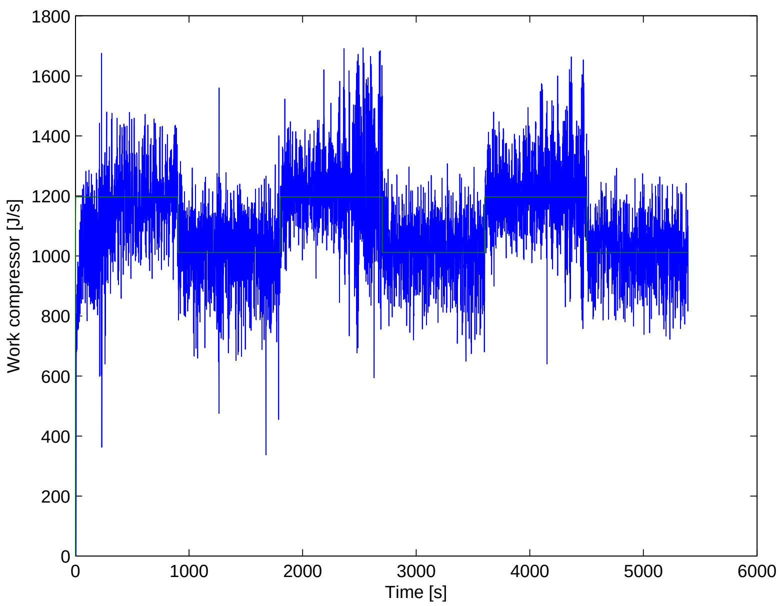


**Figure 7.1:** Q to W: simulated system identification results vs. measured experimental results

Comparing the output of the simulated system with the output of the work for the real system, with the same input signal as used for the system identification, gives the results given in Figure 7.1.

The model of the outside temperature $T_{OUT}$ to the disturbance on the work $W$ is determined by measuring the work of the compressor with the input for the condensor temperature $T_C$ being a sine wave. The last has a mean of 36.85 $^oC$, amplitude of 3.85 $^oC$ and a period of 1800 s. The range of the temperature of $T_C$ is calculated with equation 4.4 on page 34, by using the operating range of $P_C$ from 8.5-12 bar. The reason for taking this sinewave as an input

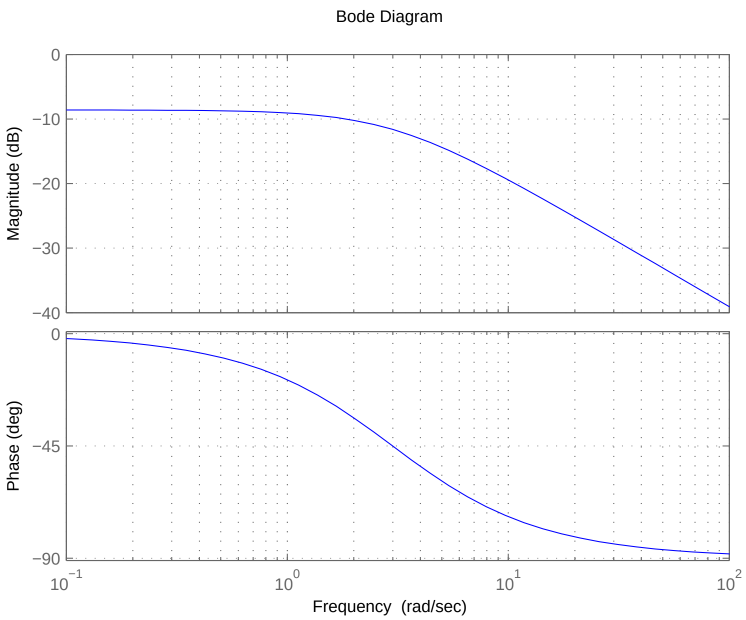


**Figure 7.2:** Bode diagram of the transfer from Q to W

signal is that the required transfer from $T_{OUT}$ to $W$ is only required for this input signal, since the $T_{OUT}$ will be considered constant. The outside temperature $T_{OUT}$ will be taken 15 $^oC$ lower then $T_C$, giving a realistic range for the real world which will be implemented as temperature reference in the MPC controller.

The model derived, which is thus fitted for the implemented disturbance with a frequency of $\frac{1}{1800}$ Hz is:

$$\frac{3.55s + 3.55}{s + 0.08} \tag{7.7}$$

The bode diagram of this fit is given in Figure 7.3 on the facing page. The least squares error of the fit is 13

Comparing again the output of the simulated system with the output of the work for the real system, caused by the given disturbance on the outside temperature, gives the results given in Figure 7.4 on the next page. An offset on the input of the disturbance has to be used when the system is implemented though, since the mean of the disturbance should correspond with a $W_{COMP}$ = 0. Therefore 21.85 °C is subtracted at the input. The necessary correction is given in Figure 7.5 on page 58.

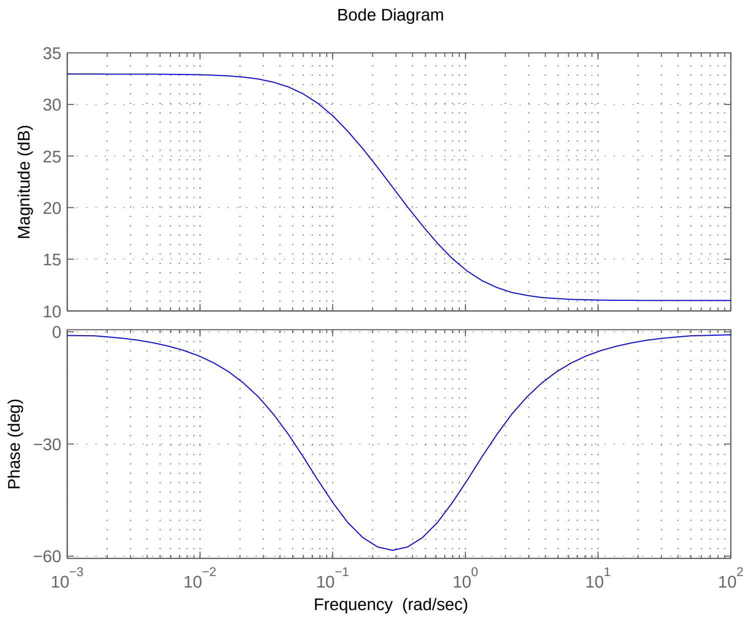


**Figure 7.3:** Bode diagram of equation 7.7 on the facing page

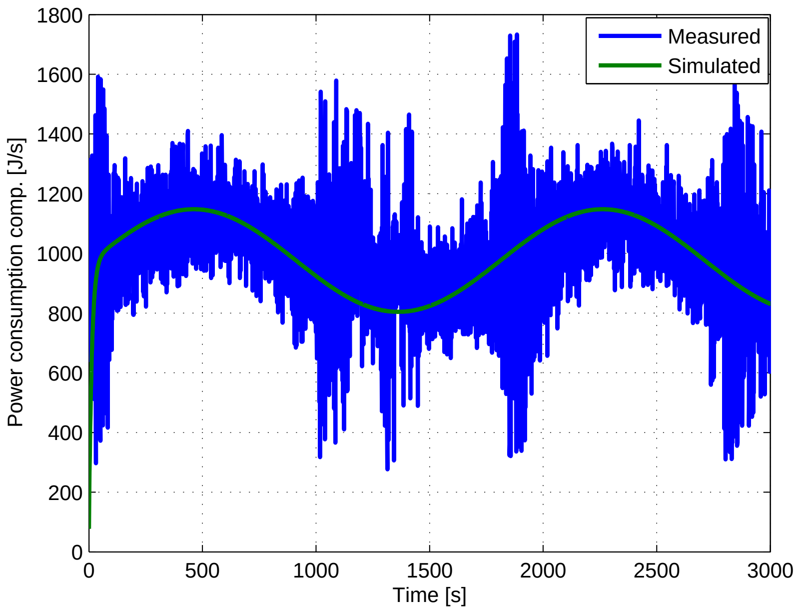


**Figure 7.4:** $T_{OUT}$ to W: simulated system identification results vs. measured experimental results

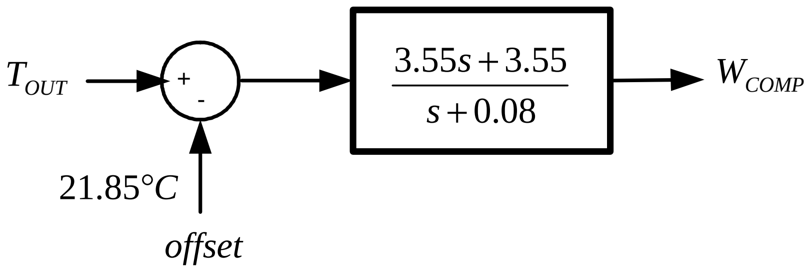


**Figure 7.5:** Diagram of the calculated transfer by Senstools for weather influence on work done by the compressor, with an offset on the input needed for the correct implementation

## 7.2 $Q_{COOL}$ to T$_{COOL}$

The model which is determined in this section is a model going from a steady state cooling capacity of the system to the temperature of the cooling room.

The steady state temperature is given when the energy in to the system is the same as the the energy taken out. The energy put in to the system is given by Equation 7.8 as mentioned in the objectives.

$$\begin{aligned} Q_{IN} &= C_{ROOM} \cdot (T_{ROOM} - T_{COOL}) \\ Q_{IN} &= 222.\overline{2} \cdot (24^oC - T_{COOL}) \end{aligned} \tag{7.8}$$

Q$_{COOL}$ is the energy taken out by the cooling system, which in steady state is equal to the energy Q$_{IN}$. The calculation of Q$_{COOL}$ is shown in Equation 7.9. The $h_A$ and $h_D$ are linearized in the working range of the cooling system by plotting test results in to the pressure-enthalpy diagram and from that finding a linear connection between T$_{CONDENSOR\ OUT}$ and enthalpy and the same was done for finding the connection between the T$_{EVAPORATOR\ OUT}$ and enthalpy. The only difference is that for this last calculation the pressure does influence the enthalpy, which is not the case in the liquid area.

$$\begin{aligned} Q_{COOL} &= m_{\dot{C}OOL} \cdot (h_A - h_D) \\ h_A &= \frac{407-400.5}{10-2} \cdot T_{E\ OUT} + \frac{407-408}{3-2.5} \cdot P_E + 407 - 2.125 \\ &= 0.8125 \cdot T_{E\ OUT} - 2 \cdot P_E + 404.875 \\ h_D &= \frac{257-246}{40.7-33} \cdot T_{C\ OUT} + 246 - 47.14 \\ &= 1.429 \cdot T_{C\ OUT} + 198.86 \end{aligned} \tag{7.9}$$

The Equations for h$_A$ and h$_D$ are calculated from the minimum and maximum values of the working area, drawn in Figure 7.6 on the next page

The dynamics of T$_{COOL}$ in the laboratory cooling system depends on the heat load circuit, as this is used to simulate a real cooling room. This means that the dynamical model depends on the mass of the water ($m_W$) in the heat load circuit, the specific heat capacity ($C_W$) of the water in the heat load circuit and the energy taken out or put into the heat load circuit. This is written in Equation 7.10

$$\begin{aligned} Q_{IN} - Q_{COOL} &= C_W \cdot m_W \cdot \frac{dT_{COOL}}{dt} \\ \frac{dT_{COOL}}{dt} &= \frac{Q_{IN} - Q_{COOL}}{C_W \cdot m_W} \end{aligned} \tag{7.10}$$

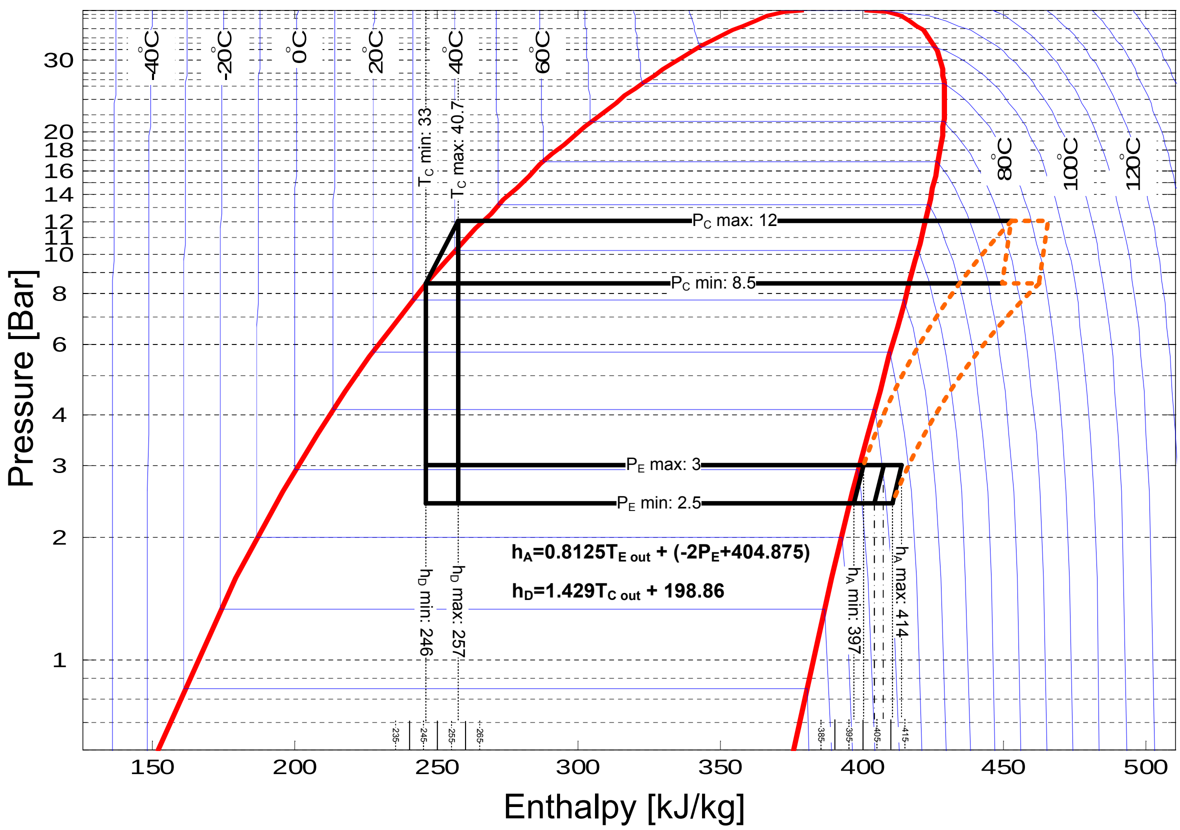


**Figure 7.6:** Illustration of the working range of the cooling system

To derive the state space model from $Q_{COOL}$ to $T_{COOl}$ equation 4.6 on page 38 is used to substitute the following in 7.10 on the preceding page:

$$Q_{IN} = 222.22 \cdot T_{COOL} + 5333 \tag{7.11}$$

In the frequency domain this gives:

$$\frac{T_{COOL}}{Q_{COOL} + 5333} = \frac{\frac{-1}{C_w m_w}}{s + \frac{222.22}{C_w m_w}} \tag{7.12}$$

The bode diagram of this transfer is given in Figure 7.7 on the next page.

Since the model has a offset in the input, an offset in the output when implementing the model of 23.68 °C is acquired to not change the temperature range of the working area defined in section 4.2.2 on page 35, see Figure 7.8 on the next page.

For measuring the influence of $Q_{COOL}$ on $T_{COOL}$ the heat input used is a sinus function with a mean of Q = 2723 J/s, an amplitude of 500 J/s and a period of 1800 s. The difference between the model and the real system is given in Figure 7.9 on page 62.

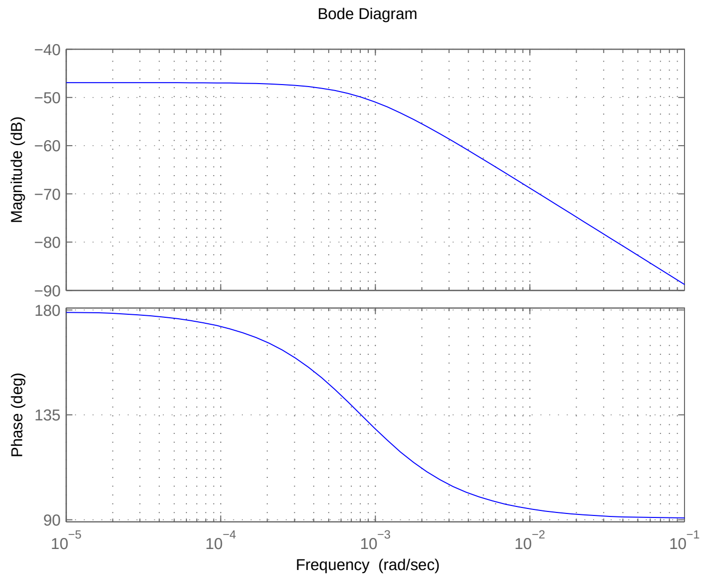


**Figure 7.7:** Bodeplot of the transfer given in equation 7.12 on the preceding page of $Q_{COOL}$ to $T_{COOL}$

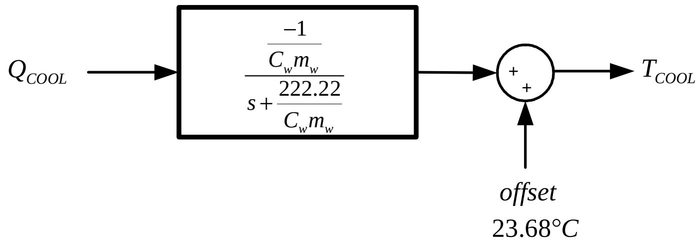


**Figure 7.8:** Diagram of the transfer function from cooling to temperature in the cooling room

### 7.2.1 The Mass of the Water

$C_W$ is for water 4186 J/kg but the water in the heat load system has been given an anti-freeze lubricant. The anti-freeze lubricant is Ethylene Glycol where 30% of the water has been replaced by this. In [The-Engineering-ToolBox 2005] the scaling factor for a 30% Ethylene Glycol mixture is 0.89 Equation 7.13 calculates the $C_w$ for this mixture.

$$C_W = 4186[J/kg] \cdot 0.89 = 3725.5[J/kg] \tag{7.13}$$

The mass of the water has been determined by making a test where the energy put into the tank is measured to 1705.1 W, the cooling system is switched off and because the temperature $T_{COOL}$ is near the room temperature of the surroundings no significant energy is transferred between the surroundings and the heat load circuit. From this it was found that

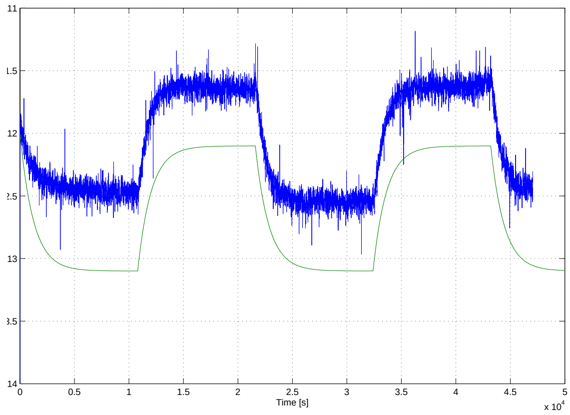


**Figure 7.9:** $Q_{COOL}$ to $T_{COOL}$: Simulated results of system model vs. measured experimental results

the slope of T$_{COOL}$ is 6.2068m $^{o}C/$s. Equation 7.14 shows how the mass of the water in the heat load circuit is calculated.

$$\begin{aligned} Q_{IN} &= c_W \cdot m_W \cdot \Delta T \\ 1705.1 &= 3725.5 \cdot m_W \cdot 0.006068 \\ m_W &= 73.7kg \end{aligned} \tag{7.14}$$

## 7.3 From transfer function to state space model

To implement the system in the MPC toolbox the transfer functions of the systems from the previous sections are rewritten in the general state space form:

$$\begin{aligned} \dot{x}(t) &= Ax(t) + Bu(t) \\ y(t) &= Cx(t) + Du(t) \end{aligned} \tag{7.15}$$

To go from time domain to Laplace domain the equations are rewritten:

$$\begin{aligned} sX(s) - x(0) &= AX(s) + BU(s) \\ Y(s) &= CX(s) + DU(s) \end{aligned} \tag{7.16}$$

,in which $x(0)$ is the initial state of the system. From 7.16 the following relationship between input and output can be derived:

$$Y(s) - \frac{Cx(0)}{sI - A} = \frac{Ds + CB - DA}{sI - A} \cdot U(s) \tag{7.17}$$

If the initial state is considered 0, the ratio $G(s)$ from input to output is:

$$G(s) = \frac{Ds + CB - DA}{sI - A} \tag{7.18}$$

Therefore the state space models in the form of 7.15 on the facing page derived from equation 7.18 need to have an offset on the output to correct it to the real value in case the initial state is non-zero. This is however not the case for the previous derived models since the initial values are 0.

The combined state space matrices derived are:

$$\begin{aligned}
\begin{pmatrix} \dot{T}_{COOL} \\ W_{COMP} \end{pmatrix} &= \begin{pmatrix} -0.809m & 0 \\ 0 & -3.084 \end{pmatrix} \cdot \begin{pmatrix} T_{COOL} \\ W_{COMP} \end{pmatrix} + \begin{pmatrix} -3.642\mu \\ 1.178 \end{pmatrix} \cdot Q_{COOL} \\
\begin{pmatrix} T_{COOL} \\ W_{COMP} \end{pmatrix} &= \begin{pmatrix} 1 & 0 \\ 0 & 1 \end{pmatrix} \cdot \begin{pmatrix} T_{COOL} \\ W_{COMP} \end{pmatrix} + \begin{pmatrix} 0 \\ 0 \end{pmatrix} \cdot Q_{COOL}
\end{aligned} \tag{7.19}$$

$$\begin{aligned}
\begin{pmatrix} \dot{W}_{COMP} \\ \psi \end{pmatrix} &= \begin{pmatrix} 0 & 0 \\ 0 & -0.079 \end{pmatrix} \cdot \begin{pmatrix} W_{COMP} \\ \psi \end{pmatrix} + \begin{pmatrix} 0 \\ 2 \end{pmatrix} \cdot T_{OUT} \\
\begin{pmatrix} T_{COOL} \\ W_{COMP} \end{pmatrix} &= \begin{pmatrix} 0 & 0 \\ 0 & 1.634 \end{pmatrix} \cdot \begin{pmatrix} W_{COMP} \\ \psi \end{pmatrix} + \begin{pmatrix} 0 \\ 3.551 \end{pmatrix} \cdot T_{OUT}
\end{aligned} \tag{7.20}$$

The first state space model gives the behavior of the system without any disturbance influence and the second model gives the disturbance influence on the system. There is no influence of the disturbance on the temperature, but this output is included in the state space model to add the outputs more easy later on.

After deriving these state space models the controllability matrix is calculated in Matlab and checked for full rank, guaranteeing the system is controllable.

# Optimization 8

## Contents



*This chapter will explain how MPC is using a performance function in the optimization just as is done in optimal control. First a explanation of what needs to be optimized is described and a connection between this an what in possible to optimize using the MPC-toolbox is made. This makes it possible to estimate the weights in the performance function and thereby in the toolbox.*

## 8.1 Performance Function

The main goal of the optimization is to get the system to perform according to the goals set in chapter 4 on page 31, as "cheap" as possible. In Optimal Control where a cost function is used, the optimal controller which is designed minimizes the total cost of the individually weighted system signals. The optimal controller however does not have the ability to predict the future behavior of a system. It is here the predictive controller has its advantages. To be able to to make a Model Predictive Controller (MPC), the "cost" of the input and outputs must be evaluated. This makes it easier to determine the what the difference weights must be in the performance function that must be minimized by the controller.

### 8.1.1 Cost Of Input And Outputs

The cost of the input and outputs are chosen from price on electricity and how important the output is.

**Input:** $Q_{COOL}$ is not weighted, because it does not directly cost anything that this value is high or low, but it does effect the temperature $T_{COOL}$.

**Output:** $T_{COOL}$ is not directly weighted as long as it is kept inside the limitation setup for the temperature. The limit must newer be broken and thereby this is weighted vary high.

**Output:** $W_{COMP}$ is to be weighted because this is what you pay for when you buy electricity. It will be natural to put the weight to a number near the price of the electricity. The limitation of $W_{COMP}$ must never be broken and thereby this is weighted vary high.

This setup will in situations where there are not limitation on the temperature the system will shutdown to avoid using any power because power cost money. Because there are limitations to the temperature and the power consumption the system will always try to go to the maximum temperature, because this uses the least amount of power. But it has to use prediction to be sure not to violate the two constrains, one on power consumption and one on the temperature.

In a general performance function the signals are written as squared deviations from the reference. To rewrite the equation as a squared function, the reference values for Q, W and T are needed. The reference for $Q_{COOL}$ is chosen to be zero and because $Q_{COOL}$ is not weighted it does not matter what the reference value is. The reference value for $W_{COMP}$ is set to to zero because the more power that the system uses the more it cost. The reference for $T_{COOL}$ is also set to zero, with the same reasons as for $Q_{COOL}$. This makes the deviations equal to the measurements as shown in Equation 8.1.

$$\begin{aligned}
e_Q &= Q_{COOL} - Q_{COOL\ REF} = Q_{COOL} \\
e_W &= W_{COMP} - W_{COMP\ REF} = W_{COMP} \\
e_T &= T_{COOL} - T_{COOL\ REF} = T_{COOL}
\end{aligned}$$

Equation 8.1 shows the squared performance function. It must be noticed that $F_{T_{COOL}}$ and $F_{W_{COMP}}$ if functions given zero when the measurement is within the limitations and outside the limits it gives the value of how much it has broken the limit.

$$\begin{aligned}
J &= \sum_{t=0}^{\infty} \alpha_1 e_Q^2 + \alpha_2 e_W^2 + \alpha_3 (F_{T_{COOL}})^2 + \alpha_4 (F_{W_{COMP}})^2 \qquad (8.1) \\
F_{T_{COOL}} &= \begin{cases} T_{COOL\ MIN} - T_{COOL} & if\ T_{COOL} < T_{COOL\ MIN} \\ T_{COOL} - T_{COOL\ MAX} & if\ T_{COOL} > T_{COOL\ MAX} \\ 0 & if\ T_{COOL\ MIN} \leqq T_{COOL} \leqq T_{COOL\ MAX} \end{cases} \\
F_{W_{COMP}} &= \begin{cases} W_{COMP\ MIN} - W_{COMP} & if\ W_{COMP} < W_{COMP\ MIN} \\ W_{COMP} - W_{COMP\ MAX} & if\ W_{COMP} > W_{COMP\ MAX} \\ 0 & if\ W_{COMP\ MIN} \leqq W_{COMP} \leqq W_{COMP\ MAX} \end{cases}
\end{aligned}$$

Comparing this performance function to the performance function given in [Bemporad et al. 2005] that are shown in Equation 8.2

$$J = \sum_{t=0}^{\infty} \omega_{e_u} e_u^2 + \omega_{\Delta u} \Delta u^2 + \omega_{e_y} e_y^2 + \beta_\xi \xi^2 \qquad (8.2)$$

By comparing these two equations it can be seen that there are a lot of similarities. $\xi$ is the violation of the constrains on the inputs and the outputs, and can be divided in to four separate $\xi$ one for the input, one for the change en the input and one for each output. If the inputs and the outputs are defined as shown in Equation 8.3

$$\begin{aligned} u &= Q_{COOL} \\ y &= \begin{pmatrix} T_{COOL} \\ W_{COMP} \end{pmatrix} \end{aligned} \tag{8.3}$$

By rewriting the performance to use matrixes and use the notation for the cooling system the performance function will be as shown in Equation 8.4

$$\begin{aligned} J = \sum_{t=0}^{\infty} \quad & \alpha_1 \, e_Q^2 + \omega_{\Delta Q} \, \Delta Q^2 + \begin{pmatrix} T \\ W \end{pmatrix}^T \omega_{e_y} \begin{pmatrix} T \\ W \end{pmatrix} \\ & + \beta_Q \, \xi_Q^2 + \beta_{\Delta Q} \, \xi_{\Delta Q}^2 + \alpha_3 \, \xi_T^2 + \alpha_4 \, \xi_W^2 \end{aligned} \tag{8.4}$$

This means that the weight $\alpha_1$, $\omega_{\Delta Q}$, $\beta_Q$, $\beta_{\Delta Q}$ is set to Zero and the weight $\omega_{e_y}$ is $\begin{pmatrix} 0 & 0 \\ 0 & \alpha_2 \end{pmatrix}$. This will give a more simple the parts in the equation that are weighted zero can be taken out. This is shown in Equation 8.5

$$J = \sum_{t=0}^{\infty} \quad \begin{pmatrix} T \\ W \end{pmatrix}^T \omega_{e_y} \begin{pmatrix} T \\ W \end{pmatrix} + \alpha_3 \, \xi_T^2 + \alpha_4 \, \xi_W^2 \tag{8.5}$$

The following will try to explain what the different weights will do in the optimization.

$\alpha_1$ **High:** The controller will minimize the cooling capacity of the system.

$\omega_{\Delta Q}$ **High:** The controller will not change the cooling capacity of the system.

$\omega_{e_{y22}} \Leftrightarrow \alpha_2$ **High:** The controller will minimize the power consumption of the system, meaning that it will cool more in the cold period,night time, than when it is hot because when it is hot the power consumption is higher for a constant Q$_{COOL}$ when it is hot.

$\alpha_3$ **High:** The system will not go outside the limitation set for the temperature, meaning the temperature will be below this upper boundary for the temperature.

$\omega_{e_{y11}}$ **High:** The temperature will not deviate from the reference.

$\beta_Q$ **High:** The controller will not allow that the $Q_{COOL}$ to be bigger than the limit for $Q_{COOL}$.

$\beta_{\Delta Q}$ **High:** The controller will not allow changes of $Q_{COOL}$ to be bigger than the limit for this.

$\beta_W$ **High:** The controller will not allow the power consumption to be bigger than the limit.

Understanding how the weights influence the controller, is essential when the controller has to be tuned.

# Implementation 9

## Contents



*This chapter describes how the system and MPC-toolbox is setup in the simulations and implemented in the laboratory as this is the same. Each of the different "windows" in the MPC-toolbox is described one by one and an explanation is given to the individual choices is given.*

The implementation of the controller must be made in SIMULINK as the rest of the system is implemented in SIMULINK as well. SIMULINK has the possibility to use the MPC-toolbox and will be used here. In this chapter the way to use this toolbox is explained. Figure 9.1 shows how the MPC-toolbox is connected to the cooling system.

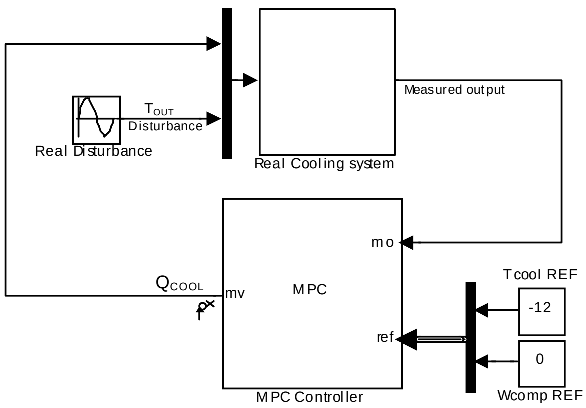


**Figure 9.1:** Connecting the MPC-toolbox to the real cooling system.

When designing a MPC controller you have to include the system model and the disturbance model into the MPC-toolbox. The model can be given as a state space model where the disturbance model is a part of the system model. Equation 9.1 on the following page combines the two state space models, given in section 7.3 on page 62, to one state space model. The matrix elements without a subscript $d$ are from the first system state space model in section 7.3 on page 62 and with subscript $d$ are from the the second state space model giving the disturbance influence on the system.

$$\begin{aligned}
\underbrace{\begin{bmatrix}\dot{T}_{COOL}\\ \dot{W}_{COMP}\\ \dot{\psi}\end{bmatrix}}_{\dot{X}3x1} &= \underbrace{\begin{bmatrix}A_{11}&0&0\\0&A_{22}&0\\0&0&A_{d22}\end{bmatrix}}_{A_{TOT}:3x3}\underbrace{\begin{bmatrix}T_{COOL}\\W_{COMP}\\\psi\end{bmatrix}}_{X:3x1} + \underbrace{\begin{bmatrix}B_{11}&0\\B_{21}&0\\0&B_d\end{bmatrix}}_{B_{TOT}:3x2}\underbrace{\begin{bmatrix}Q_{COOL}\\T_{OUT}\end{bmatrix}}_{u:2x1}\\
\underbrace{\begin{bmatrix}T_{COOL}\\W_{COMP}\end{bmatrix}}_{Y:2x1} &= \underbrace{\begin{bmatrix}C_{11}&C_{12}&0\\C_{21}&C_{22}&C_d\end{bmatrix}}_{C_{TOT}:2x3}\underbrace{\begin{bmatrix}T_{COOL}\\W_{COMP}\\\psi\end{bmatrix}}_{X:3x1} + \underbrace{\begin{bmatrix}D&0\\0&D_d\end{bmatrix}}_{D_{TOT}:2x2}\underbrace{\begin{bmatrix}Q_{COOL}\\T_{OUT}\end{bmatrix}}_{u:2x1}
\end{aligned} \tag{9.1}$$

Equation 9.2 shows how the final system matrixes with numbers, this is the system used in the implementation.

$$\begin{aligned}
\begin{bmatrix}\dot{T}_{COOL}\\ \dot{W}_{COMP}\\ \dot{\psi}\end{bmatrix} &= \begin{bmatrix}-0.809m&0&0\\0&-3.084&0\\0&0&-79.5m\end{bmatrix}\begin{bmatrix}T_{COOL}\\W_{COMP}\\\psi\end{bmatrix} + \begin{bmatrix}-3.642\mu&0\\1.178&0\\0&2\end{bmatrix}\begin{bmatrix}Q_{COOL}\\T_{OUT}\end{bmatrix}\\
\begin{bmatrix}T_{COOL}\\W_{COMP}\end{bmatrix} &= \begin{bmatrix}1&0&0\\0&1&1.634\end{bmatrix}\begin{bmatrix}T_{COOL}\\W_{COMP}\\\psi\end{bmatrix} + \begin{bmatrix}0&0\\0&3.551\end{bmatrix}\begin{bmatrix}Q_{COOL}\\T_{OUT}\end{bmatrix}
\end{aligned} \tag{9.2}$$

The new combined model now has a non-zero D matrix, which is only possible to include in the MPC-toolbox if this part is from the disturbance model and it is defined as either a measured disturbance or unmeasured disturbance in the toolbox as shown in Figure 9.2.

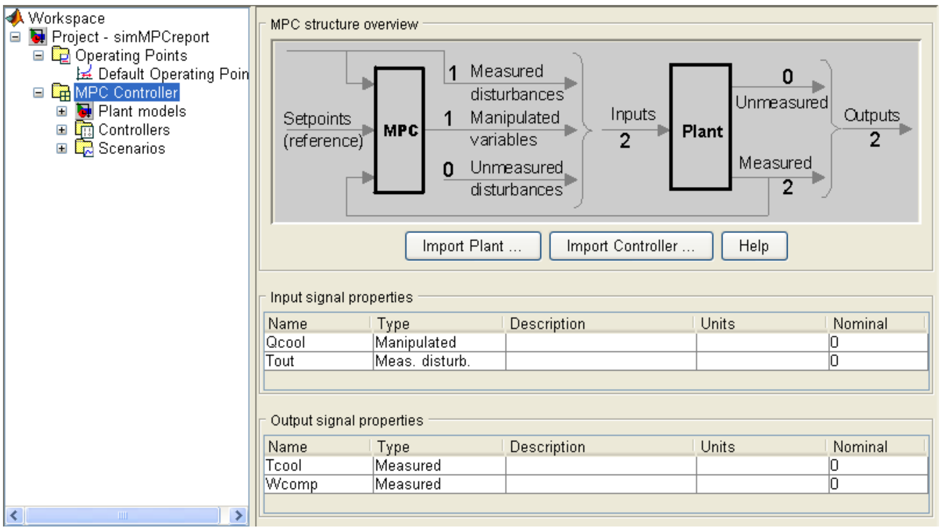


**Figure 9.2:** Screen shot of the MPC-toolbox model options.

## 9.1 The MPC-Toolbox

In the tuning of the controller the MPC-toolbox is very useful because it is possible to simulate the scenario you setup. In this section there are some screen shots that are used to explain how the tuning is done. Under "Controller-MPC1" there are four tabs in the top.

### 9.1.1 Model And Horizons

On this tab the Control interval, Prediction horizon and Control horizon are set. Figure 9.3 shows how this tab is build up.

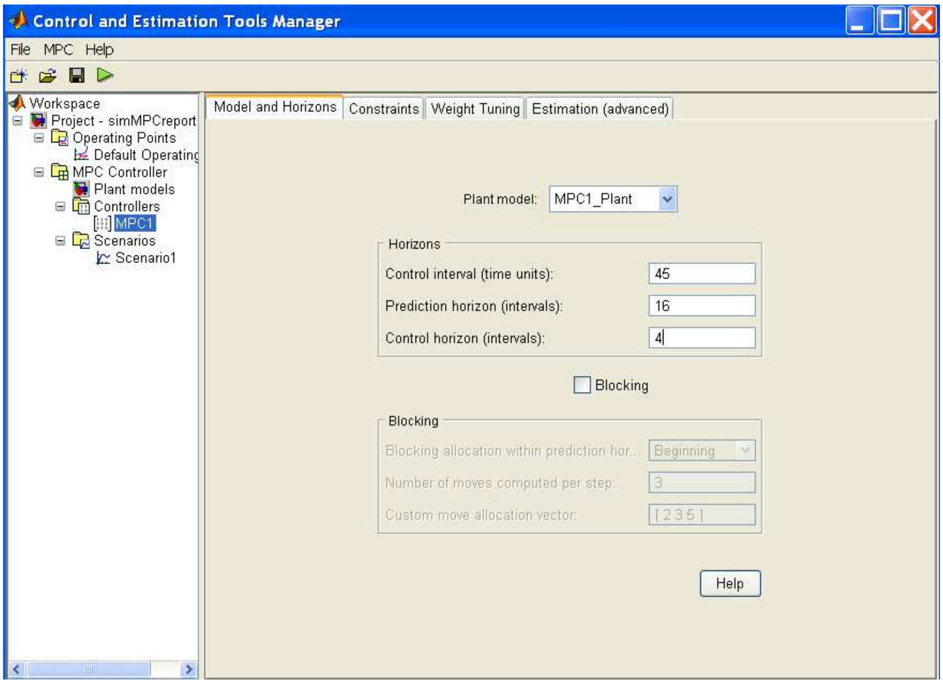


**Figure 9.3:** Screen shot of the MPC-toolbox - Controller options.

**Control interval** This is the interval of how often the MPC-controller recalculates a new output. The time unit is set to 1 sec on the system and the simulated disturbance has a period of 1800 sec, which scales a day of 24 hours down to 30 minutes. It has been chosen to have 40 recalculations among one disturbance period, this gives a control interval of 45 sec.

**Prediction Horizon** this is as it says the prediction horizon, in control intervals, as the controller can use the knowledge of the future disturbances. In this case 40% of the disturbance is assumed to be enough, the higher this number is the more computations are needed to calculate the controller.

**Control Horizon** This is the number of future steps the controller uses to calculate the next controller output. This number must be chosen as a low number because it makes the computation vary hard and demanding. The highest possible number in real time is 4, with a prediction horizon of 16 and constraints put in. This gives a control horizon on 10% of the disturbance period.

### 9.1.2 Constraints

Under this tab, it is possible to give constrains to the input and the outputs of the system. Figure 9.4 shows how this tab is buildup.

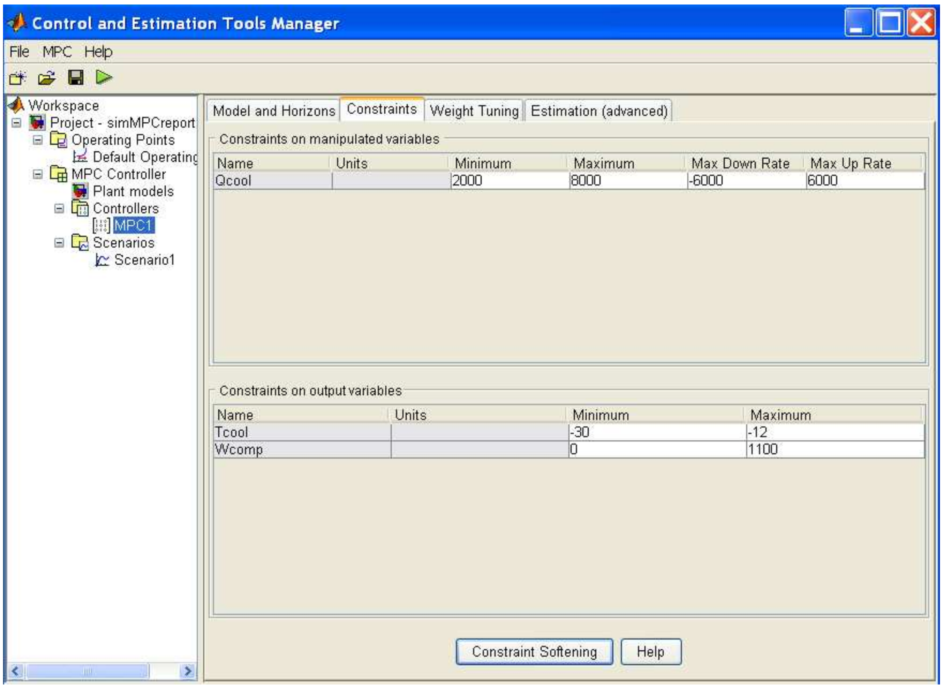


**Figure 9.4:** Screen shot of the MPC-toolbox - Controller options.

**Input -** $Q_{COOL}$**:** It is possible to give constrains to the input in two ways. Maximum and minimum constrains and maximum and minimum rate constrains. The maximum is put is as 8000 and the minimum is put in as 2000 as this is the limits for the cooling capacity. The maximum and minimum is put in as -6000 and 6000 giving the system the possibility to go from one limit to the other in one step.

**Output -** $T_{COOL}$**:** is limited to maximum temperature at -12 $°C$ as this is the temperature that the system not must be above and a minimum temperature at -30 $°C$ as the lower limit to illustrate that the temperature is allowed to be very low.

**Output -** $W_{COMP}$**:** The limits is set to zero and 1100 this means that it is not allowed to use more than 1100 W. this limit is chosen so the system have to use more than 1100 W to keep -12 $°C$ when no prediction is made.

### 9.1.3 Weight Tuning

Under this tab, it is possible to give Weight to the input and the outputs of the system, but it is also possible to weight how robust the controller must be compered to fast. Figure 9.5 shows how this tab is buildup.

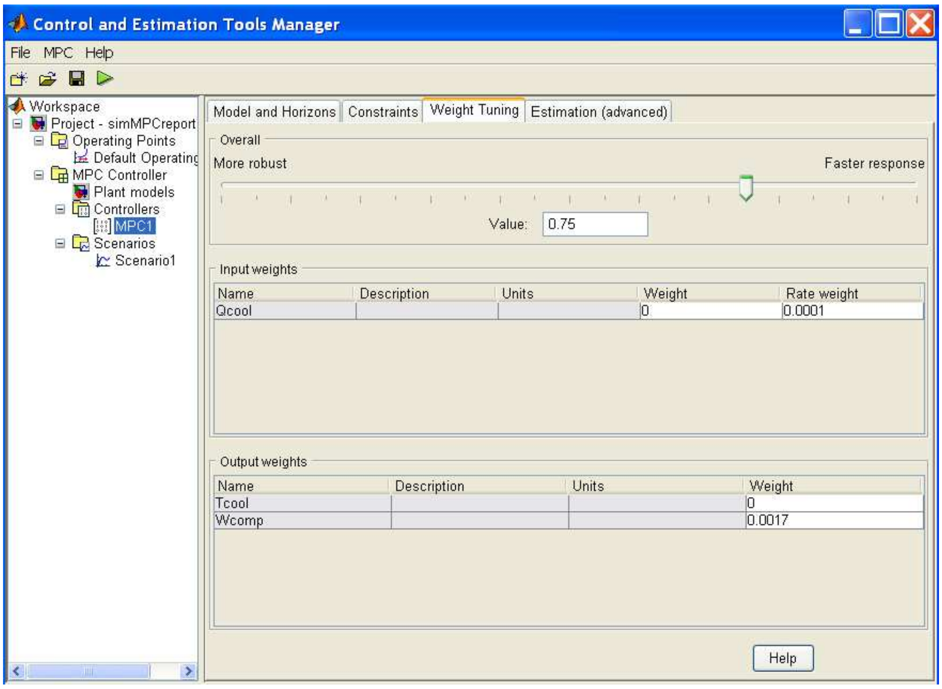


**Figure 9.5:** Screen shot of the MPC-toolbox - Controller options.

**Input -** $Q_{COOL}$**:** In the explanation of the optimization, it was stated that it is not important how large the cooling capacity is, therefor the weight is set to zero, and the weight of the rate of the input is given a vary small number, less that 10 times smaller than the weight of the power consumption. the larger the rate is the less will the controller try to change the $Q_{COOL}$.

**Output -** $T_{COOL}$**:** The weight of $T_{COOL}$ is set to zero, because it is not important what the temperature is as long it is below the upper limit, at -12 $^{\circ}C$

**Output -** $W_{COMP}$**:** The weight of $W_{COMP}$ is set to be 0.0017 as this is the price of one W/h in Denmark.

**Overall:** This the parameter that allows you to make the controller more robust or have a faster response, it has been chosen to set this to 0.8 in favor of the response, what is the standard setting. The faster response the less deviations from the assumed disturbance the controller can handle. If the value is chanced, it will correct the weights in the controller next time you start up and again be at 0.8.

### 9.1.4 Estimation (Advanced)

Under this tab, it is possible to implement models of the sensor noise or input expected white noise on the measurements. For previously test runs on the cooling system, it has been noticed that the measurement noise on the temperature has approximately 0.1 $°C$ amplitude and it is assumed to be white noise. The measurement noise on $W_{COMP}$ is much higher approximately 50 W in amplitude and this is also assumed to be white noise. When knowing these values it will make sense to input the measurement noise in the MPC-Toolbox. Because all outputs in this system is measured only the tab Measurement Noise is given the assumed values as shown in Figure 9.6.

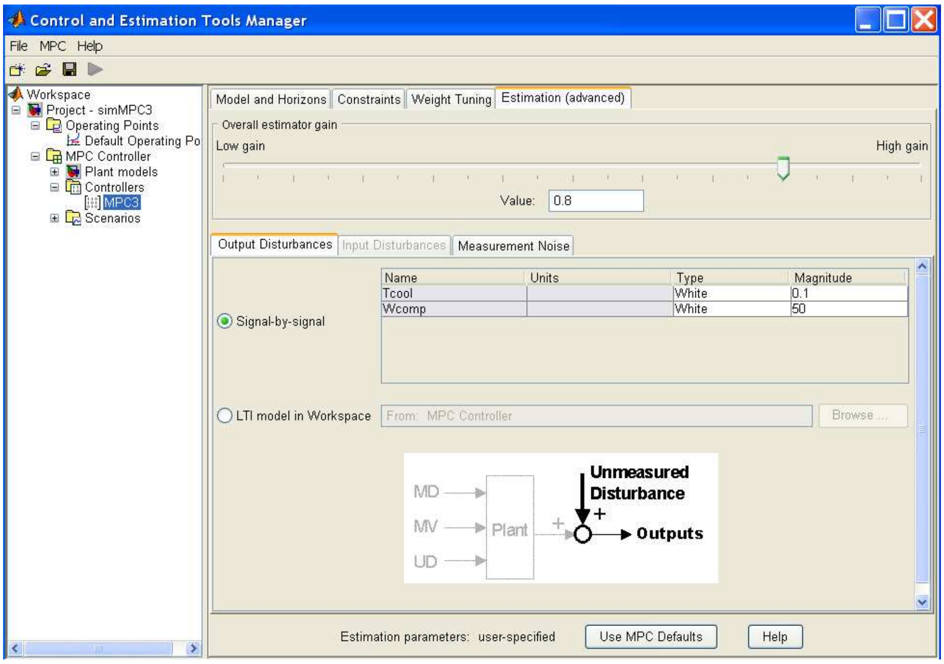


**Figure 9.6:** Screen shot of the MPC-toolbox - Controller options.

The Control Estimator Gain in the top of the figure is the estimator gain for the observer inside the MPC-toolbox. If this is chosen as a low number, it means that you trust the model more than you trust the measurements and opposite. The gain is put at 0.8 stating that the measurements is more likely to be correct than the model in the observer, thereby the model is corrected more than if a low gain was chosen.

# Experiments

# 10

## Contents



*This chapter will explain how the different experiments show how well the controller performs according to the objectives. Starting by describing the disturbance that is similar for all the scenarios. Then follows a description of each scenario where the specific setup is given and what the expected results will be. After this the simulation results are given and explained for each scenario. Next the experiments on the laboratory system are presented and explained. Finally a comparison of the simulation and the experiments of scenario 3 is made.*

## 10.1 Disturbance

In all scenarios the disturbance is exactly the same, this is done to be able to compare the measurements properly. Figure 10.1 on the following page shows the disturbance used in all experiments.

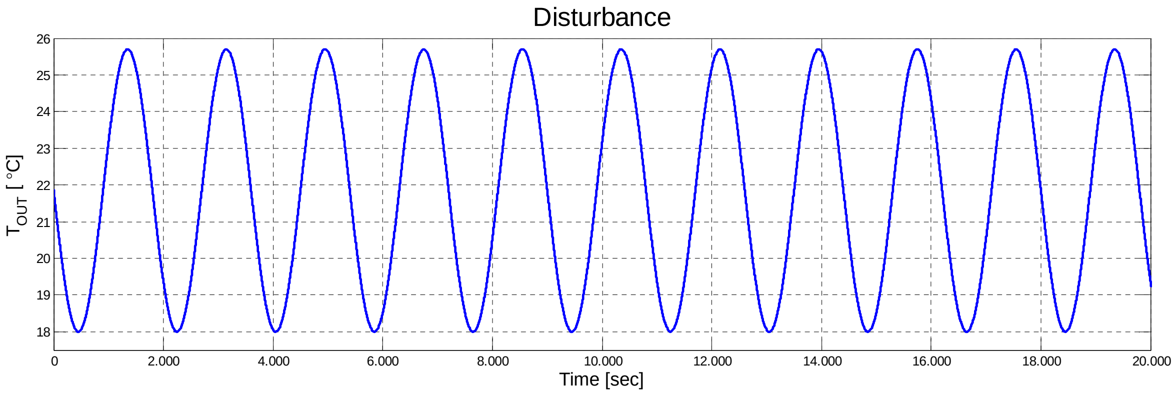


**Figure 10.1:** Plot of the disturbance used in all experiments.

## 10.2 Performance Objectives

This section describes what the tests are supposed to show and what the expected results are of the specific test. The test are evaluated on 5 different parts, $T_{COOL}$ temperature, maximum power consumption (max. $W_{COMP}$), average power consumption (Average $W_{COMP}$), Coefficient Of Performance (COP) and Power consumption per degree(W/T).

In all simulations COP is expected to be the same due to linearization. The disturbance temperature $T_{OUT}$ is the same in all scenarios, the disturbance is changing the COP by changing the slope of the linear model. As stated in section 7 on page 53 the relation between $Q_{COOL}$ and $W_{COMP}$ is a squared relation. Figure 10.2 on the facing page illustrates why the COP is expected to be the same in all the simulations, shown for the mean value of $T_{OUT}$ being 21.85 $^{\circ}C$.

Three different scenarios are tested:

### 10.2.1 Scenario 1

A classic controlled cooling system where there are no limits on the power consumption and the temperature reference is set to keep -12 $^{\circ}C$.

**Purpose:**

The purpose of this test is to show the result of a normal controlled cooling system and use the results from this test mainly as reference values for comparison.

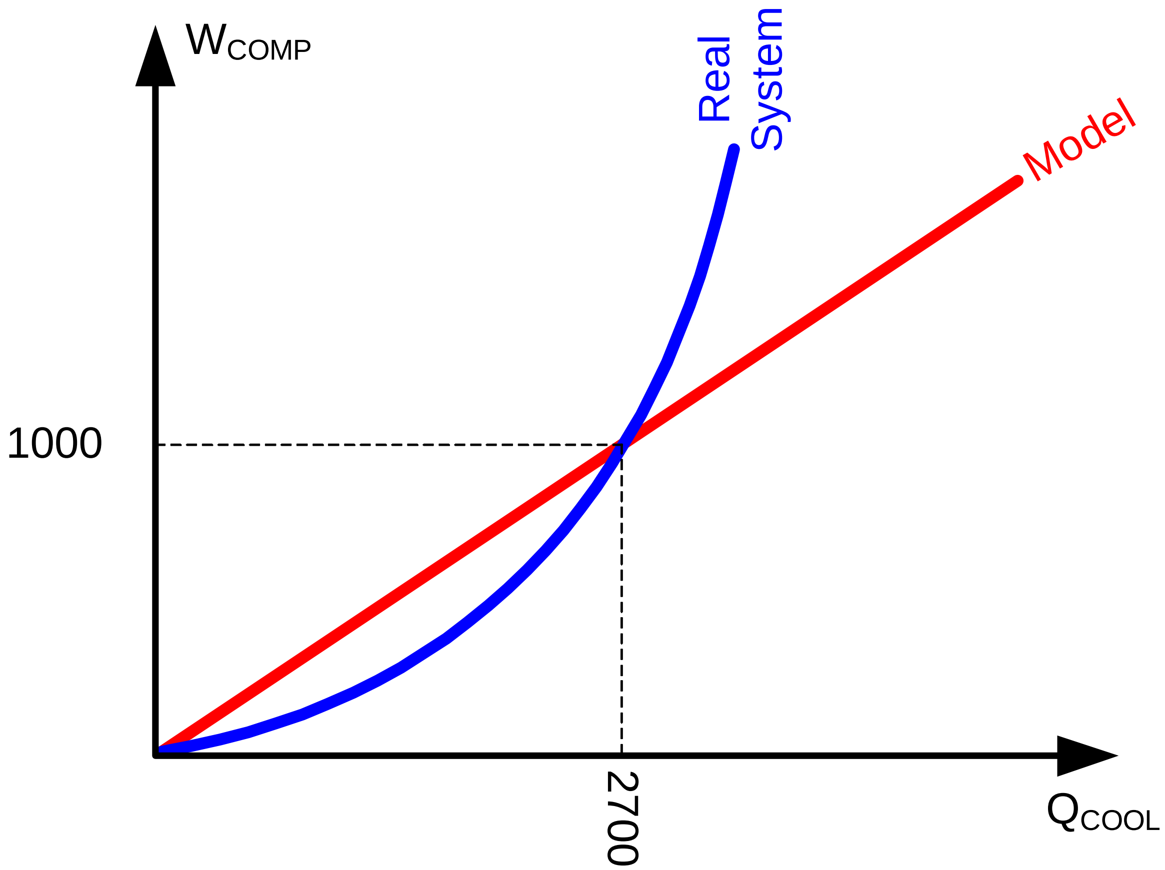


**Figure 10.2:** Illustration of linearization of $Q_{COOL}$ to $W_{COMP}$.

**Expectation:**

**Temperature $T_{COOL}$:** It is expected that this controller will have no problem keeping the reference temperature.

**Maximum $W_{COMP}$:** It is also expected that this controller will make the cooling system use more than 1100 W in the periods with a high outdoor temperature ($T_{OUT}$).

**Average $W_{COMP}$:** It is expected that the Average power consumption is near 1000 W, as this is approximate the power used in the working point.

**COP:** It is expected that the COP in the simulation interval between 2 and four, highest when $T_{OUT}$ is lowest. We expect that the COP is approximate 2.7 because $Q_{COOL}$ is approx 2700 W and $W_{COMP}$ is approx 1000 W in the working point.

### 10.2.2 Scenario 2:

A power restricted controller is implemented on the cooling system in this case. The power will be restricted to 1100W and the temperature limit is set to -12 $°C$ as in scenario 3.

**Purpose:**

The purpose of this test is to make a reference system that keeps the same limits as the predictive controller in scenario 3. This makes it possible compare the two scenarios, where the only difference is, that scenario 3 has prediction implemented and that is not the case in scenario 2.

**Expectation:**

**Temperature $T_{COOL}$:** It is expected that the temperature $T_{COOL}$ will not be -12 $°C$ in the whole test, when the disturbance temperature is high, $T_{COOL}$ will rise above -12 $°C$.

**Maximum $W_{COMP}$:** It is expected that this controller will restrict the power consumption to have a maximum of 1100W.

**Average $W_{COMP}$:** It is expected that this setup will have a lower average power consumption than scenario 1, due to a higher average $T_{COOL}$.
Comparing to scenario 3, it is expected that it will be a bit lower due to a higher average $T_{COOL}$ but on the other hand, scenario 3 is using prediction to cool more in the region with high COP, thereby this scenario will use less power keeping the same or a bit higher average $Q_{COOL}$, resulting in a lower $T_{COOL}$. The gives an expectation of a total higher power consumption in scenario 2 than scenario 3.

**COP:** Simulation: The COP is expected to be 2.7, as in all simulations.
Real: It is expected the the COP will be higher than in simulation because the average temperature is a bit lower, thereby the average $Q_{COOL}$ will be a bit lower. This will according to Figure 10.2 on the previous page give a higher COP.

### 10.2.3 Scenario 3:

The final Controller, with weather prediction and constrains on both power limitation and temperature.

**Purpose:**

The purpose of this test is to show how well the final controller works.

**Expectation:**

**Temperature $T_{COOL}$:** It is expected that this controller is able to keep the temperature constrain.

**Maximum $W_{COMP}$:** It is expected that this controller will restrict the power consumption to have a maximum of 1100W. It is expected to be possible to see that the system is predicting on the power consumption, compared to scenario 2.

**Average $W_{COMP}$:** It is expected that the power consumption is lower that the power consumption of Scenario 2, because this controller is cooling more when the COP is high due to the prediction even if the average temperature is lower in this scenario, what gives a higher power consumption. Comparing to scenario 1, it is expected that the average power consumption is lower, because it is expected that scenario 2 has a lower power consumption than scenario 1.

**COP:** Simulation: It is expected to give 2.7 in COP as all the simulations.
Experiments: COP is expected to be higher than the two other scenarios because this controller is cooling more in the cold period where the COP is high and because it is expected that the average power consumption is lower than the two other scenarios and this according to Figure 10.2 on page 77 gives a higher than the simulation.

**Power consumption per degree:** This is calculated from the average $W_{COMP}$ and average $T_{COOL}$ as shown in Equation 10.1.
Simulation: It is expected that this value will be lower for the MPC controller due to a lower average power consumption compared to the average temperature.

$$\frac{W}{T} = \frac{\text{Average } W_{COMP}}{\text{Average } T_{COOL}} \tag{10.1}$$

## 10.3 Simulation

This section shows the results from each of the three scenarios and at the end a comparison of the three scenarios is made.

### 10.3.1 Scenario 1

Figure 10.3 on the following page shows a plot of the simulation. It is clear that the disturbance plotted in Figure 10.1 on page 76 only influences the power consumption $W_{COMP}$. This is due to the fact that there are no constrains on $W_{COMP}$ in this scenario and thereby do the disturbance not does not affect $Q_{COOL}$ or $T_{COOL}$. The temperature is settling a -12 $°C$ as expected. $W_{COMP}$ is going above 1100 W.

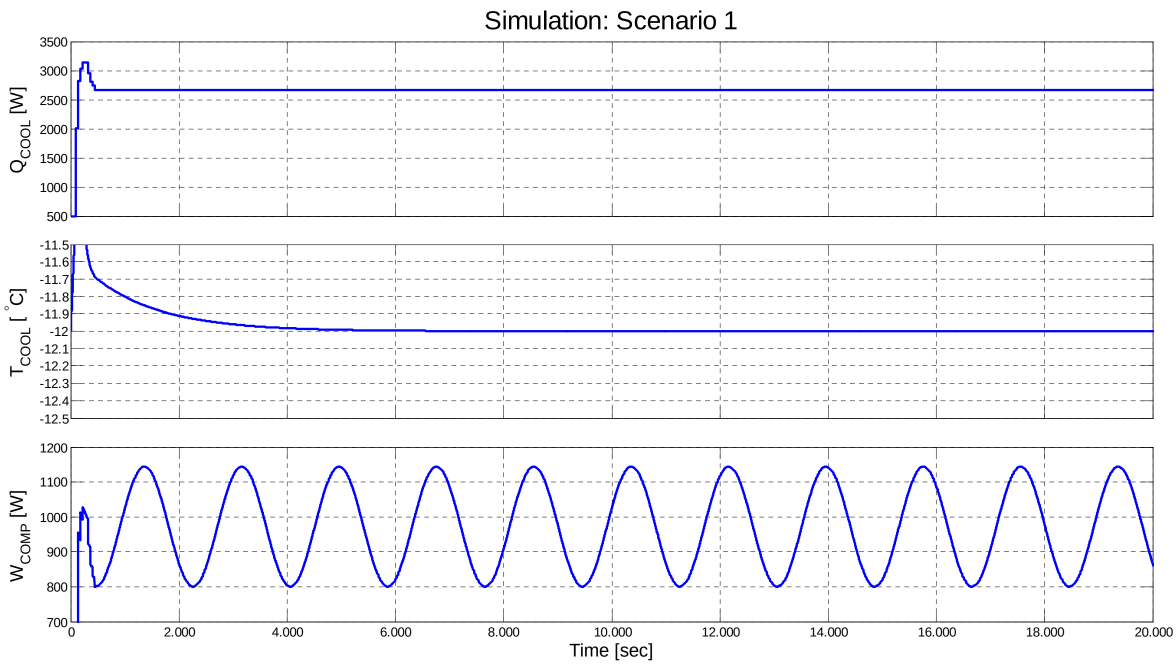


**Figure 10.3:** Simulation: Test result of scenario 1.

The average power consumption is calculated to 990.37 W, what is close to the expected value of 1000W.

The average COP is calculated to 2.69, this was expected to be 2.7

#### Sub Conclusion

The simulation give the expected values and the simulation is considered to show the correct values.

### 10.3.2 Scenario 2

Figure 10.3 shows a plot of the simulation. The simulation shows that the power consumption is limited at 1100 W, the limit has a saw tooth shape. This shape is due to the 45 sec control interval. The temperature $T_{COOL}$ is kept at -12 $°C$ except when the power is restricted.

The average power consumption is calculated to 987.14 W, this is as expected lower than scenario 1 where it was 990.37 W. This lower power consumption is due to the lower average temperature of $T_{COOL}$.

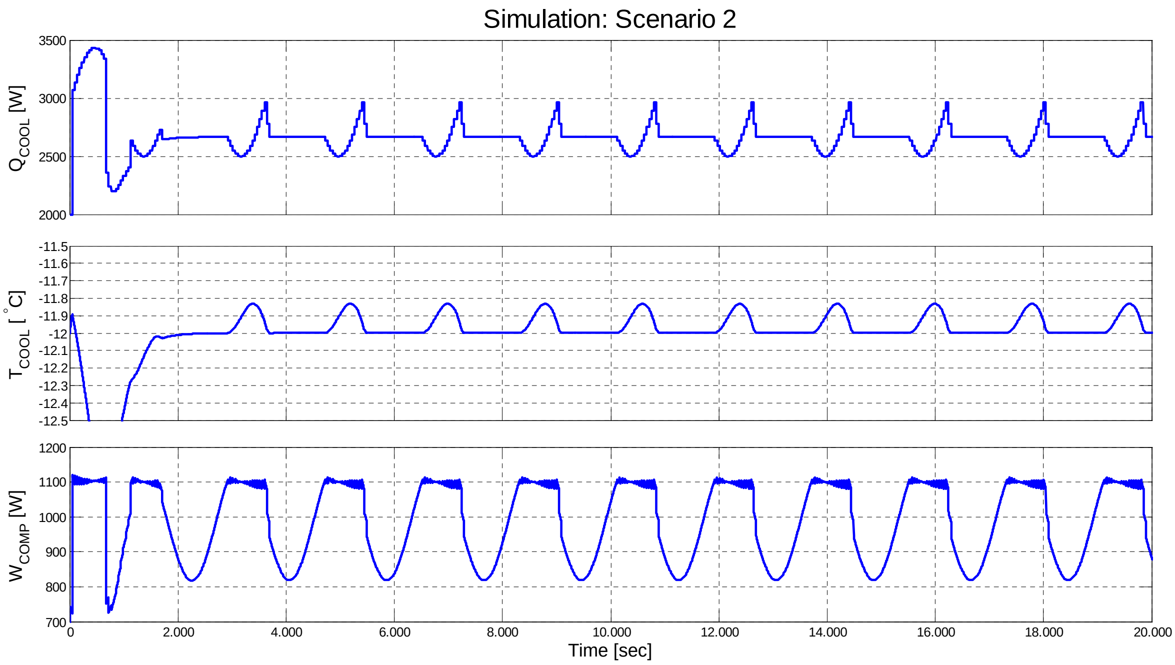


**Figure 10.4:** Simulation: Test result of scenario 2.

The calculate average COP is 2.69 as expected this is exactly the same value as calculated in scenario 1.

### Sub Conclusion

It is concluded that this simulation gives the expected values.

### 10.3.3 Scenario 3

Due to hardware or software limits in the laboratory implementation, it is not possible to implement the MPC designed, with a control interval at 45 sec and a prediction horizon at 16 and a control horizon at 16. In the laboratory the implemented controller has a smaller control horizon it is here 4. This will give a different result. To show the different a simulation of both controllers has been made and plotted in Figure 10.5 on the next page

In both simulations the constrains on $W_{COMP}$ and $T_{COOL}$ are kept. It is clear that the controller with the long control horizon is the best controller when you compare the temperature $T_{COOL}$, the mean temperature is higher in this simulation and thereby it will use less power. Comparing the power consumption $W_{COMP}$ in these two tests, there are some differences caused be the difference in control horizon, but in general they follow the same

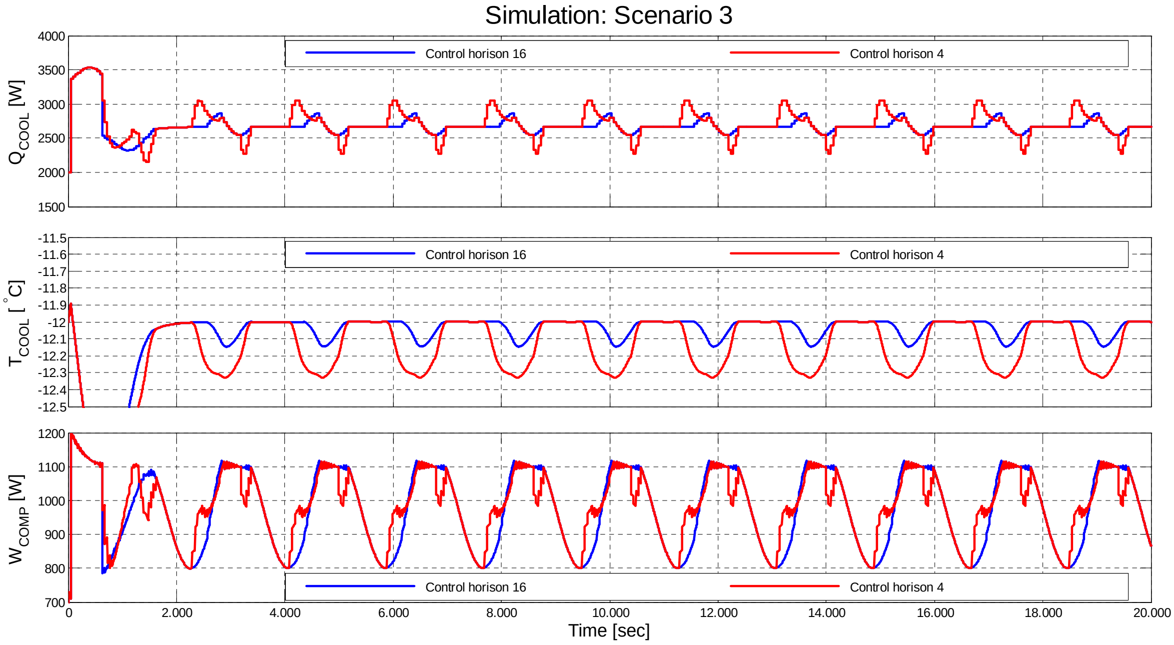


**Figure 10.5:** Simulation: Test result of scenario 3.

curve.

The mean power consumption is calculated to 995.15 W in the simulation with the prediction horizon at 16 and 1005.80 W with a control horizon at 4. This is in both cases higher than both scenario 1 and scenario 2, this was not expected, but can be explained by a lower mean $T_{COOL}$. 1 $°C$ lower average temperature will give an increase of $\frac{222.22}{2.69} = 82.53$ W in power consumption.

The calculated average COP is as expected 2.69 in both scenarios.

### Sub Conclusion

Both test give a satisfying result, but it is not possible to get a lower power consumption this way. To be able to lower the average power consumption, a cooling room with better isolation is needed. Better isolation will mean that the average power consumption will increase less than $\frac{222.22}{2.69} = 82.53$ W to keep a 1 $°C$ lower $T_{COOL}$. Another solution is to have a more full cooling room, in this laboratory system, which means, increase the water in the water heater from 73 liters of water to e.g. 100 liter water. This will mean that it is possible to store more energy in the water considering the same temperature change. This will mean that in scenario 3 the mean temperature of $T_{COOL}$ will be lower thereby saving energy. The best controller of the two shown in scenario 3 is the controller with the long prediction horizon but both are keeping the given constrains.

| | Average $T_{COOL}$ [$^\circ C$] | Average $W_{COMP}$ [W] | W/T [W/$^\circ C$] |
|---|---|---|---|
| Scenario 1 | -12.0000 | 990.4 | 82.5314 |
| Scenario 2 | -11.9608 | 987.1 | 82.5316 |
| Scenario 3,16 | -12.0579 | 995.1 | 82.5312 |
| Scenario 3,4 | -12.1868 | 1005.8 | 82.5312 |

**Table 10.1:** Table to compare all scenarios including power consumption pr. degree.

### 10.3.4 Comparison

The simulation of the three scenarios has already been shown, here the three results are plotted in the same figure to be able to compare the three results, from scenario 3 only the best controller is plotted, this is the controller with a control horizon at 16.

Figure 10.6 is a plot of all three scenarios and makes comparing easy. On all three graphs it is easy to see that the controller from scenario 3 is starting to cool before the controller in scenario 1 and 2, this is due to the prediction.

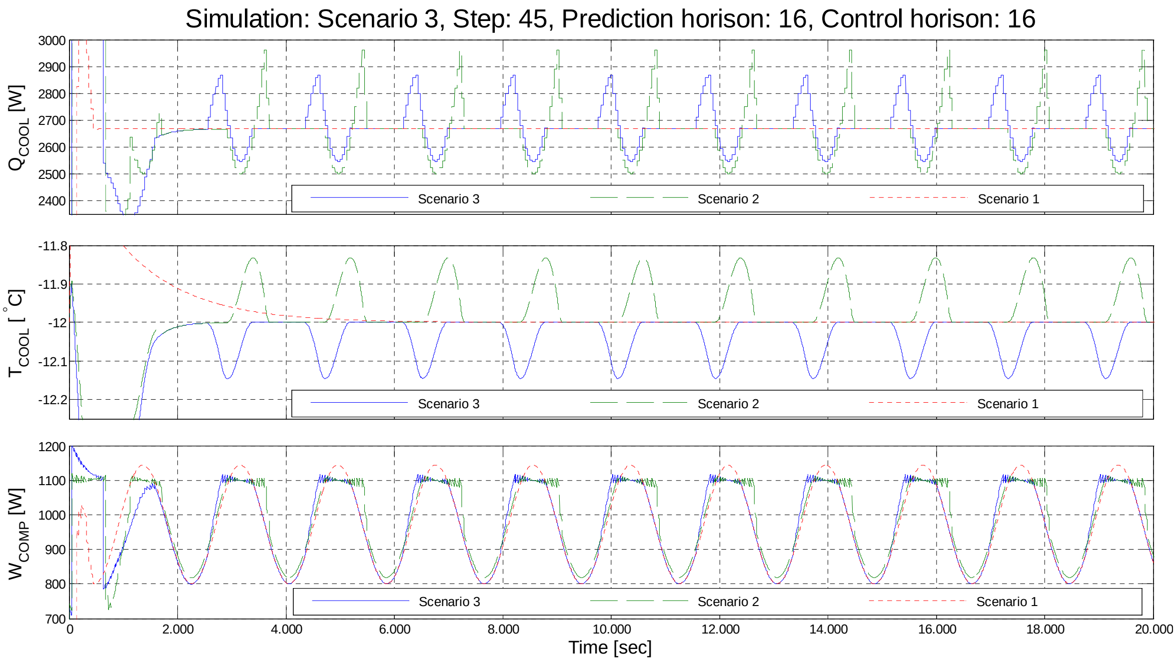


**Figure 10.6:** Test result of the three scenarios in the simulation.

Finally it makes sense to compare the power consumption per degree for the three scenarios. The reason for this comparison is the the average temperature is not the same in any of the scenarios. Table 10.1 Shows the calculated values for each of the scenarios.

### 10.3.5 Sub Conclusion

There er no surprises in how the controllers work, they all behave as expected. But it was not expected that the predictive controller uses more power than the two other controllers. This can easily be explained by the fact that the mean temperature in the cooling room is lower in scenario 3.

## 10.4 Experiments

This section shows the results from the experiments on each of the three scenarios and at the end a short comparison of the three scenarios is made.

### 10.4.1 Scenario 1

Figure 10.7 on the facing page shows a plot of the experimental results. The temperature $T_{COOL}$ is not as expected a flat line, it is deviating approx. 0.3 $°C$. The power consumption $W_{COMP}$ is as expected following the disturbance, but by looking a bit closer on $W_{COMP}$ it must be noticed that the shape is not an exact sine wave. It seems as that the low power consumption is a bit more flat than the high power consumption. $Q_{COOL}$ is plotted in the top graph and in this the measured, filtered measurement and reference are plotted. Notice that the filtered does does deviate from the reference. It must be noticed that this deviation is appearing each time the power consumption $W_{COMP}$ is getting near a minimum. This is where the COP is high in the cooling system due to a low disturbance temperature $T_{OUT}$, this means that the compressor speed is getting closer to its minimum. This is a problem because the $Q_{COOL}$ controller designed has a fast response and thereby a big overshoot and a fairly long settling time when getting close to the limits for the control signal, the overshoot in the limited direction will be limited changing the mean value of the control signal, thereby increasing the real cooling capacity $Q_{COOL}$ that is measured.

#### Sub Conclusion

This experiment did not give the wanted result, but because the mean temperature of $T_{COOL}$ is approx. -12 $°C$ the results from this experiment can still be used when comparing the three scenarios.

### 10.4.2 Scenario 2

Figure 10.8 on page 86 shows a plot of the experimental results. The temperature $T_{COOL}$ does not show the expected curve. It was expected that the temperature was going to keep -12 $°C$, except when the power consumption was limited at 1100W, where the temperature

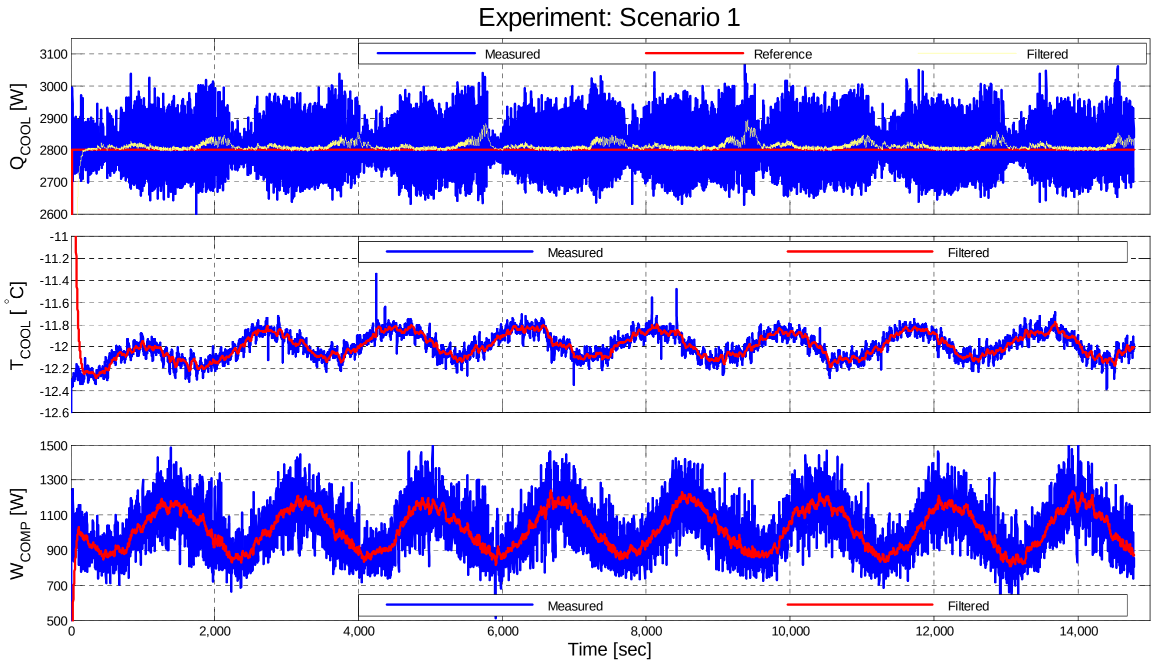


**Figure 10.7:** Experiment: Test result of scenario 1.

was expected to increase. Somehow it looks as it has twice the disturbance frequency. This can be explained by using the result from scenario 1 as this has the disturbance frequency in the temperature measurement due to the $Q_{COOL}$ controller and it was expected to see the disturbance frequency in this temperature measurement as well. Due to the phases of the two frequencies not being the same, measurements with the sum of the frequencies appears.

The temperature does not seem to settle at the -12 $^\circ C$ as it is the limit put in to the controller, this can be explained by the model and system mismatch from $Q_{COOL}$ to $T_{COOL}$ as explained in Section 7.2 on page 59. If this would be corrected in the laboratory cooling system the temperature shout drop approx. 0.3 $^\circ C$ and if the $Q_{COOL}$ is corrected it is assumed it will increase the temperature 0.1 $^\circ C$ giving a resulting temperature of -12 $^\circ C$ as in the simulation. All this will maybe be easier to understand when a comparison of simulation and experiment is made later on in this Chapter.

### Sub Conclusion

It can be concluded that the controller did not behave exactly as expected but it seems as there are explanations of why it does not behave that way. The results given in this scenario ares weighted low when comparing the scenarios and are not used as pros or cons on the predictive controller.

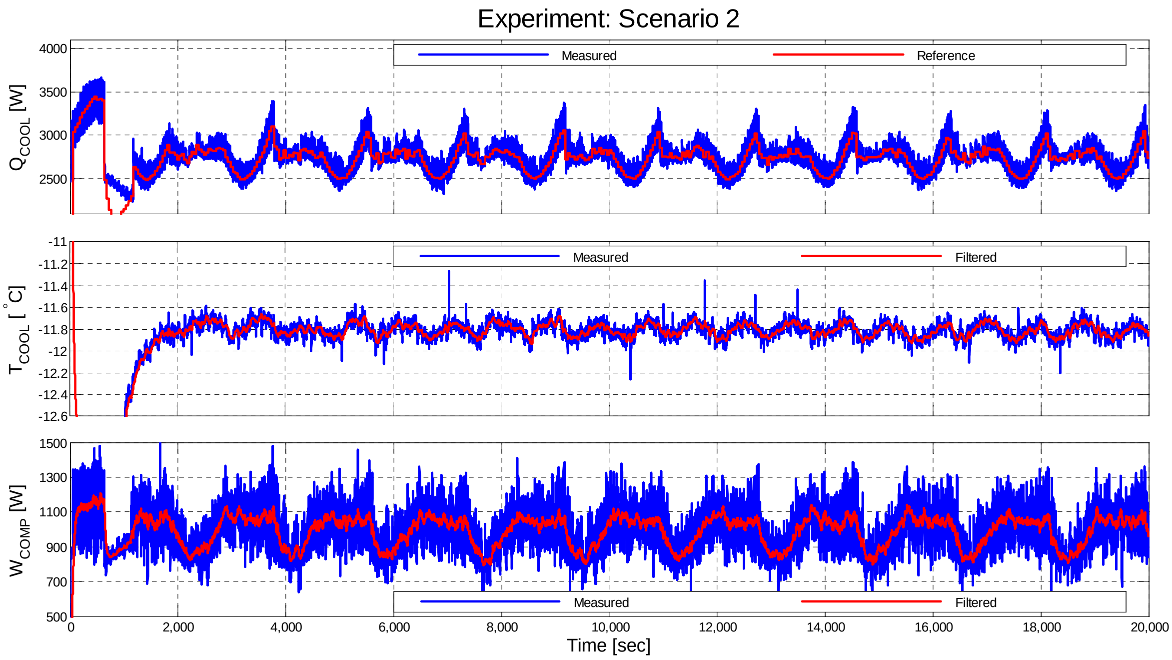


**Figure 10.8:** Experiment: Test result of scenario 2.

### 10.4.3 Scenario 3

Figure 10.9 on the next page shows a plot of the experimental results. This scenario shows the most convincing results and was indeed the that was run. Starting by examining the temperature plot of $T_{COOL}$, a temperature of approx. -11.8 $°C$ seems to be the upper limit instead of the expected -12 $°C$, this is again explained by the mismatch in the model from $Q_{COOL}$ to $T_{COOL}$ as explained in Section 7.2 on page 59. It also seems as the limit in not totaly flat, this behavior is explained by the $Q_{COOL}$ controller that made the temperature in scenario 1 oscillate due to limitation in the control signals to the compressor. Again a replacement of the $Q_{COOL}$ controller will without doubt solve this problem.

In the power consumption $W_{COMP}$ a clear repeating pattern is visible. It is clear that the controller is predicting because $W_{COMP}$ is changing from the lowest value to the limit of 1100 W in approx. one step at the time. It does this until the temperature $T_{COOL}$ is at the limit where it is settled. Looking at the graph of $Q_{COOL}$ only supports this statement.

#### Sub Conclusion

It is concluded that the controller is behaving as expected if neglecting the offset in temperature caused by the model mismatch as mentioned. The power constrain on $W_{COMP}$ is at the 1100 W as put in to the controller.

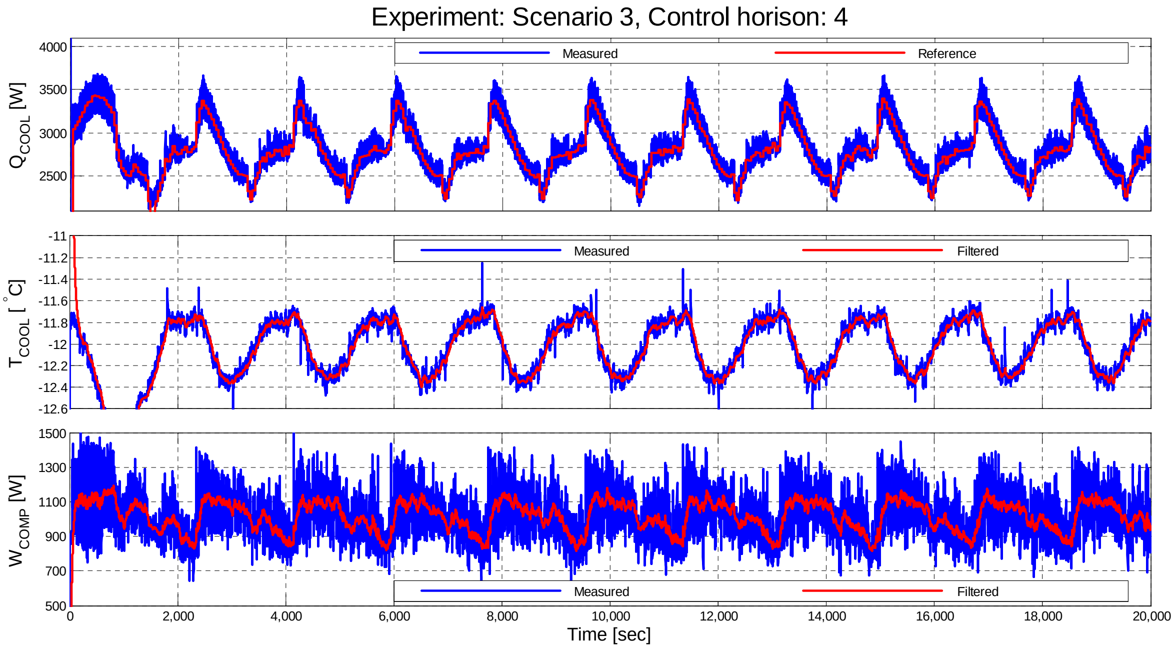


**Figure 10.9:** Experiment: Test result of scenario 3.

### 10.4.4 Comparison

Comparing the three scenarios is in the experiments not possible to do in a graph, due to noise on the measurements making it impossible to be able to see the three scenarios.

The comparing is only done by calculating the differen values given in Table 10.2 the values easy to compare is COP and W/T. In the table COP is a slightly higher in scenario 3 and W/T is slightly smaller both supporting higher efficiency for the cooling system. It are both still very small changes and because the scenario does not keep the temperature limit due to system implementation errors as earlier described, it does not seem as lower power consumption is a valid argument alone for implementing MPC on this cooling system.

| | Average $T_{COOL}$ [$°C$] | Average $W_{COMP}$ [W] | COP | W/T [W/$°C$] |
|---|---|---|---|---|
| Scenario 1 | -12.0014 | 1021.6 | 2.7545 | 85.1237 |
| Scenario 2 | -11.8424 | 998.7 | 2.7515 | 83.4868 |
| Scenario 3 | -12.0328 | 1015.2 | 2.7628 | 84.3736 |

**Table 10.2:** Table to compare all three scenarios.

## 10.5 Comparison

This section will try to compare the simulation from scenario 3 to the real system tests of scenario 3. The real system has a control horizon of 4 and therefor is the comparison made with the simulation with the same control horizon.

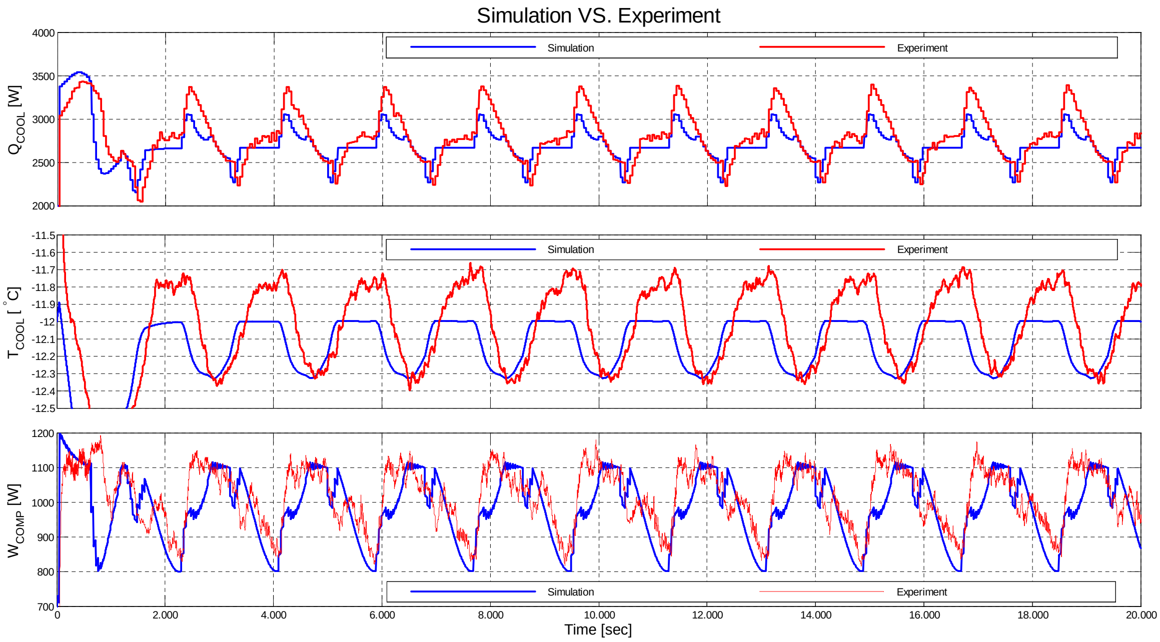


**Figure 10.10:** Plot of simulation compared with real test.

When comparing the power consumption from the simulation and the real test, it is clear the the same pattern is seen in both the simulation and the real test.

Comparing the temperature $T_{COOL}$, it is clear that the real test has a offset in the temperature as mentioned earlier. A phase delay is also seen when comparing the temperature. The delay is approx. 250 seconds. This delay is due to model error in the model from $Q_{COOL}$ to $T_{COOL}$ as this is derived only from energy equations neglecting the evaporator dynamics and the water heater dynamic as mentioned earlier. The temperature offset is as mentioned descending from a offset in $Q_{IN}$ to the water heater on approx. 80W, what will give a temperature decrease at approx. 0.3 $°C$. $Q$, what will make the temperature go below -12 $°C$

# Conclusion

# 11

The first main objectives of the weather predictive controller are to keep the maximum limit of 1100 W on the power consumption of the compressor and a maximum limit of -12 $^oC$ on the temperature in the cooling room. In both simulation and experiment the limit on the power consumption of the compressor is kept by the designed MPC controller. In simulation the desired cooling temperature $T_{COOL}$ is restricted to the desired level of -12, however in experiment this limit is passed by approximately 0.3 $^oC$, up to -11.7 $^oC$. This is due to an error in the laboratory test facility, where there is assumed to be an error in the power measurement of $Q_{IN}$, which is the power added to the water in the water heater.

When the maximum temperature is reached in the experiments, it does not give a constant temperature during that time. This is caused by the controller on the compressor for the cooling level $Q_{COOL}$. Since it is desired to have a very fast rise time, no overshoot and no steady state errors in the controller design, a compromise was made in the control design resulting in a controller which has a fast rise time and this gives also a large overshoot causing transient behavior. This transient behavior gives inaccuracy when the control signal is getting near its lower limit for the control signal, which leads to slow temperature settling that can be seen in the maximum temperature. This effect will however be of much less significance if the system has a disturbance period in day cycles, which is many times bigger than the half hour cycles used here and causing the controller to settle.

When comparing simulations and experiments a time delay of approximately 200 sec. is observed. This is (mainly) caused by the absence of the time delay in the model from $Q_{COOL}$ to $T_{COOL}$, which in the real system is present because of the delay in the evaporator and water heater. This will also be less significant if the disturbance is in day cycles.

By comparing the simulations and experiments from scenario 3, it can be concluded that the match between simulation and experiment of the Model Predictive Controlled cooling system is accurate. If neglecting the temperature offset caused by the measurement of $Q_{COOL}$, simulations and experiments fit very closely. This means both the restrictions on the power and temperature are kept. The exact same pattern can be seen in both cases. Therefore the simulations are further on used to make conclusions regarding the use of Model Predictive Control compared to classic control and restricted control. Using simulations has the advantage that it is faster to get results, there is no noise, even values with small differences can be compared. The disadvantage is that it uses linearized equations, thereby COP is constant.

Since the mean COP is a constant of 2.69 in the different simulation scenarios due to linearization of $Q_{COOL}$ to $W_{COMP}$, no conclusion can be directly drawn for the performance of the system regarding the direct benefits of predictive control on the average power use of the system. Therefore the work needed to cool the temperature in the room by one degree is used as a way to compare the how much the performs in the different scenarios. In the classic controlled scenario the value is 82.5314 W/$^oC$ and in the predictive scenario this is 82.5312 W/$^oC$. This gives that the predictive controller has a higher efficiency than the classic controller, due to prediction. The average power used by the two controllers is 995 W for the predictive and 990 W for the classic controller. This shows that even if the predictive controller has a higher efficiency, which can be seen in the work per degree, it does not save money due to the mean temperature being lower in the predictive case.

The overall conclusion is that weather MPC controller can be very valuable for use in cooling systems, especially for keeping restrictions on maximum power use and maximum temperature inside the cooling room.

However, it appears that a mismatch of the real system and system model, which is used in the design of the MPC controller, can cause serious errors in keeping the required restrictions. The MPC controller shows not to have enough feedback action to correct for significant model mismatches, even if the observer gain is high.

A positive effect on the power use of the system can be obtained if the water reservoir, which in real life is e.g. meat in the cooling room, is increased in volume or the cooling room is better isolated. An increase in water will mean that the temperature will not need to be lowered as much to store the same amount of energy. Thereby lowering the average $Q_{IN}$ making the power consumption be lowered. It is in general the same by better isolation $Q_{IN}$ is lowered and thereby the power consumption.

In situations where the disturbance is more natural, not a sine wave having the same amplitude and offset each day, the predictive controller will behave as a classic controller, if the power is not being limited. As soon as the predictive controller notices that the disturbance will cause a power constraint violation, the predictive controller will start to make use of prediction to change the temperature in the cooling room to make sure that the temperature never goes above the maximum allowed. This will cost a small amount of extra total power consumption due to a lower mean temperature, but the power consumption constraints are not violated in the MPC case.

The big advantage in using MPC if compared to a classic constrained controller, is that all constraints are kept without lowering the temperature constraints, as would be necessary in the classic control case to be sure not to go above a predefined limit and lowering the temperature will increase the power consumption. MPC will work as an optimal controller in days with low disturbances and only when necessary lower the temperature.

# Recommendation 12

*This chapter is giving recommendations for future work, starting by recommending small corrections that will improve the performance of the designed Model Predictive Controller. The further you read in this chapter, the more it will be placed in bigger perspective.*

To get the experiments to be even more like the simulations, two recommendations are given:

1. Make the experiments again with the disturbance as a real time disturbance, meaning the the frequency of the disturbance period must be set to one day.
2. Make a new $Q_{COOL}$ controller using knowledge about the disturbance and $QCOOL$ to $W_{COMP}$ to make some Feed Forward thereby getting a fast rise time and tune the controller to get a low overshoot and a fast settling time.

The next step is implementing the weather forecast trough e.g. DMI to get real weather forecast to be used in the predictive controller.

In the bigger perspective using MPC in a building air-conditioning system, where it is not possible to under cool the air temperature, but instead install a big isolated tank with e.g. water and add an extra condensor, where already cooled water from the isolated tank is used to lower the condensor pressure on hot days. And in night time when the outside temperature is lower use the extra cooling capacity to cool the water in the isolated water tank. This will make it possible to keep the inside temperature in the building at a lower temperature and if the isolated tank is big enough and the isolation is good to be able to save some power. This will again be a optimization problem. How big the tank must be and how good the isolation is needed.

Seen from another angle. If this is implemented in the production of e.g. an air-conditioning plant, it will be possible to make a smaller system perform as good as a bigger system without the MPC part implemented. Thereby making the system cheaper but this depends of course of the cost of applying the extra parts on the system.

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