MASTER

Optimizing climate control systems for museums
a case study for the Hermitage Amsterdam

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Preface
This report is the result of my final project for the Master’s degree program in Building Services at the Eindhoven University of Technology. After I had finished my Bachelor’s degree program at Hogeschool Utrecht, and the premaster program during the Bachelor’s degree program, I started the Master’s degree program in February 2010. I followed this program as a part-time student and combined my study with my work at Kuijpers Installaties B.V. After I had finished my M1-project, under the supervision of Henk Schellen, I started with my search for a subject for my final thesis. After some informative discussions within Kuijpers Installaties B.V., a subject that dealt with energy consumption and physics of monuments came up. It was important for me to select a subject for my thesis that fitted with my motivated interest in modeling together with the interest of Kuijpers Installaties B.V. Because I had very good experiences with Henk as a supervisor, Kuijpers Installaties B.V. saw the usefulness of the research and I thought the topic of the research was really interesting, a starting point for my thesis was found in the subject of optimizing climate control systems for museums.

First of all I want to thank Kuijpers Installaties B.V., especially the establishment in Roosendaal, because they have given me the opportunity to carry out this study in fulltime employment. Of course I want to thank some people of Kuijpers Installaties B.V. I want to thank Bas Kuypers for helping me finding a subject for my thesis and his involvement in the future PhD research. I want to thank Rob Spee for his supervision during the research, for reading and correction my thesis and for the transfer of his knowledge about the control of HVAC systems. I also want to thank Aad Juffermans and Fred Hauser for giving me insight in the climate control systems of museums and for providing me climate data of the Hermitage Amsterdam.

I would also like to thank Sebastiaan Lagendaal, maintenance coordinator of the Hermitage Amsterdam, for his information about the Hermitage Amsterdam, the opportunity he gave me to visit the museum and gather measurement data and information from the building management system and his enthusiastic approach of my thesis.

The project meetings were planned every two to three weeks. During these meetings, I discussed together with dr.ir. Henk Schellen (and often also dr. ir. Jos van Schijndel) the progress and results I had so far. These meetings were very helpful to maintain the progress of the project and were necessary to keep the project on the right track. Besides these meetings I had also the possibility to contact dr. ir. Jos van Schijndel with additional questions about the simulation part of my thesis. I would like to thank them sincerely for their assistance and their very positive, motivating and enthusiastic approach of my thesis.

Bram, ever since the last period of our Bachelor’s degree program in Utrecht we can get along very well. You decided to join me in the premaster program, and later the Master’s degree program, at Eindhoven University of Technology. During my study, and certainly also during the writing of my thesis, you were always willing to answer a question from me. Often, questions of ‘mathematical nature’ were your specialty and vice versa, I was able to help you with the ‘organizational matters’. Thank you very much for helping me out several times during the last three years and we will certainly meet in the ‘professional life’.
And last but not least I want to thank, from the bottom of my heart, my mom and dad for their unwavering mental and ‘financial’ support. It was a long road I've traveled since I've quite the Norbertus College without a diploma and you always encouraged me, study after study. And of course Isabelle, where would I be without you? I know, living with me as a student was not always easy and often not pleasant but you always continued to support me. I admire your patience and thank you for giving me this opportunity. Now there will be time for new things!

Marco Maas
Roosendaal, November 2012
Summary

Several authors have stated that allowing a wider range of indoor climate conditions and allowing a temperature set point that fluctuates with the season will result in lower energy costs in museums. Martens projected this hypothesis on the Dutch situation. However, he only used a simplified fictive simulation model to draw conclusions and did not substantiate this with measurement data. Therefore, this lack of research and the (bad) functioning of (oversized) climate systems and the associated high energy consumption are used as a starting point for this thesis to investigate opportunities for energy conservation in Dutch museums with preservation of collections, building and comfort for visitors and staff. The objective of this research is to give insight in the optimization of climate systems, especially of an air handling unit of the Hermitage Amsterdam. The research should clarify whether the optimization could lead to less energy consumption without a decrease of the conservation conditions of cultural objects and the comfort for staff and visitors of the museum. In this research, the possible energy savings by changing set points for indoor temperature and relative humidity is investigated. This is done by making use of simulation software HAMBASE. The modeling of the climate control systems was done by making use of MatLab/Simulink. Gradually the focus shifted from the simulation of the whole climate control system to only the simulation of the heating coil and eventually to the modeling of the control of the heating coil.

To achieve the objectives of this research, the Hermitage Amsterdam is used as a case study to make a model of a museum building and climate control systems. The building was built in 1683. The building has been substantially renovated in 1970, the recent restoration dates from 2007-2009. The Hermitage Amsterdam is located in the center of Amsterdam. The museum has no collection of its own. All exhibited artworks are on loan from The State Hermitage Museum in St. Petersburg, Russia. The focus of the research is on the optimization of climate control systems for museums, therefore only the sections where the exhibition areas are situated are important. The section parallel to the Nieuwe Herengracht, especially the main exhibition are in this section (B.2.06 ‘Wisselexpositie’), is chosen to be researched. In the current situation, the air is kept at a very strict level, with a very small bandwidth. This is an advantage when the effect of changing the set point and bandwidth on the energy consumption of the building is investigated, because possibly significant results might be obtained. The exhibition area is conditioned by only one air handling unit (only for this area) which makes it more easy to model the system in MatLab/Simulink.

A nine zone HAMBASE model was constructed to simulate the building of the Hermitage Amsterdam. Architectural/mechanical drawings and engineering specifications were used to gather information about the dimensions, structure and building physical aspects of the building. The building management system was used to provide measurement data, of the year 2011, for outdoor and indoor temperature and relative humidity data. Also measurement data from the air handling unit was obtained from the building management system. During the research it was attempted to build a complete model of an air handling unit. As a starting point, components from an older research were used. While investigating and adjusting these components it was discovered that the model of the cooling coil might not function when condensation occurs. The mixing section and the humidifier were not validated because measurement data was missing. Only the heating coil and its control was investigated.
From the results in this research one can conclude that the optimization of temperature and relative humidity set points could possibly lead to less energy consumption while maintaining the right indoor climate conditions for the preservation of the collection and the thermal comfort of visitors and staff. Total energy savings in the range of 2 to 64% can be achieved. The two best optimizations/combinations of the temperature and relative humidity set points, according to energy consumption, conservation of cultural objects and comfort of staff/visitors, are the one that applies a bandwidth to the set points for temperature and relative humidity (±2°C and ±10%). This option might realize an energy saving of approximately 55% (65 MWh). And the one where temperature and relative humidity set points are applied that fluctuate with the seasons based on a sine curve (set point 41). The equilibrium position is set to 18/23°C and 40/50% and the amplitude is set to ±2°C and ±10%. A possible energy saving of 64% might be achieved (76 MWh). However, the conservation conditions and thermal comfort of staff/visitors are less optimal for this option. The reproduction of the research of Mecklenburg resulted in possible an energy saving of approximately 10% (12 MWh) when the relative humidity fluctuation is set to ±7%. Optimizing the set points of the exhibition area might help restoring the imbalance of the thermal energy storage system.

It can also be concluded that it is not yet possible to exactly reproduce the current indoor climate and air handling unit by making use of modeling software. An exact validated HAMBASE model of the building is difficult to construct. Problems occur when modeling the components of the air handling unit, especially when modeling the control of the components. However, the preliminary results are promising and progress has been made in the area of modeling the control of climate systems. The determination of a (linear) function to model a component while circumvent the control part could be a good way to achieve the desired results. Recommendations for future research are: simulate with other year and outdoor climate data, simulate and optimize primary systems (systems that generate the heat and cold e.g. heat pumps), focus on the development and/or adjustment of the model of the cooling coil, mixing section and humidifier, look at the initial costs of the climate control systems, log additional measurement data in order to carry out a good analysis of the air handling unit and the control of its components and investigate other approaches to validate the HAMBASE model and to substantiate the internal sources, especially the method of ‘inverse modeling’.
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1 Introduction

This chapter describes the background, the problem, the objectives, research questions, used methodology and structure of this research.

1.1 Background

Firstly some background information is given about the reason to initiate this research. Secondly some researches that deal with energy in museums are explained. Thirdly a conclusion is stated that is used as a starting point for this thesis.

The Dutch situation of the preservation of museum collections was, up until the late eighties and early nineties, not very good. Several areas of improvement were indicated, like climate conditions, registration, accommodation etc. Therefore, the Deltaplan [1] was started around 1990 to eliminate the arrears of conservation. Based on this report, and led by (strict) national and international guidelines for preservation conditions, museums decided to improve their preservation environment especially by installing Heating, Ventilation and Air conditioning (HVAC) systems to control the indoor climate. No indoor climate specifications were defined in the Deltaplan, so each individual museum defined own climate specifications according to their needs and experiences.

The absence of uniform (Dutch) guidelines of the indoor climate for preservation of museum collections resulted in the development of new guidelines [2] and the use of international guidelines [3]. More recently Ankersmit [4] developed a risk analysis procedure for the Dutch situation, which is currently used as a standard guideline for the indoor climate conditions in the Dutch situation. Often too strict guidelines are used. It must be kept in mind that small bandwidths are not always better, some narrow bandwidth are expensive and impractical or impossible. Standard methods of measurement of relative humidity have uncertainties larger than the specified ranges and some indoor climate conditions cannot even be reached in buildings with a certain construction. The large HVAC systems that accompany these strict guidelines can damage the monumental buildings wherein a museum is situated or can cause a high energy consumption and other problems (condensation, dangerous indoor climate conditions when system failure occurs etc.).

The idea, based on previous researches and experiences, was that the current guidelines might be too strict. Therefore, Martens [5] investigated the degradation risks of museum objects in relation with measured and simulated indoor temperature and relative humidity data. He developed a more risk-based approach, rather than the strict guidelines. This method showed that the risks to the degradation of museum objects in old buildings with a simple installation might not be much different from the risks in adjusted, HVAC planned monuments. This result could possible lead to the adjustment of the guidelines (future), the adjustment of the design of HVAC systems for the museum environment (future) but also to the optimization of HVAC systems in the museum environment (present).

Several researches have been performed to investigated the influences of the (set point) optimization of climate control systems in the museum environment on the energy consumption of the systems.

Ayres et al. [6] defined a building model of modern construction and determined, with computer simulations, the cost sensitivity of operating that structure at various locations around the United States with different baseline environmental parameters. They concluded that a relative humidity control of 50% has the lowest energy consumption compared to 40% or 60%. The widening of the bandwidth of relative humidity did not result in a significant reduction of the energy consumption.
Also the changing of temperature in winter and summer did not result in a significant reduction of the energy consumption. As a result of these conclusions the authors suggest to control the environmental conditions at a cost-effective mid-range level for the less vulnerable museum objects and to use microclimatological solutions for the fine art conservation. It must be noted that this research was conducted approximately twenty years ago and that the results of such a research largely depend on the location and/or region of the used building.

In contrast to the investigation of Ayres et al., Mecklenburg concluded that the widening of the bandwidth of relative humidity did result in a significant reduction of the energy consumption. Mecklenburg concludes that increasing the relative humidity tolerance from ±2% RH to ±7% RH might reduce energy costs by 55% (see Figure 1). New climate specification (21°C ±2°C and 45% RH ±8% RH) were adopted which led to enormous energy savings. In this research measurements were obtained from nine museums of the Smithsonian Museum. All museums of the Smithsonian Museum are located on the west coast of the United States and are exposed to the same humid subtropical climate. The conclusions are drawn based on the different museums. A better approach would have been to look at one museum at the time and see what reduction of energy consumption can be achieved by changing the bandwidth for relative humidity fluctuation.

![Figure 1. Extracted from [7]: Smithsonian FY 1993 Energy Costs Correlated to Relative Humidity Control (source: unpublished, provided to Artigas by Marion Mecklenburg).](image)

Also Artigas [7] concluded, based on measurements, that there is an exponential relationship between the energy costs for climate management and the level of control; as the variance of the indoor conditions increases, the costs decrease exponentially. He also concluded that the energy costs are not only determined by the level of control but also by the type of system and the type of energy used, that maintaining constant indoor climate conditions causes the most stress on the climate control system during the mixed season (spring and autumn), and the least stress on the system during the heating season and that controlling the indoor climate for the comfort of the visitors results in less control of the indoor conditions and higher energy costs. Disadvantages of Artigas’ research are that the research is performed for only five museums in the United States of America, several assumptions are made and no practical changes to the climate control systems are implemented to investigate the effect on the energy consumptions and costs. All buildings have different constructions and climate control systems.
Not only fluctuations (bandwidths) of temperature and relative humidity are important, but also seasonal fluctuation can play an important role in optimizing climate control systems to reduce the energy consumption. Ascione et al. [8] used computer simulations to investigated various strategies used to reduce energy requirements for climate control systems in an exhibition room of a modern museum located in Rome, Italy. This was done for four different system configurations. When an indoor temperature set point that wasn’t fixed but fluctuated with the season was applied (e.g. 21°C in winter, 23°C in summer), energy saving ranging between 6% and 13% were achieved. By applying a wider range of relative humidity control (50% ± 2% to 50% ±10%) an energy saving of 40% was accomplished. In another research, Ascione et al. [9] found that, by allowing a lower winter temperature (20°C instead of 22°C) and a higher summer temperature (26°C instead of 24°C) an energy saving of 20% in winter, more than 40% in the intermediate season and 11% in summer can be achieved without exceeding the thermal comfort boundary. Results were obtained by computer simulations of a historical building located in Benevento (Southern Italy).

As part of his PhD thesis, also Martens [5] investigated the influence of set points (and bandwidths) of temperature and relative humidity on the indoor climate and the energy use. He uses a fictive building and distinguishes four qualities of envelope, and four levels of control. Based on simulations, he concludes that lowering the annual mean temperature set point by 1°C saves energy and increases the object’s lifetime (not in well insulated buildings with higher internal loads). And risks can be reduced and energy saved by changing set points to follow the outdoor climate, because the differences between indoor and outdoor climate are reduced. Therefore, the most optimal climate guidelines in the Dutch situation are ASHRAE B (in unchanged buildings) and ASHRAE As (in new or improved monumental buildings). He also found that a larger bandwidth reduces the amount of energy needed and that the ventilation rate of a building plays an important role on energy use.

Based on the literature above one can conclude that many authors have stated that allowing a wider range of indoor climate conditions and allowing a temperature set point that fluctuates with the season will result in lower energy costs, but only Martens projected this hypothesis on the Dutch situation. However, he only used a simplified fictive simulation model to draw conclusions and did not substantiate this with measurement data. Therefore, this lack of research and the (bad) functioning of (oversized) climate systems and the associated high energy consumption are used as a starting point for this thesis to investigate opportunities for energy conservation in Dutch museums with preservation of collections, building and comfort for visitors and staff.
1.2 Problem description

The Delta Plan, that was started around 1990 to eliminate the arrears of conservation, has largely ended up in museums that decided to improve their indoor environment. Led by national and international (strict) guidelines, (large) systems were purchased to control the indoor climate in the museums. That this is not always a good solution is evidenced by the many damaged monuments (ducts, pipes, attics and basements full of equipment and building physical problems such as condensation and rot), the not always optimal functioning of the system (possible extra risk) and the high energy consumption. Moreover, local microclimates make sure the climate is not everywhere equal to the previously desired climate. Instead of providing additional safety, in practice the systems often provide extra trouble.

Thanks to the participation of a large number of Dutch museums it became possible to do a PhD research [5] on the relationship between buildings, climate control and preservation of collections. A new method of risk analysis, rather than strict guidelines, shows that the risks to the degradation of museum objects in old buildings with a simple installation might not be much different from the risks in adjusted, HVAC planned monuments. Perhaps the old, strict guidelines can be adjusted to make way for this more risk-based approach. This is also the way to act on indoor climates in monuments in a minimal way, in order to better preserve the building and lower the energy use, without causing more (or often even less) risk to the collection.

The (bad) functioning of (oversized) climate systems and the associated high energy consumption are a trigger for further study opportunities for energy conservation in Dutch museums with preservation of collections, building and comfort of visitors and staff [10].

1.3 Project objective

The main research question of this thesis is: “How to optimize climate control systems for museums to create a (better) indoor environment with less energy consumption and less impact on the collections and the building?”

The objective of this research is to give insight in the optimization of climate systems, especially of an air handling unit of the Hermitage Amsterdam. The research should clarify whether the optimization could lead to less energy consumption without a decrease of the conservation conditions of cultural objects and the comfort for staff and visitors of the museum. It is investigated if it is possible to reproduced the current indoor climate by making use of modeling software (MatLab(HAMBASE)/Simulink).

To achieve the objective of the research four sub research questions were formulated:

1. Could the optimization of T/RH points lead to less energy consumption, while maintaining the right indoor climate conditions for the preservation of the collection and the thermal comfort of visitors and staff?
2. Which optimization/combination of the temperature/relative humidity set points is the best according to energy consumption, conservation of cultural objects and comfort of staff/visitors?
3. Is it possible to reproduce the current indoor climate and air handling unit by making use of modeling software?
4. What are the issues, when modeling the indoor climate and air handling unit, that deserve more attention in future research?
1.4  Project approach
The method used to perform this research has been divided into several parts with each its specific activities. The used methodology is visualized in Figure 2. The contents of each part are described below.

"Optimizing climate control systems for museums: A case study for the Hermitage Amsterdam"

![Diagram of methodology](image)

Figure 2. Ball scheme of used methodology.

In the introduction part of the research the problem description is defined as well as the problem objective and approach. Based on this, a research proposal is written.

A literature review was performed to gain insight in the knowledge that is available about the research subject. This review gained insight in the preconditions (indoor climate, degradation principles and risk assessment) that must be taken into account when optimizing a climate control system of a museum. Furthermore, information about the climate systems, principles and control strategies that are used in museums is collected. Successively, literature about building performance simulation software and how this software can be used to simulate and optimize a climate system of a museum is reviewed. Finally, the aspects related to energy and museum are pointed out. Especially the relation between temperature and relative humidity set points/bandwidths and energy consumption.

Information about the Hermitage Amsterdam is collected in order to make it possible to model the building and climate systems of this museum. General information is collected by studying literature and visit the museum. Also measurement data, for the validation of the models, is collected by visiting the museum. Architectural/mechanical drawings and engineering specifications are used to gather information about the dimensions, structure and building physical aspects of the building as well as data about the climate control system and its parameters.
In the fourth part of the research, based on the information gathered in the previous part of the research, a HAMBASE model [11] of the building of the Hermitage Amsterdam is made. This model simulates the heat and vapour flows within multi zone buildings. This model is validated based on energy consumption, calculated from measurement data. The results of this validation are analyzed and the HAMBASE model is adjusted in order to approach the real situation. The next part of the research, the dynamic simulation model of the climate control systems, especially the heating coil, is constructed in the software environment of MatLab/ Simulink. The chosen in- and output structure of the model makes it possible to connect the model to a building simulation model. The theoretical model of the heating coil is described and implemented in an S-function. The model is validated on a static situation and the results are presented.

After that, the results of the research are defined. These results are divided into two parts; an energy part and a control part. Set point optimizations (temperature, relative humidity) of the HAMBASE model are used to obtain results for the energy part. The results of the energy part are analyzed based on energy consumption, risk assessment of the collection and thermal comfort and conclusions are drawn. By using the Simulink model of the heating coil, results are obtained for the control part of the research. The control of the heating coil is modeled, analyzed and conclusions are drawn. An alternative approach of controlling the heating coil is defined and the results and recommendations are pointed out. Finally, these results are combined and discussed. Based on this discussion, recommendations are made.

1.5 Structure of the report

Based on the methodology mentioned in paragraph 1.4, the report starts in chapter 1 with an introduction of the research subject. First the most important results of the literature review, performed before this research, are defined. After that the problem description, project objective and approach are described. Finally, the structure of the report is pointed out. Chapter 2 describes the case study used in this research. Firstly, a general description of the museum is given. Secondly, the climate specifications for the museum are summarized. After that the building structure and building services are analyzed. Finally, the building management system and the current situation are illustrated. Successively, chapter 3 describes the modeling of the building and systems simulation model. In the first paragraph the used simulation software is clarified. In the next paragraph the climate data used in the simulation model is explained. Next, the modeling of the building and heating coil is described. Chapter 4 deals with the results of the research. The results are divided into two parts, an energy part which elaborates on the possible energy savings based on the set point optimizations (temperature, relative humidity) of the HAMBASE model and a control part which describes some possibilities to model the control of the heating coil and the accompanying difficulties. Chapter 5 starts with the discussion which describes some general comments and the research questions of this research are answered. Successively, the conclusions are formulated and the report ends with recommendations for improvement or further research.
2 Case study: Hermitage Amsterdam

In this research, a case study is used to make a model of a museum building and climate control systems to gain insight in the possibilities of customizing the climate systems, control systems and strategies to achieve energy savings. Firstly, a general description of the museum is given. After that the building structure and building services are analyzed. Finally the climate specifications for the museum are summarized and the building management system is illustrated.

2.1 General

In this paragraph, general information is given about the Hermitage Amsterdam. The history of the museum and building, the location of the building and the collection of the museum are pointed out. Paragraph 2.3 and 2.4 elaborate on the building structure and services.

2.1.1 History

The State Hermitage Museum, since 1917 the official name of the museum, is located in St. Petersburg, Russia, and currently owns approximately three million items. Obviously, the museum is too small to house all these artworks. Therefore, in the early nineties of the 20th century The State Hermitage Museum explores the possibilities for an external branch of the museum in Western Europe. During the organization of exhibitions from The State Hermitage Museum in the Nieuwe Kerk in Amsterdam, the idea of a permanent branch of the Russian Museum in Amsterdam was created [12].

In the same period, the foundation ‘Nursing home Amstelhof’ notes that the ‘Amstelhof’ building no longer meets the requirements of modern care. It was decided to give the monumental building ‘Amstelhof’ a cultural function and it was restored and renovated to Russian cultural center and exhibition hall for the Hermitage.

The building, probably designed by city architect Hans Jansz Petersom, was built in 1683. The original name of the building is “Diaconie Oude Vrouwen Huys”. In the year 1953 it was named ‘Amstelhof’. The building was for 324 years a home for elderly. Characteristic for the building is the classical facade on the Amstel side of the building. The building is simple in design and symmetrical. Over the centuries a lot of changes were made to the building. Additional buildings were built (‘Neerlandia’) and the layout was modified. Technical adaptations found their way to the ‘Amstelhof’ such as central heating in 1860. The building has been substantially renovated in 1970, when the ‘Amstelhof’ transformed into a modern nursing home. The rear wing was demolished, including all extensions, to make a new main entrance of the building. The recent restoration dates from 2007-2009.

Changing the nursing home in a museum was done in two phases. Phase I was executed in 2004. At that moment the building ‘Neerlandia’ was changed into a museum. The public could get acquainted with art from the collections of The State Hermitage Museum. This part of the museum is now completely intended for children: Hermitage for Children. Because this ‘test’ was very successful phase II was started. During Phase II, the most recent restoration (2007-2009), the whole ‘Amstelhof’ building was changed into a modernized museum.
2.1.2 Location

The Hermitage Amsterdam is located in the center of Amsterdam. East of the Amstel street, between the Nieuwe Herengracht, Keizersgracht and the Nieuwe Weesperstraat. Below, the location can be seen. In the left corner the Amstel street is situated, the top canal is the Nieuwe Herengracht, the bottom canal is the Keizersgracht and the street in the right corner is the Nieuwe Weesperstraat.

![Figure 3. Location Hermitage Amsterdam [13].](image)

2.1.3 Collection

In paragraph 2.1.1 was already pointed out that the Hermitage Amsterdam is an external branch of The State Hermitage Museum in St. Petersburg, Russia. Therefore, the museum has no collection of its own. All exhibited artworks are on loan from The State Hermitage Museum in St. Petersburg, Russia. There are several exhibitions a year organized, so the collection varies. Exhibitions that are held conclude: “Het Van Gogh Museum in de Hermitage Amsterdam”, “Rubens, Van Dyck & Jordaens”, “Matisse tot Malevich”, “St.-Petersburg in beeld”, “Art Nouveau”, “Pelgrimschatten”, “Venezia!” and “Nicolaas & Alexandra”.

The collection of The State Hermitage Museum in St. Petersburg consists of several parts: Western European art, Oriental art, Russian culture, antique art, Eastern and Siberian archeology, antique weapons and armor (the arsenal) and a thousands of publications an books (the library) [13].

The Western European art is the most important and largest single department with more than twenty paintings by Rembrandt, impressionist and post-impressionist works, paintings by Da Vinci, Raphael, Titian, Giorgione and the finest collection of French art outside the Louvre in Paris.

The other collections conclude: Sasanian silver collection, Coptic textiles, Persian carpets, steel furniture, Roman sculptures and portraits, coins and medals, antique weapons and armor, rare books and manuscripts.
2.2 Climate specifications

The climate specifications for The Hermitage Amsterdam are drawn from the architectural/mechanical engineering specifications [12]. The specifications are shown in Table 1.

<table>
<thead>
<tr>
<th>Climate Specifications</th>
<th>Old</th>
<th>New</th>
<th>Fluctuation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Temperature</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Summer</td>
<td>20 °C±2 °C</td>
<td>21 °C±3/±2 °C</td>
<td>Max. 2 Max. 3</td>
</tr>
<tr>
<td>Winter</td>
<td>20 °C±2 °C</td>
<td>21 °C±3/±2 °C</td>
<td>Max. 2 Max. 3</td>
</tr>
<tr>
<td><strong>Relative humidity</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Paintings, furniture, wooden objects</td>
<td>55 [%]±2 [%]</td>
<td>55 [%]±5 [%]</td>
<td>-</td>
</tr>
<tr>
<td>Mixed collections</td>
<td>45-50 [%]±2 [%]</td>
<td>45-50 [%]±5 [%]</td>
<td>-</td>
</tr>
<tr>
<td>Only metal collections</td>
<td>40-50 [%]±2 [%]</td>
<td>40-50 [%]±5 [%]</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 1. Climate specifications Hermitage Amsterdam.

The column with the old specifications contains values that are extracted from the architectural/mechanical engineering specifications. The column with the new specifications is based on these old specifications but adjusted following an interview with the maintenance coordinator of The Hermitage Amsterdam. The coordinator mentioned that the bandwidth for the fluctuation of the relative humidity is changed from ±2% in the old situation to ±5% in the new situation. However, in the architectural/mechanical engineering specifications was already a bandwidth of ±5% mentioned. So maybe there has been a misunderstanding during the commissioning of the systems or the bandwidth in the engineering specifications is just changed/wrong.

It can be seen that there is no seasonal fluctuation integrated in the set points for temperature and relative humidity. A very strict bandwidth is adopted. Only the set points for relative humidity differ based on the materials of the collections.

2.3 Building structure analysis

In this research, simulation software is used to model the heat and vapour flows in the Hermitage Amsterdam so possible conclusions can be drawn according energy use and savings. To be able to construct a model that practically represents the real situation, an analysis of the building must be made. This analysis is done in this chapter. Firstly, the whole building is described on the basis of several images of the floor plans. Secondly, the part of the building is described where the exhibition area is where this research focuses on. Some additional information is given about the usage of the area.

2.3.1 The building

The building has a square floor plan, with in the middle a courtyard, and consists roughly of four parts, namely: a section on the side of the Amstel, a section parallel to the Nieuwe Herengracht, a section parallel to the Nieuwe Keizersgracht and a section on the side of the backyard. The building consists of five floors (‘souterrain’/ground floor, ‘beletage’/first floor, second floor/attics (technical area), roof). Below, only the first three floors are shown in Figure 4, Figure 5 and Figure 6. It can be seen in these pictures that the sections parallel to the Nieuwe Herengracht and the Nieuwe Keizersgracht are almost identical. The sections are mirror images of each other.
The ‘souterrain’ has several functions which are not intended for exhibition purposes. The building can be entered on street level at point 1 in the image above. By walking through the courtyard (point 7) one reaches the entrance lobby at point 2, from this point the museum visit can be started. Point 3 and 4 represents the Hermitage lobby for kids, which is a separate (and older) part of the museum which will not be used in this research. Point 5 indicates the museum stores and at point 6 the study area is situated. The rooms at the Amstel side (near point 1) house all kinds of supporting functions like, an archive, a storage, a workplace etc.

The ‘beletage’ (Figure 5) can be entered with several stairs, staircases and elevators, mostly situated at the end of the different sections. Point 1 and 2 indicate the exhibition areas.
Point 1 represents the main exhibition areas and point 2 represents the exhibition cabinets. Near point 3 the museum café and restaurant can be found which can be entered by the entrance lobby. The regents and regentesses room (point 4 and 5), the former offices of the board of the house for the elderly (Amstelhof), are located at the Amstel side of the building. The regent room is now intended for ‘Friends of the Hermitage’. The church hall (point 6), for a long time the longest hall of Amsterdam and therefore often used for welcoming important guests, is also situated at the Amstel side of the building. The rooms near point 7 are used for a constant historical exhibition. And again, the Hermitage for kids is indicated by point 8.

The last floor that is pointed out is the first floor. Point 1 represents the auditorium which is used for presentations, lectures, conferences, dinners and receptions. The exhibitions cabinets can also be found here (point 3). It can be seen that the main exhibition area (area between the exhibition cabinets) of the ‘beletage’ covers two floors, so the height of the area is two times the height of a normal area. Also here, the Hermitage for kids is indicated by point 4.

2.3.2 Section: Nieuwe Herengracht

The focus of the research is on the optimization of climate control systems for museums, therefore only the sections parallel to the Nieuwe Herengracht and the Nieuwe Keizersgracht are important because here the exhibition areas are situated. As said before, both sections are almost identical. Therefore, only one of the two sections must be chosen. Because the first measurement data was available for the section parallel to the Nieuwe Herengracht, this section, and especially the main exhibition area in this section, is chosen to be researched. Below, an analysis of the building structure of this section is performed. In Figure 7, a cross-section of the section parallel to the Nieuwe Herengracht (“Herenvleugel”) is shown.
The ‘souterrain’ houses several functions, namely: a museum shop, a storage for the museum shop, a study area, staircases, an elevator, circulation areas and a technical area. All functions are shown in Appendix 2.1. The floor of the ‘souterrain’ is approximately 1 m below ground level. The height of the ‘souterrain’ varies, from approximately 3.1 m in the study area to approximately 2.7 m in the museum shop, technical area and circulation space. The exterior walls of the ‘souterrain’ consists of 0.5 m of solid masonry, 0.17 m of concrete, battens with a variable thickness, 0.1 m mineral wool, a vapour barrier and 0.005 m of painted plaster. A detail of the exterior wall at the side of the Nieuwe Herengracht is shown in Appendix 2.8. The construction varies per location. Each exterior wall contains several windows and doors. The exterior facades and their windows and doors can be seen in Appendix 2.6 and 2.7. The window openings vary in height, shape and material. The exterior doors are made of hardwood. The windows are made of new laminated intrusion resistant glass. The width of the window openings are on each floor almost the same. The less complex interior walls are made of lime stone (thickness: 0.2 m). The more complex interior walls, for instance between the study area and the circulation area are made of a 0.36 m thick masonry wall, a large cavity for technical purposes and construction panels. Some interior walls contain showcases. The ‘souterrain’ is the only floor that has floor heating. The floor consist of a new or existing concrete floor, with a varying thickness of 0.25 till 0.4 m. On top of this concrete, a bonding / smoothing and finishing layer is applied. The top layer depends on the function of the room, the following layers are used: natural stone, screed and a steel slatted floor. The ceilings of the different areas adjoin the exhibition and circulation areas on the ‘beletage’. Some rooms feature a lowered ceiling. The entrance of the museum shop, the main exhibition area, the circulation are and the staircase can be seen in Figure 8.
The ‘beletage’ houses only exhibition functions, except for the circulation areas. The floor plan and the functions are shown in Appendix 2.2. The main exhibition area (B.2.06 ‘Wisselexpositie’) is situated in the middle of the building section. This area covers two floors and has a height of approximately 8.4 m. At one side of the exhibition area (side of the Nieuwe Herengracht) several exhibition cabinets can be found. The opposite side which adjoins the exhibition area is used as a circulation space. The height of the exhibition cabinets is approximately 3.35 m and of the circulation area 3 m. The circulation area features a lowered ceiling. The exterior walls of the ‘beletage’ consists of 0.22 m of solid masonry, battens with a variable thickness, 0.1 m mineral wool, a vapour barrier and 0.005 m of painted plaster on 0.018 plywood. A detail of the exterior wall at the side of the Nieuwe Herengracht is shown in Appendix 2.10. The construction varies per location. Each exterior wall contains several windows and doors. The exterior facades and their windows and doors can be seen in Appendix 2.6 and 2.7. The windows in the exhibition cabinets have window frames and windows with a sun blocking coating. The windows in the circulation area (courtyard side) have also window frames but no coating. The lightweight partition walls in the exhibition cabinets are made of lime stone (thickness: 0.2 m). The more complex interior walls, for instance between the main exhibition area and the exhibition cabinets, are made of (from cabinets to main hall) painted plaster (0.004 m), gypsum fiber board (0.013 m), MDF (0.018 m), a wooden frame and battens, a large cavity for technical purposes (0.38 m), existing masonry (0.22 m), MDF (0.018 m), gypsum fiber board (0.013 m) and painted plaster (0.004 m). The floors of the exhibition cabinets are made of (from bottom to top) existing painted plaster (0.01 m), an existing concrete floor (0.17 m), a topping layer (0.05 m), a leveling adhesive (0.01 m) and 0.02 m of parquet. A detail of the floor can also be seen in Appendix 2.10. The top layer differs, the following layers are used: natural stone and parquet. The construction of the floor in the main exhibition hall is the same is described above, only here natural stone is used as top layer. The ceilings of the different areas adjoin the exhibition areas on the first floor.

The first floor is almost identical to the ‘beletage’. As said before, the main exhibition hall covers two floors. The exhibition cabinets are situated on both long sides of the main hall. From several openings one can glance into the main hall. The floor plan and the functions are shown in Appendix 2.3. The building structure of the exterior and interior walls, the windows, the doors, the floors and the ceilings are the same as described in the part of the ‘beletage’ floor. The ceilings of the exhibition cabinets adjoin the technical areas on the second floor. The ceiling of the main exhibition hall contains a large area with interior sun blinds. Above these sun blinds, a large glass covered area is placed. The total area of the laminated intrusion resistant glass is approximately 140 m². A detail of this glass covered area can be seen in Appendix 2.12.

The second floor is only used as a technical area. The floor plan is shown in Appendix 2.4. The air handling units, control equipment, piping, ducts etc. are located here. This floor can be entered by a staircase and an elevator nearby the Amstel side of the building. The floors are made of existing wooden beams and parts (0.32 m), an existing concrete floor (0.17 m), a vapour barrier and plywood (0.018 m) glued on flameproof insulation (0.1 m). A detail of the floor and a part of the roof can be seen in Appendix 2.9. Some parts of the floor are separated by old masonry walls. No windows can be found on this floor.
2.4 Building services analysis

As can be seen in paragraph 2.3, there are different functions of the rooms in the Hermitage Amsterdam. Each function has its own climate specification and usage profiles. For instance, some spaces are conditioned with floor heating and other spaces are only conditioned with heated, cooled, humidified or dehumidified air. The focus of this research is especially on the museum environment. These exhibition spaces are conditioned with an all-air system. The air is conditioned in air handling units and transported to the exhibition spaces. The building services components such as air ducts, valves, air supply ornaments etc., are integrated into thickened piers or interior walls. All horizontal distribution of air ducts, conduits, etc. are positioned in the attic from where the vertical connections to the exhibition spaces, on the lower building level are constructed.

Because this research is about optimizing HVAC systems in museums, the focus of the modeling and simulation is on an exhibition area called B.2.06 ‘Wisselexpositie’ in the Hermitage Amsterdam. This exhibition area is conditioned by an air handling unit. In this chapter, not the air handling unit itself, but the systems that supply the heat, cold and air to this air handling unit are described. The P&ID schemes of the systems can be found in Appendix 3.2.

2.4.1 Air handling system

The supply (outdoor) air of the whole building comes from four central outdoor air intake openings. The entire air handling system consists of ten air handling units. In the air handling units for the exhibit areas the air is treated by: cooling and dehumidification, humidification by an electrical humidifier, heating, mixing-recirculation and special filtering (against bacteria, spores, fungi, bacteria, harmful gases and particles). By using CO\textsubscript{2} measurements it is determined whether there are people in the exhibition areas and depending on these measurements the required share of outdoor air is determined.

2.4.2 Heating system

The primary energy for heating is a long term energy storage system. Two heat pumps are installed, each with a capacity of 600 kW (heating). The heat pumps extract water from the hot well (15°C) and upgrade this base temperature to a useful temperature (45°C) for the low temperature heating. The generation system is designed for the temperature range 45°C / 35°C. The two heat pumps are connected in parallel to a shorted collector, from where the water is distributed to the heating coils in the air handling units.

2.4.3 Cooling system

Two systems are applied for the production of chilled water, namely: a cold storage system with a capacity of 1200 kW and two heat pumps with a cooling capacity of 480 kW. The cold buffered in the winter is used directly in the high temperature cooling system (12°C / 19°C) for cooling air by means of the cooling coils in the air handling units.

In addition to the high temperature cooling system, a low temperature system is necessary for the dehumidification of the air. For the generation of the required low-temperature cooling water, the two heat pumps are used. In the standard situation, cooling water in the range of 6°C / 12°C is generated by the two heat pumps. This cooling water is used when a relative humidity of 55% in the exhibition areas is desired. If a relative humidity of 45% is desired in the exhibition areas, then the temperature range is lowered to 4°C / 10°C.
The cold which is released from the heat pumps ("residue") is stored in the thermal storage system (soil). This cold water is used for base cooling of the building in the summer period. If the capacity of the heat / cold storage system is insufficient in the summer (i.e. when there is not enough cold energy stored in the winter period), the heat pumps will additionally be used as cooling machines for the low temperature cooling system.

When more cooling than heating is needed, the excess heat is removed. For this purpose, two systems are used, namely: storing additional heat in the thermal storage system (hot well) and storing additional cold by a closed heat exchanger consisting of a boxcooler that is placed in the water of the Amstel. The removed heat is limited so that a minimum supply heating water temperature of 40°C is maintained.
2.5 Climate control system

The climate control systems that are described in chapter 2.4 are the primary systems that should ensure the conditioning of the museum. The control of these systems take place on ‘building level’, like for instance controlling the supply water temperature from the heat pumps to the building. The focus of this research is one room in the museum, which is conditioned by a single air handling unit. The control of this air handling unit and its components takes place on ‘room level’, like for instance controlling the supply air temperature to the exhibition area by comparing the calculated and measured supply air temperature. In this chapter this part of the control will be pointed out.

To be able to describe the control of the air handling unit, and indirect the indoor climate conditions of the exhibition area, information from the building management system of the different components of the air handling unit must be investigated. In Table 2 the functions, master-slave specifications and control settings, extracted from the building management system, of the components of the air handling unit are shown.

<table>
<thead>
<tr>
<th>Component</th>
<th>Function</th>
<th>Control</th>
<th>Control settings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling coil (HT)</td>
<td>Cooling</td>
<td>Measured/Desired</td>
<td>Measured/Desired</td>
</tr>
<tr>
<td></td>
<td>Dehumidification</td>
<td>Capacity LT</td>
<td>measured T_out,HT</td>
</tr>
<tr>
<td>Humidifier</td>
<td>Humidification</td>
<td>Calculated/Measured</td>
<td>Measured RH</td>
</tr>
<tr>
<td></td>
<td>Limitation</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Cooling coil (LT)</td>
<td>-</td>
<td>Measured/Desired</td>
<td>T_out,LT</td>
</tr>
<tr>
<td>Heating coil</td>
<td>-</td>
<td>Measured/Desired</td>
<td>T_dew point</td>
</tr>
</tbody>
</table>

Table 2. Function, master-slave specification and control settings extracted from the building management system of the components of the air handling unit.

In the table above the terms ‘master’, ‘slave’, ‘P-control’ and ‘I-control’ are mentioned. To be able to understand the table and the description of the control in the paragraphs below, a short explanation of these terms is given here.

In Table 2 one can see that most of the components are controlled by a master-slave control. This means that there are two control loops, the master control loop which controls the slow process and the slave control loop which controls the fast process. This principle is shown in Figure 10. The output signal of the master control is not directly used for control of, for instance, a valve, but is used as a set point of the slave control loop.

![Figure 10. Schematic drawing of a master-slave control.](image-url)
“A proportional – integral – derivative controller (PID controller) is a generic control loop feedback controller” [15]. The input ("error" value) of a PID controller is the difference between a measured process variable and a desired set point. The controller attempts to minimize the error by adjusting the process control inputs. In Figure 11 the configuration of a PID controller in Simulink is displayed.

![Figure 11. Configuration PID controller in Simulink.](image)

A PID controller consists of three main parts. The proportional gain, the integral gain and the derivative gain [16] [17]. With these gains and the error described earlier, the control signal is calculated.

The proportional gain multiplies the error by a factor \( K_r \) (P). This gain determines the reaction to the present value of the error. The larger the error, the larger the control signal. So when the system deviates only a bit from the set point, the control does almost nothing to correct it. For example, an error of 5.0 % with a \( K_r \) of 2.0, results in an adjustment of \((2.0 \times 5.0) = 10\%\).

The integral gain multiplies the sum of recent errors by a factor \( K_i \) (I). This gain determines the reaction to the sum of recent (past) errors and its purpose is to avoid stationary deviation from the set point. It ensures that, when a permanent error exists between the desired and measured value, the calculation will be continuously adjusted. If the error value is non-zero for any length of time the control signal gets larger and larger as time goes on. For example, when \( K_r = 10\% \) and \( K_i = 2 \) min., then when the error remains constant, the computation will be adjusted every 2 minutes with 10%. The integral gain can be related to the proportional gain as follows: \( K_i = K_p / \tau_i \), where \( \tau_i \) is the integral time.

To speed up the system responses, the derivative term can be added. The derivative gain multiplies the rate of which the error has been changing by a factor \( K_d \) (D). This gain determines the reaction to the rate of which the error has been changing. The more quickly the error responds, the larger the control effort. The derivative term can be defined as a function of the future values of the error. For example, an error magnification of 1.0 per 10 seconds, with a \( K_r \) of 2.0, will result in an adjustment of 2%. If \( K_d \) is set to 40 seconds, then the controller will once be increased by 4%.

In most HVAC systems it is not necessary to use the derivative part of the PID-controller. This is because most of the processes in a HVAC system are slow processes. A PID control takes the past, present and future values of the error into account in assigning its control value. This is the reason why the controller is widely used, in for instance climate control systems.
In this research MatLab/Simulink is used to simulated the air handling unit and its control. A standard block of a PID controller is integrated in Simulink (Figure 11). Several options can be selected in this block. Two main options are pointed out and explained here, because it are fundamental differences in the theory of the PID controller and therefore, it can have a huge effect on the control signal. It is possible switch between two forms of a PID controller, namely: an ideal PID controller and a parallel PID controller. Below, the formulas of these forms can be seen.

In the ideal form, the proportional gain is applied to the integral and derivative term, which results in:

\[ u(t) = K_p \cdot \left( e(t) + \frac{1}{\tau_i} \cdot \int_0^t e(\tau) \cdot d\tau + \tau_d \cdot \frac{d}{dt} \cdot e(t) \right) \]  

\[ (2.1) \]

Where

\[ K_p \] = proportional value [-]
\[ e \] = error value (depends on parameter to control)
\[ \tau_i \] = integral time [s]
\[ \tau_d \] = derivative time [s]
\[ t \] = time [s]
\[ \tau \] = variable of integration [s]

In the parallel form all terms are added, which results in:

\[ u(t) = K_p \cdot e(t) + K_i \cdot \int_0^t e(\tau) \cdot d\tau + K_d \cdot \frac{d}{dt} \cdot e(t) \]  

\[ (2.2) \]

Where

\[ K_p \] = proportional gain [-]
\[ K_i \] = integral gain [-]
\[ K_d \] = derivative gain [-]
\[ e \] = error value (depends on parameter to control)
\[ t \] = time [s]
\[ \tau \] = variable of integration [s]

The most commonly used form of the PI-controller is the ideal form (equation 2.1) [15]. Here, the parameters have a clear physical meaning. The summation between the brackets determines a new error value which is compensated for future and past errors. The error value in the future, at \( \tau_d \) (seconds/samples), is predicted by the proportional and derivative components. The error value is compensated for the sum of all past errors by the integral component, and tries to eliminate these errors in \( \tau_i \) (seconds/samples). The new error value is scaled by the proportional gain \( K_p \). The parallel form is the basic, and most general and flexible, form of the PID algorithm. Here, the different gains are just added. The parameters in this configuration have the least physical interpretation and this form is generally reserved for theoretical treatment of the PID controller.

Sometimes a PID-controller is restricted. For instance, the position of a two-way valve to control the volume flow of water through a heating coil is restricted to some minimum and maximum value. These minimum and maximum values are called the lower and upper saturation. The two-way valve cannot be ‘more closed’ than 0% and not be ‘more opened’ than 100%. In the Simulink block a saturation option can be activated with a lower and upper saturation limit.
No problems occur when the PID-controller stays within the upper and lower limits of the saturation block. When an integral term is used in the controller, a problem can occur. When the controller exceeds these limits (becomes saturated) the integrator will continue to build up its value for as long as the sign of the error stays the same (positive or negative). When the desired process value has been reached, the controller tries to get out of the saturation but it will fail because of the large accumulated value of the integrator. The integrator value will start to decrease only when the sign of the error changes; this behavior is called integrator wind-up. Integrator wind-up can cause the system to overshoot the desired value up to several times before settling down. It may even cause the system to become unstable [17].

When the saturation option is activated in the Simulink block, no or two anti-windup algorithms can be selected to cope with the wind-up of the integrator when in saturation. These anti-windup mechanisms discharge the integrator when the block is saturated, which occurs when the sum of the block components exceeds the output limits. The anti-windup algorithms are back-calculation and clamping [18]. Instead of these anti-wind algorithms, also a resettable PID-controller can be used. Here, the integrator is being turned off for periods of time (resetting) until the response falls back into an acceptable range.

Back-calculation discharges the integrator when the block output saturates using the integral-gain feedback loop (Figure 12). A value for the Back-calculation coefficient (Kb) can be specified.

![Figure 12. Extracted from [18]: Back-calculation.](image1)

Clamping stops integration when the sum of the block components exceeds the output limits and the integrator output and block input have the same sign. Integration resumes when the sum of the block components exceeds the output limits and the integrator output and block input have opposite signs. The integrator part of the block can be seen in Figure 13.

![Figure 13. Extracted from [18]: Clamping.](image2)
2.5.1 Temperature control

The temperature in the exhibition areas is controlled by a heating coil and a HT cooling coil. Below, a description of the general temperature control and the temperature control of the heating coil and HT cooling coil are given.

The temperature control is based on the calculated and desired supply air temperature. Depending on the difference between the desired and measured temperature in the exhibition area, a supply air temperature is determined (calculated) by using a proportional integral controller. This calculation is limited by a minimum and maximum value. Depending on the temperature control, the heating and cooling coil are controlled in a certain order. The temperature limits, with accompanying time delays, are adjustable per controller. With these settings it is possible to create dead bands between the on and off settings of the controls.

- Heating coil
  Depending on the measured and desired air temperature in the exhibition room, the supply air temperature is determined with a proportional integral controller. In the air handling unit, based on the measured and desired supply air temperature, the position of the actuator of the heating coil or HT cooling coil is determined with a proportional integral controller. When no heating is needed and there is an outdoor temperature of <5°C, the valve of the heating coil will be set at an adjustable minimum position (frost protection).

  The power output of the heating and cooling coil depends on the mass flow of the hot/cold supply water to the coils. The mass flow is varied by an equiprocentual two-way valve. The valve (02-43CV04) of the heating coil is controlled by an actuator, which sets the valve to the calculated position. During operation, the actuator is controlled with a minimum control range. When no heating is needed, the position of the controller is 0% (closed). The calculated position of the actuator is based on the calculated and measured temperature of the supply air and is determined by a proportional integral controller.

  To prevent frost during the startup of the air handling unit, a delayed start-up period is used. The valve of the heating coil is activated when the outdoor temperature and return water temperature of the heating coil are below a certain set point. When the return water temperature of the heating coil is above a certain set point, the system returns to its normal operation control mode. The outdoor air valve remain closed during this time. During the delayed start-up period, the supply air temperature will be set at an increased value for a certain period. To return gradually to the normal operation position after this period, the supply air temperature can be reduced with a tenth grade by an adjustable time period.

- HT cooling coil
  When there is a dehumidification demand, the HT cooling coil is controlled at the desired HT cooling coil air outlet temperature of 15°C. The position of the control valve is determined by a proportional integral controller.

  During the dehumidifying process, the HT cooling coil is controlled based on the supply air temperature measured after the HT cooling coil (02-50TT05). The calculated air temperature after the HT cooling coil is determined by the required capacity of the LT cooling coil (dehumidification).
If the LT cooling coil and the flow control are fully controlled, than the calculated temperature for the HT cooling coil slowly decreases. When the capacity of the LT cooling coil and the flow control are sufficient again, the calculated air temperature slowly increases. At a set time, the calculated value can be increased or decreased by 0.1°C. By means of the actuator, the output (02-43CV02) is sent to the calculated position of the PID-controller.

2.5.2 Humidity control

The humidity in the exhibition areas is controlled by a LT cooling coil (dehumidification) and a steam humidifier. Below, a description of the general humidity control and the humidity control of the LT cooling coil and steam humidifier are given.

The humidity control is based on the desired relative humidity in the exhibition area and the measured relative humidity in the exhibition area. The current value of absolute humidity of the supply air is determined by means of the measured supply air temperature and the measured supply air relative humidity. The current value of absolute humidity of the return air is determined by means of the measured return air temperature and the measured return air relative humidity.

The control of the room air relative humidity consist of a combination of a limited hygrostatic control and humidification/dehumidification. Therefore, also the room air temperature is slightly adjusted. The change of the room air temperature, based on the room air relative humidity, can be up to a maximum of 2K/hour (with a maximum of 3K/day). If a more rapid change is desired, based on the room air relative humidity, than the room air temperature will be adjusted with the maximum permitted speed, and the subsequent action (humidifying or dehumidifying) is released for control.

Depending on the measured and desired humidity in the exhibition area, the systems are activated and controlled by a proportional integral controller in the following order: the steam humidifier, the desired room air temperature (18°C-20°C), the air flow through the low temperature cooling coil (dehumidification) and the room air temperature (20°C-22°C).

- LT cooling coil (dehumidifier)
  Depending on the measured and desired air outlet temperature (dew point temperature), the LT cooling coil is proportional integral controlled. The desired dew point temperature depends on the requirement of relative humidity in the exhibition areas.

  The cooling demand is switched on or off depending on the difference between the measured and desired values. The temperature limits and delay times for activating and deactivating the cooling demand can be adjusted. The absolute humidity is determined based on the temperature (in 0.1°C) and relative humidity (in 0.1%). Depending on the absolute humidity in 0.1 g/kg, the dew point temperature is determined.

  Based on the difference between the desired set point value and measured value (in tenths) a position of the actuator is calculated by means of a PID controller. By means of the actuator, the output (02-43CV03) is sent to the calculated position of the PID-controller.

  In the standard situation, cooling water in the range of 6°C / 12°C is generated by two heat pumps. This cooling water is used when a relative humidity of 55% in the exhibition areas is desired. If a relative humidity of 45% is desired in the exhibition areas, then the temperature range is lowered to 4°C / 10°C.
- **Steam humidifier**

  The humidifier controls the humidity of the air based on the calculated and measured relative humidity in the exhibition area. The limits of the calculated and measured humidity of the air and the accompanying delay times are adjustable.

  The steam humidifier (02-63BV05) is controlled by an actuator, which sets the valve to the calculated percentage of humidity demand. During operation, the actuator is controlled with a minimum control range. When no humidification is needed, the position of the controller is 0% (closed). The calculated position of the actuator is based on the calculated and measured relative humidity of the air in the exhibition area and is determined by a proportional integral controller.

  The limit control prevents condensation in the supply air duct. The maximum desired relative humidity of the supply air can be determined by a set point. The percentage of humidity demand is decreased by means of a PID controller, based on the maximum and measured relatively humidity of the supply air. The limit control is activated when the measured supply air relative humidity is higher than the maximum desired supply air relative humidity. The scheme of the limit control is canceled when the calculated position of the limit control is above the calculated position of the humidity control. The final percentage of humidity demand is determined by the lowest percentage of humidity control and limit control. When the maximum humidistat (02-63MA01) is activated, the humidifier will be turned off immediately.

2.5.3 **Ventilation (Air quality / CO₂) control**

Because mostly the conditions of the air in the exhibition areas must be kept between strict boundaries of temperature and relative humidity, the conditioning of the air causes a high energy consumption. Therefore, most of the supply air in the exhibition areas is recirculated. The amount of outdoor air for ventilation is determined by CO₂ measurements. By using CO₂ measurements, it is determined whether there are people in the exhibition areas and depending on these measurements the required percentage of outdoor air is determined. The proportion of outdoor air is up to 20%, the remaining air is recirculated.

The air quality control regulates the desired and measured CO₂ content in the air. A set point determines the maximum desired CO₂ content in the air. The air quality (02-52QT05) is measured in ppm CO₂. The proportion of the outdoor air is varied based on the number of visitors. For this purpose, depending on the measured and desired maximum load of 800 ppm of CO₂ in the return air, the proportion of outdoor air is proportionally controlled.

The outdoor air valve (02-42CD02) is sent to the calculated position. The final position of the valve is determined by the highest percentage of temperature and air quality control. The air recirculation valve (02-42CD06) is controlled at the opposite state of the outdoor air valve and return air valve (100% minus percentage of outdoor air valve). The return air valve is not controlled by the substation, but is linked to the valve section.

At an outdoor temperature of <7°C, the temperature control switches to complete recirculation. After an adjustable delay (10 minutes), after a minimum return air temperature of 17°C is reached, the air quality control system is released.
2.6 Building management system

The climate systems are controlled by the building management system. The building management system logs a large amount of measurements. These vary from supply air conditions to the position of an air valve. The building management system that is used to store these measurements is Priva [19]. Priva is a building management system for the agricultural, utility and industrial sector. Priva Top Control (TC) is mostly used in the built environment.

Priva Top Control is based on a large, freely configurable, database of control modules for nearly all occurring equipment and systems in a building. The control modules consist of flow diagrams and detailed descriptions which automatically adjust to the chosen configuration. The Hermitage Amsterdam has, as an addition to the standard configuration, an extra option in its building management system called ‘TC History’. This is an application for long-term data storage, which collects selected data online and stores them in an open database. The measurements, that are logged and used in this research, are listed in the table below. The data of the measurements is logged every 16 minutes.

<table>
<thead>
<tr>
<th>Substation</th>
<th>Label – serial number</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outdoor climate</td>
<td>GRFMET – 1</td>
<td>Temperature sensor – 03-75N03</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 2</td>
<td>Relative humidity sensor – 03-75N03</td>
</tr>
<tr>
<td>Indoor climate</td>
<td>GRFSYS – 241</td>
<td>Temperature sensor 26</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 242</td>
<td>Temperature sensor 27</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 243</td>
<td>Temperature sensor 28</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 244</td>
<td>Temperature sensor 29</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 246</td>
<td>Relative humidity sensor 26</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 247</td>
<td>Relative humidity sensor 27</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 248</td>
<td>Relative humidity sensor 28</td>
</tr>
<tr>
<td></td>
<td>GRFSYS – 249</td>
<td>Relative humidity sensor 29</td>
</tr>
<tr>
<td>Air Handling Unit</td>
<td>GRFMET – 1</td>
<td>Air quality – 02-52QT05</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 15</td>
<td>Return air temperature – 02-52TT03</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 17</td>
<td>Return air relative humidity – 02-52MT03</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 20</td>
<td>Air temperature after first cooling coil – 02-50TT05</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 23</td>
<td>Air temperature after second cooling coil – 02-51TT02</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 14</td>
<td>Supply air temperature – 02-51TT04</td>
</tr>
<tr>
<td></td>
<td>GRFMET – 16</td>
<td>Supply air relative humidity – 02-51MT04</td>
</tr>
</tbody>
</table>

Table 3. Used measurement data from Priva TC History.

The sensors that are used in the climate control systems (in ducts etc.) are manufactured by Catec [20]. The sensors that are used to measure the room temperature and relative humidity are manufactured by Hanwell [21]. These sensors communicate with the building management system. The accuracies of the room temperature and room humidity sensors are determined by the manufacturer and can be found in Appendix 10.1 and 10.2. After some time, the measurement equipment has to be calibrated. Information about the calibration of both brands of the sensors is not available.
The building management system contains also control settings. These settings can be found in Appendix 7.2 and 7.3. The relevant control settings of the different components can be used as parameters in the models of the components. The databases of the measurements extracted from Priva can be found in the digital appendix.

2.7 Analysis current situation

In this chapter, the working of the climate control systems is analyzed for the year 2011. Based on measurement data, the heating, cooling, humidification and dehumidification processes of air handling unit 5 (for the conditioning of room B.2.06 ‘Wisselexpositie’) is investigated. The analysis is done for both temperature and relative humidity. Firstly, the analysis is done for one year to be able to see the trends in the conditioning process. Secondly, one week in July is pointed out. Finally, one day in November is analyzed. A description is given of the process and the graphs are discussed. Figure 14 shows the configuration of the air handling unit. The data of the measurement points in the red frames is used in this research.

![Figure 14: Measurement points of the air handling unit (used temperature and humidity loggers in red).](image)

From the graphs below and the graphs in Appendix 11.1 and 11.2, one can conclude that the air temperature and relative humidity is kept most of the time at a very strict level, with a very small bandwidth. This is an advantage when the effect of changing the set point and bandwidth on the energy consumption of the building is investigated, because possibly significant results might be obtained.

From Figure 16, one can conclude that only a small amount of cooling is supplied to the air by the cooling coil (TT05). Though, the dehumidification cooling coil causes a large decrease of temperature (TT05). The heating coil causes only a slight increase of the air temperature (TT04).

In the beginning of the year, the temperature of the indoor air is a bit lower than the rest of the year. This is probably because of the fact that there was no exposition during that time. It seems that the outdoor conditions do not have any influence on the (indoor) air conditions. This is probably due to the air tight building, the proper insulation of the building and the fact that almost all air is recirculated. Measurement faults occur which must be deleted.
2.7.1 Year

In the figure below, data is shown of the temperature sensors in the air handling unit of the year 2011. Also the outdoor and indoor temperature are displayed, in this way the reaction of the air handling unit on the changing indoor and outdoor air temperature can be investigated. The graphs of the relative and absolute humidity can be found in Appendix 11.1.

The indoor air temperature has a minimum fluctuation around 20°C. This is the same for the return air temperature (TT03). The temperature after the cooling coil (TT05) fluctuates between circa 14 and 20°C. The temperature after the humidifier and dehumidification cooling coil (TT02) deviates between approximately 7 and 20°C. The heating coil (TT04) causes a slight increase of the air temperature (circa 2°C). No real trend can be spotted in the air conditions below. Only in the beginning of the year, the temperature of the indoor air is a bit lower than the rest of the year. This is probably because of the fact that there was no exposition during that time. Also, some weird values for the temperatures during this part of the year can be seen. Also in the beginning of December weird values occur. For this deviations, no other reason than measurements faults can be found. The outdoor temperature doesn’t affect the indoor air temperature and temperature of the air in the air handling unit.

![Figure 15. Outdoor temperature (purple), average temperature in exhibition area (red) and temperature sensors in the air handling unit (blue), plotted for the year 2011 (TT03: T return air, TT05: T air after cooling coil, TT02: T air after dehumidification cooling coil, TT04: T air after heating coil).](image)
2.7.2 Week

In the figure below, data is shown of the temperature sensors in the air handling unit of one week in July 2011. Also the outdoor and indoor temperatures are displayed, in this way the reaction of the air handling unit on the changing indoor and outdoor air temperature can be investigated. The graphs of the relative and absolute humidity can be found in Appendix 11.2.

In Figure 16, a clear distinction can be made between the days of the week. The outdoor temperature fluctuates between approximately 24 and 14°C. It seems that this temperature doesn’t have any influence on the indoor air temperature. This is probably due to the air tight building and the fact that almost all air is recirculated. It can be seen that the return air temperature (TT03) is very constant and is located just above 20°C. The temperature after the cooling coil (TT05) fluctuates more, especially during the day. The average temperature lies around 18°C. The most deviations occur in the temperature after the humidifier and dehumidification cooling coil (TT02). Here, the temperature fluctuates between 5 and 20°C. A lower limit is around 5°C. The temperature rises after the heating coil (TT04) to an average value of approximately 18°C.
2.7.3 Day

In the figure below, data of the temperature sensors in the air handling unit, and the air temperature in the exhibition area, are plotted for one day in November 2011. Also the relative and absolute humidity are displayed. The outdoor conditions are not shown, because these conditions deviate tremendously from the conditions below and it seems that these conditions don’t have any influence on the indoor air conditions at all. The influence of the outdoor temperature and relative humidity is negligible, because of the air tight building and almost all air is recirculated.

It can be seen that during the night, the air temperatures remain almost constant. The supply air temperature (light blue, TT04) has the highest temperature during the night (21 °C). During day time, from 10:00 AM, the supply air temperature is lowered with circa 1 °C. The return air temperature (dark blue, TT03) and the indoor air temperature (purple) stay almost constant, at about 20 °C. During day time, the temperature after the cooling coil (green, TT05) and the temperature after the humidifier and dehumidification cooling coil (red, TT02) fluctuate the most. The lowest temperature can be found after the cooling coil, between 17 and 18.5 °C. This temperature is ‘followed’ by the air temperature after the humidifier and dehumidification cooling coil, but is a bit higher (18.5 °C). The relative humidity of the indoor air remains almost constant (47 %). However, the return air relative humidity decreases during daytime, from circa 45.5 to 44 %. The supply air relative humidity changes in the opposite way, it is increased during daytime, from approximately 43.5 to 45.5 %. Also here, during the night the values remain almost constant. The absolute humidity of the supply air and indoor air are almost the same. These values deviate between 6.8 and 8.0 g/kg. The absolute humidity of the return air is much lower and the deviations are very small.

![Figure 17. Temperature, relative humidity and absolute humidity of the indoor air in the exhibition area and the air in/nearby the air handling unit, plotted for one day in November 2011 (TT03/MT03: T/RH return air, TT05: T air after cooling coil, TT02: T air after dehumidification cooling coil, TT04/MT04: T/RH air after heating coil, X03: absolute humidity return air, X04: absolute humidity supply air).](image-url)
3 Modeling

In this chapter the basic components of the modeling are explained. Paragraph 3.1 describes the simulation software that is used in this research. In paragraph 3.2 the used outdoor and indoor climate data are clarified. In paragraph 3.3 the building model of the case study and the validation are pointed out. After that, a description of the different components in the air handling unit is given in section 3.4.1. Finally, in paragraph 3.4.2 and 3.4.3, the modeling of the heating coil is explained and the validation is illustrated.

3.1 Simulation software

The software that is used in this research, for the modeling of the building and climate control systems, is MatLab/Simulink. HAMBASE \cite{11}, which is a program(-code) that works in the MatLab environment, is used for modeling of the building. HAMBASE is a simulation model for heat and vapor flows in a building which is developed at Eindhoven University of Technology (TU/e). With the model the indoor temperature, the indoor air humidity and energy use for heating and cooling of a multi-zone building can be simulated. For the modeling of the climate control systems Simulink is used. Simulink, which is integrated with MatLab, is an environment for the simulation of dynamic and embedded systems. It provides an interactive graphical environment and a customizable set of block libraries that makes it possible to design, simulate, implement, and test a variety of time-varying systems. Simulink can be coupled to HAMBASE \cite{22}, which enables the coupling of models with different time constants: HVAC components and controllers (order of seconds) and building response (order of hours).
3.2 Climate data

To obtain simulations results for indoor climate and energy consumption of the case study used in this research, the climate data has to be adapted. In this paragraph it is explained how the climate is modeled and implemented in the simulation software HAMBASE. Climate data of the year 2011 is used, because of this year the most complete measurement data was available. The least errors were found in this measurement data.

3.2.1 Outdoor

For simulating the indoor environment of the case study, firstly standard outdoor climate data from the Royal Netherlands Meteorological Institute (KNMI) are used. In the HAMBASE program folder, climate data for the period 1971 until 2010 are available. These climate files contain data from “De Bilt” station of the Royal Netherlands Meteorological Institute (KNMI) and exist of the following data columns; (1) diffuse solar radiation [W/m²], (2) temperature [0.1 °C], (3) direct solar radiation [W/m²], (4) wind speed [0.1 m/s], (5) wind direction(degrees 0-360), (6) relative humidity [%], (7) extra-long wave radiation to the sky dome (not used), (8) precipitation [0.1 mm/h] and (9) cloud coverage (1...8).

Because the Hermitage Amsterdam is located in the center of Amsterdam, the KNMI station that is the closest to this location, “Schiphol” station, is selected. Because the data of this station is not available in the HAMBASE program folder a text file with climate data has to be downloaded from the site of the KNMI and this file must be converted with a MATLAB code (see Appendix 5.9) to create a file which can be used by HAMBASE. These data files contain the same columns as mentioned above.

The “Schiphol” station is located at an airport, in a semi-open landscape. It is mainly surrounded by grass and arable land interspersed by infrastructure / buildings. However, the Hermitage Amsterdam is located in the center of Amsterdam so it is expected that the temperature and relative humidity will deviate from the temperature of “Schiphol” station due to the heat island effect. Therefore the outdoor climate data that is obtained from the building management system is compared to the outdoor climate measured by the “Schiphol” station. Below two graph can be seen were both data sets of the year 2011 are compared.

From Figure 18, one can conclude that a large deviation may exist between the temperature measured at “Schiphol” station and the center of Amsterdam. In the bottom graph of Figure 18 the difference between the outdoor temperature can be seen, here the KNMI data is subtracted from the measurement data on location. The difference can reach values up to 8°C (incidentally 14°C). In Figure 19 the difference in relative humidity is shown. Here the difference can reach values up to 25% (incidentally 35%).

Because of these differences, the temperature and relative humidity of the climate data file of “Schiphol” station is replaced by temperature and relative humidity data measured on the location of the Hermitage Amsterdam, the other climate conditions are not adjusted. The KNMI stores data with time steps of one hour, while the building management system saves data with a time step of eight or sixteen minutes. Therefore, the data of the building management system must be interpolated to create data files with the same time steps. The MATLAB code for this conversion can be found in Appendix 5.10.
Optimizing climate control systems for museums

M.P.E. Maas

Figure 18. Comparison of outdoor temperature from Schipol station (KNMI) and location (measurements).

Figure 19. Comparison of outdoor relative humidity from Schipol station (KNMI) and location (measurements).
3.2.2 Indoor

In paragraph 2.3, the floor plan of the museum already has been pointed out. It can be seen that the building, in principle, consists of two almost identical sections which are used to exhibit art. Alternately, in one of the two sections an exhibition is held. A first set of indoor climate data was made available by the Hermitage Amsterdam of three different exhibitions including the “Matisse to Malevich” exhibition. Below the floor plans are shown that give an overview of the logger positions during this exhibition.

![Figure 20. Location of logger positions during the “Matisse to Malevich” exhibition.](image)

At several positions the temperature and relative humidity is measured; this is done by Hanwell temperature and humidity loggers which are also connected to the building management system. The focus of the simulation of Hermitage Amsterdam is on room B.2.06 ‘Wisseleposisitie’ which is situated in the ‘Herenvleugel’. Therefore, data from the loggers 26, 27, 28 and 29 is used. The first set of indoor climate data missed a lot of data because it contained only the data for the exhibition periods of several months. When the whole database of the building management system became available this data was used to create indoor climate data sets for simulation and validation. A climate data set of average temperature and relative humidity, created from the data of the loggers mentioned above, is made for simulation and validation (Figure 21). To use the measurement data successfully, the sample time of the measurement data is converted to seconds and the ‘errors’ and/or extreme data are removed.
3.2.3 Air Handling Unit

To be able to validate and calibrate the different components of the air handling unit, data from measurement points inside the air handling unit was extracted from the building management system. This data was also useful to understand the principles and working of the air handling unit. The measurement and control points can be seen in chapter 2.7, Figure 14. As said before (paragraph 2.6), certain measurement data is stored in and used by the building management system. The data of the measurement points in the red frames is used in this research; they are also mentioned in Table 3. To use the measurement data successfully the sample time of the measurement data is converted to seconds and the ‘errors’ and/or extreme data are removed.
3.3 Building

The HAMBASE model that is used in this research is pointed out in this chapter. Firstly, the building model and its specifications are described. Finally, the validation of the model is explained.

3.3.1 HAMBASE: Building specifications

The focus of the simulation of Hermitage Amsterdam is on room B.2.06 ‘Wisselexpositie’. The material properties, floor plans, usage of the building etc. were derived from the architectural and mechanical building specifications [12] and some final documents and technical drawings provided by Kuijpers Installaties B.V. Rooms that have more or less the same indoor climate (and function) can be modeled as one zone [11]. Figure 22 shows the floor plan of the building in which the nine zones are indicated and the color, function and description of the zones can be found in Table 4.

<table>
<thead>
<tr>
<th>Zone</th>
<th>Color</th>
<th>Function</th>
<th>Floor</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Purple</td>
<td>Circulation area</td>
<td>Basement</td>
<td>Room: S.2.02, S.2.04, S.2.05, S.2.07, S.2.08, S.2.09, S.2.10, S.2.13</td>
</tr>
<tr>
<td>2</td>
<td>Yellow</td>
<td>Technical area</td>
<td>Basement</td>
<td>Room: S.2.14, S.2.15, S.2.18, S.2.19</td>
</tr>
<tr>
<td>3</td>
<td>Orange</td>
<td>Shop/Study area</td>
<td>Basement</td>
<td>Room: S.2.06, S.2.16, S.2.17</td>
</tr>
<tr>
<td>4</td>
<td>Magenta</td>
<td>Circulation area</td>
<td>‘Beletage’</td>
<td>Room: B.2.01, B.2.02, B.2.03, B.2.04, B.2.07, B.2.08, B.2.09, B.2.10, B.1.02</td>
</tr>
<tr>
<td>5</td>
<td>Green</td>
<td>Exhibition area</td>
<td>‘Beletage’</td>
<td>Room: B.2.05A t/m B.2.05H</td>
</tr>
<tr>
<td>6</td>
<td>Green</td>
<td>Exhibition area</td>
<td>‘Beletage’</td>
<td>Room: B.2.06</td>
</tr>
<tr>
<td>7</td>
<td>Magenta</td>
<td>Circulation area</td>
<td>First floor</td>
<td>Room: 1.2.01, 1.2.03, 1.2.04, 1.2.07, 1.2.08, 1.2.09, 1.2.10, 1.1.02</td>
</tr>
<tr>
<td>8</td>
<td>Green</td>
<td>Exhibition area</td>
<td>First floor</td>
<td>Room: B.2.01A t/m B.2.05H</td>
</tr>
<tr>
<td>9</td>
<td>Green</td>
<td>Exhibition area</td>
<td>First floor</td>
<td>Room: B.2.01A t/m B.2.05I, B.2.06, B.2.13</td>
</tr>
</tbody>
</table>

Table 4. Zones for simulation Hermitage Amsterdam.

Figure 22. The floor plan (with zone numbers), from left to right: basement, ‘beletage’ and first floor.
A detailed description of the zones with characteristics and material use is given in Table 5. The museum is open all week and almost all year. The visiting hours are Monday till Sunday from 09:00 (10:00) in the morning until 17:00 in the afternoon. Only on Christmas Day and Queen’s Day the museum is closed.

<table>
<thead>
<tr>
<th>Zone</th>
<th>Materials and construction characteristics</th>
</tr>
</thead>
</table>
| 1    | - Zone consists of eight adjacent rooms (circulation area, storage) with a total volume of 1322.2m$^3$  
- External wall (masonry, concrete, cavity) with thickness of 0.7m (insulation, construction panels and plaster on the inside)  
- Enclosed with simple internal walls of 0.2m (limestone, plaster), complex internal wall (masonry, cavity, construction panels) and lowered ceiling (concrete, insulation, plaster)  
- Concrete floor (1m below ground level) with insulation, parquet top layer and floor heating, in contact with ground (constant ground temperature)  
- Windows with new laminated intrusion resistant (HR) glass, hardwood external doors and glass/closed internal doors |
| 2    | - Zone consists of four adjacent rooms (technical area) with a total volume of 389.2m$^3$  
- External wall (masonry, concrete, cavity) with thickness of 0.7m (insulation, construction panels and plaster on the inside)  
- Enclosed with simple internal walls of 0.2m (limestone, plaster), complex internal wall (masonry, cavity, construction panels) and lowered ceiling (concrete, plaster)  
- Concrete floor (1m below ground level) with insulation and slatted steel top layer, in contact with ground (constant ground temperature)  
- Windows with new laminated intrusion resistant (HR) glass, hardwood external doors and closed internal doors |
| 3    | - Zone consists of three adjacent rooms (shop, study area, storage) with a total volume of 1423m$^3$  
- External wall (masonry, concrete, cavity) with thickness of 0.7m (insulation, construction panels, plaster on the inside)  
- Enclosed with simple internal walls of 0.2m (limestone, plaster), complex internal wall (masonry, cavity, construction panels) and lowered ceiling (concrete, plaster)  
- Concrete floor (1m below ground level) with insulation, natural stone/parquet top layer and floor heating, in contact with ground (constant ground temperature)  
- Windows with new laminated intrusion resistant (HR) glass, hardwood external doors and glass/closed internal doors |
| 4    | - Zone consists of nine adjacent rooms (circulation area, storage) with a total volume of 1875.8m$^3$  
- External wall (masonry, concrete, cavity) with thickness of 0.35m (insulation, construction panels and plaster on the inside)  
- Enclosed with simple internal walls of 0.2m (limestone, plaster), complex internal wall (masonry, cavity, construction panels) and lowered ceiling (concrete, plaster)  
- Floor (concrete, finishing layer, parquet top layer) which is adjacent to circulation area zone 1  
- Windows with window frames (no sun shading) and internal glass/closed doors |
| 5    | - Zone consists of eight adjacent rooms (exhibition cabinets) with a total volume of 624.8m$^3$  
- External wall (masonry, concrete, cavity) with thickness of 0.35m (insulation, construction panels and plaster on the inside)  
- Enclosed with simple internal walls of 0.2m (limestone, plaster), complex internal walls of 0.7m (plaster, gypsum fiber board, MDF, a wooden frame and battens, cavity, masonry, MDF, gypsum fiber board, plaster) and ceiling (concrete, plaster)  
- Floor consists of concrete, finishing layer and parquet top layer, which is adjacent to zone 2 and 3  
- Windows with window frames and a sun blocking coating |
| 6    | - Zone consists of one big room (‘Wisselexpositie’), covers two floors, with a volume of 2617.6m$^3$  
- Only enclosed by internal walls: simple internal walls of 0.2m (limestone, plaster), complex internal walls (plaster, gypsum fiber board, MDF, a wooden frame and battens, cavity, masonry, MDF, gypsum fiber board, plaster) with a total thickness of 0.7m  
- Floor consists of concrete, finishing layer and natural stone top layer, which is adjacent to zone 2 and 3  
- Ceiling contains a large glass (laminated intrusion resistant glass) area of 140 m$^2$, which is adjacent to
the outdoor conditions. Other ceiling (concrete, plaster) parts are adjacent to technical area.

- Zone consists of eight adjacent rooms (circulation area, storage) with a total volume of 899.1m$^3$
- External wall (masonry, concrete, cavity) with thickness of 0.35m (insulation, construction panels and plaster on the inside)
- Enclosed with simple internal walls 0.2m (limestone, plaster), complex internal wall 0.7m (masonry, cavity, construction panels) and lowered ceiling (concrete, plaster)
- Floor (concrete, finishing layer, parquet top layer) which is adjacent to circulation area zone 4
- Windows with window frames (no sun shading) and internal glass/closed doors

- Zone consists of eight adjacent rooms (exhibition cabinets) with a total volume of 627.8m$^3$
- External wall (masonry, concrete, cavity) with thickness of 0.35m (insulation, construction panels and plaster on the inside)
- Enclosed with simple internal walls 0.2m (limestone, plaster), complex internal walls 0.7m (plaster, gypsum fiber board, MDF, a wooden frame and battens, cavity, masonry, MDF, gypsum fiber board, plaster) and ceiling (concrete, plaster)
- Floor consists of concrete, finishing layer and parquet top layer, which is adjacent to zone 5
- Windows with window frames and a sun blocking coating

- Zone consists of eleven adjacent rooms (exhibition cabinets) with a total volume of 1082.8m$^3$
- External wall (masonry, concrete, cavity) with thickness of 0.35m (insulation, construction panels and plaster on the inside)
- Enclosed with simple internal walls 0.2m (limestone, plaster), complex internal walls 0.7m (plaster, gypsum fiber board, MDF, a wooden frame and battens, cavity, masonry, MDF, gypsum fiber board, plaster) and ceiling (concrete, plaster)
- Floor consists of concrete, finishing layer and parquet top layer, which is adjacent to zone 4
- Windows with window frames and a sun blocking coating

Table 5. Description of materials and construction characteristics of the zones.

3.3.2 Validation

The developed HAMBASE model must be validated/calibrated to make sure that the model matches the real situation. The rooms are conditioned (heated/cooled/humidified/dehumidified), therefore the building cannot be simulated free floating. The capacities for heating, cooling, humidification and dehumidification are derived from the revision drawings of the principle of the air handling system and heating/cooling system. The capacities are in proportion to the amount of air that is supplied to the zones. Table 6 shows the capacities for heating, cooling, humidification and dehumidification, as well as the amount of air that is supplied to the zones. The used MatLab-file of the HAMBASE model is included in Appendix 4.

<table>
<thead>
<tr>
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<td>1</td>
<td>3.600</td>
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<td>3.839</td>
<td>0.00107</td>
<td>0.00336</td>
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<tr>
<td>2</td>
<td>2.000</td>
<td>-</td>
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<td>-</td>
<td>-</td>
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<tr>
<td>3</td>
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<td>34.085</td>
<td>20.367</td>
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<td>0.00317</td>
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<td>4</td>
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<td>16.749</td>
<td>0.00502</td>
<td>0.01359</td>
</tr>
<tr>
<td>5</td>
<td>6.400</td>
<td>39.749</td>
<td>16.378</td>
<td>0.00428</td>
<td>0.01345</td>
</tr>
<tr>
<td>6</td>
<td>17.160</td>
<td>100.240</td>
<td>42.510</td>
<td>0.01111</td>
<td>0.03392</td>
</tr>
<tr>
<td>7</td>
<td>2.100</td>
<td>3.727</td>
<td>1.535</td>
<td>0.00042</td>
<td>0.00126</td>
</tr>
<tr>
<td>8</td>
<td>5.900</td>
<td>36.644</td>
<td>15.098</td>
<td>0.00395</td>
<td>0.01240</td>
</tr>
<tr>
<td>9</td>
<td>7.800</td>
<td>42.354</td>
<td>17.451</td>
<td>0.00530</td>
<td>0.01414</td>
</tr>
</tbody>
</table>

Table 6. Capacities for heating, cooling, humidification, dehumidification and airflow for nine zones of the HAMBASE model.
The determined capacities for heating, cooling, humidification and dehumidification are of such dimension that the set point is almost always achieved, so it doesn’t fluctuate around a set point. In this way, the simulated data cannot be compared with the measured data. Therefore, the HAMBASE model is validated/calibrated based on the energy use, the approach that is used in this research is described below and is shown in Figure 23.

![Figure 23. Schematic overview of approach used for validating the building model.](image)

Firstly, a free floating model of zone 6 of the original HAMBASE model was made (so not the complete building free floating). This was done by setting the capacities for heating, cooling, humidification and dehumidification to zero and by removing the set points for heating, cooling, humidification and dehumidification. After that, the actual capacities determined from the measurements of the outdoor temperature/relative humidity, indoor temperature/relative humidity and the supply air temperature/relative humidity were supplied to the zone by overruling the values for $Q_{\text{int}}$ and $G_{\text{int}}$ in HAMBASE. An energy and moisture balance was used in Simulink to determine the actual capacities. The MatLab-code which is used to overrule the values for $Q_{\text{int}}$ and $G_{\text{int}}$ can be found in Appendix 5.3.

Most of the air in the exhibition area is recirculated, only a small amount of air is supplied from the outdoor. According to data from the building management system, a minimum of 1200 m$^3$/h is supplied from the outdoor when fresh air is needed based on the air quality control (section 2.5.3). This amount of fresh air is used to calculate the energy needed to heat or cool the outdoor air to room conditions. This amount of outdoor air is also used in HAMBASE to indicate the ventilation rate of the zone (1200 m$^3$/h divided by the volume of the zone). Also for the other zones the minimum amount of fresh air for the ventilation rate is used, here the minimum amount of fresh air is 1000 m$^3$/h. In reality this amount of fresh air is not constant, it fluctuates between 1200 m$^3$/h (1000 m$^3$/h) and 3750 m$^3$/h. This may explain deviations in the results.

The measured and simulated temperatures are plotted in Figure 24 (this is also done for the relative humidity, these graphs can be found in Appendix 6.1). In this graph can be seen that the simulated values follow the same trend but are situated lower. Because the deviation is not seasonal, in reality there is more or less energy added to the zone throughout the year. A large effect on the calculated energy supplied to the zone can be expected when the supply air temperature is slightly higher, e.g. by heating after the measurement, or a deviation in temperature sensors. Therefore, a correction is included in the red rectangles to compensate this deviation (see Figure 27 and Figure 27). A correction of 1.05 °C is included in the determination of the sensible energy and a correction of 0.175 °C is included in the determination of the latent energy to achieve the same averages in temperature and relative humidity as in the measurement data. A smaller ‘energy shift’ is used when $G_{\text{int}}$ is calculated because this is actually the case with a temperature rise in the air ducts or when a too low temperature is measured, and the relative humidity is calculated based on measured temperature and absolute humidity, which will result in a too high measurement of the relative humidity.
Something strange happens at the end of the year, apparently there is a lot more energy lost because there is a lot more energy supplied and still the indoor temperature is lower than normal. No reason for this deviation is found.

![Figure 24. Measured and simulated temperature of the exhibition area, without temperature correction.](image)

After the correction, the measured and simulated temperature show the same yearly averages, as can be seen in Figure 25. This is also done for the relative humidity, these graphs can be found in Appendix 6.2.

![Figure 25. Measured and simulated temperature of the exhibition area, with temperature correction.](image)
Optimizing climate control systems for museums

Figure 26. Simulink model to calculate the amount of energy for heating and cooling of the ventilation and recirculation air.

Figure 27. Simulink model to calculate the amount of energy for humidification and dehumidification of the ventilation and recirculation air.
Based on the Simulink models of Figure 27 and Figure 27, the amount of energy for ventilation and recirculation air is calculated, as well as the total amount of moisture supplied to the zone. MatLab is used to convert the results of the Simulink models to determine the total amount of energy needed for heating, cooling, humidification and dehumidification. The MatLab-code which is used for this conversion can be found in Appendix 5.12.

Finally, the original HAMBASE model is restored by implementing the capacities again. The measurements of the indoor temperature and relative humidity were used as the actual set points for the zone. They were supplied to the zone by overruling the values for $T_{set,min}$, $T_{set,max}$, RV$_{min}$ and RV$_{max}$ in HAMBASE. The MatLab-code which is used to overrule these values can be found in Appendix 5.3. Now, the amount of energy needed for heating, cooling, humidification and dehumidification is simulated with HAMBASE and compared with the actual values, calculated with Simulink. The results of this comparison are shown in Table 7.

<table>
<thead>
<tr>
<th>Hermitage Amsterdam: “Grote Zaal (Wisselexpositie) + Souterrain”</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>Heating</td>
</tr>
<tr>
<td>Cooling</td>
</tr>
<tr>
<td>Humidification</td>
</tr>
<tr>
<td>Dehumidification</td>
</tr>
<tr>
<td>Total</td>
</tr>
</tbody>
</table>

Table 7. Comparison of simulated and calculated energy consumption for the year 2011.

As can be seen in Table 7, the HAMBASE model matches the Simulink calculations based on measurement data quite good for the amount of cooling energy consumption. However, major deviations exist between the amount of energy needed for heating, (de-)humidification and total energy. Therefore, the ventilation rate in HAMBASE is slightly increased (infiltration) to achieve values that are more in accordance with the Simulink calculations. Some remarks must be made. No internal heat/moisture sources, not in HAMBASE and not in Simulink, are taken into account. As said before, the inconstant amount of outdoor air in reality may explain deviations in the results.

Various values for infiltration (increased ventilation rate) are investigated. The new results, based on a slightly increases ventilation rate in HAMBASE, are shown in Table 8. The value for the ventilation rate in HAMBASE is now 0.75 [1/h], this value represents an airflow of approximately 1960 m$^3$/h. Compared to the previous minimal ventilation rate of 0.46 (= 1200 m$^3$/h), this results in a difference (=infiltration) of circa 760 m$^3$/h. It can be seen that still large deviations exist. As said before, these deviations can be caused by many uncertainties in the model. One of these uncertainties, the internal sources, must be eliminated or at least be substantiated.
Below, a method is described to be able to substantiate the internal heat and moisture sources. A schematic overview of the method is shown in Figure 28. Firstly, the energy use for heating and cooling \( Q \) and humidification and dehumidification \( G \) of the HAMBASE model is plotted. Here, the HAMBASE model without internal sources and with set points for heating, cooling, humidification and dehumidification based on measurement data is used. Secondly, the energy use calculated with Simulink and based on measurement data is plotted. These are the same energy consumptions as used in Table 7, only here the hourly data of one year is plotted. Thirdly, the difference between the simulated and calculation energy use is determined and plotted and the averages of both energy consumptions is calculated. Based on these averages, the internal sources of the building are determined. The steps explained above are performed for three situations, namely: for the total energy use, for the energy use during visiting hours and for the energy use during closing hours.

The differences between the simulated and calculation energy use and the averages of both energy consumptions of the three situations are shown in Appendix 6.3, 6.4 and 6.5. The average energy use for heating, cooling, humidification en dehumidification are used to determine the internal sources. These internal sources are implemented in HAMBASE to achieve better simulation results. The implemented internal sources can be found in Table 9. These internal sources must be interpreted in another way. Negative internal sources mean, for example, that there was less moisture add to the building then was adopted and/or simulated.
However, this substantiation of the internal sources can be discussed. Differences can occur due to unknown internal sources. When, for instance, the energy consumption for humidification in Table 8 is examined one can see that the calculated (measured) value is larger than the simulated value. This difference could possibly be caused by the internal sources that are, or are not, present in the exhibition area. When the graphs and values in Appendix 6.3, 6.4 and 6.5 are examined, then one could also conclude that, at some moment the values for heating/cooling/humidification/dehumidification are undue and that they must be corrected with the internal sources described here.

<table>
<thead>
<tr>
<th>Visiting hours</th>
<th>( Q_{\text{int}} ) [W]</th>
<th>( G_{\text{int}} ) [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Closing hours</td>
<td>-90</td>
<td>-0.000079</td>
</tr>
<tr>
<td>Total</td>
<td>520</td>
<td>-0.000219</td>
</tr>
</tbody>
</table>

Table 9. Calculated averages for internal sources.

The adjusted HAMBASE model (with the inserted internal sources) results in a better simulated energy use as can be seen in Table 10. From this table, one can conclude that the model matches the real situation quite good and that it can be used to obtain the results of this research.

<table>
<thead>
<tr>
<th>Hermitage Amsterdam: “Grote Zaal (Wisselexpositie) + Souterrain”</th>
<th>HAMBASE [MWh] vv=0.75</th>
<th>Simulink [MWh] + energy shift ( h = 2491 ) [kJ/kg]</th>
<th>( \Delta E ) [MWh]</th>
<th>Deviation of real situation [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating</td>
<td>63.21</td>
<td>62.27</td>
<td>0.94</td>
<td>\approx 2</td>
</tr>
<tr>
<td>Cooling</td>
<td>23.96</td>
<td>23.92</td>
<td>0.04</td>
<td>\approx 0</td>
</tr>
<tr>
<td>Humidification</td>
<td>12.57</td>
<td>13.84</td>
<td>-1.27</td>
<td>\approx -9</td>
</tr>
<tr>
<td>Dehumidification</td>
<td>14.26</td>
<td>15.50</td>
<td>-1.24</td>
<td>\approx -8</td>
</tr>
<tr>
<td>Total</td>
<td>114.0</td>
<td>115.5</td>
<td>-1.5</td>
<td>\approx -1</td>
</tr>
</tbody>
</table>

Table 10. Comparison of simulated and calculated energy consumption for the year 2011, improved model (internal sources added).

However, some small deviations still exist between the amount of energy needed for heating, humidification and dehumidification, probably due to unknown sources, such as the exact amount of visitors, open doors, flow of air to other zones etc. and invalid data to validate based on (relative/absolute) humidity. It might be not possible to further optimize the HAMBASE model. These deviations can be isolated, and can be taken into account when simulating the model with, for instance changing set points.

As already mentioned in this section of the report, the substantiation of the internal sources can be discussed. Another possible approach to substantiate these sources could be the method of ‘inverse modeling’. Kramer [24] investigated this method and concluded that this approach is very promising for the prediction and characterization of indoor climates and energy performances. Therefore, this might be a proper method to validate the HAMBASE model and to substantiate the internal sources.
3.4 Climate systems: air handling unit

To determine whether optimization could lead to less energy consumption and a better indoor environment, the climate systems must be taken into account. By describing and modeling the components, the possibilities for optimization become clear. Firstly, the different components in the air handling unit are described, secondly one of the components (heating coil) is modeled and validated.

3.4.1 Description of components

The air handling unit consists of four basic components to condition the air: a cooling coil, a humidifier, a dehumidifier and a heating coil. In this paragraph, the principle of the component is explained, the location of the component in the air handling unit is pointed out, and the way the component is controlled is illustrated.

Table 11. Heating/Cooling capacities, mass flows and water temperature of the heating/cooling circuits of AHU 5.

<table>
<thead>
<tr>
<th>Component</th>
<th>Capacity [kW]</th>
<th>Mass flow [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating coil</td>
<td>100,2</td>
<td>2,38</td>
</tr>
<tr>
<td>Cooling coil (cooling)</td>
<td>42,5</td>
<td>1,68</td>
</tr>
<tr>
<td>Cooling coil (dehumidification 45%)</td>
<td>84,5</td>
<td>3,39</td>
</tr>
<tr>
<td>Cooling coil (dehumidification 55%)</td>
<td>84,5</td>
<td>3,39</td>
</tr>
</tbody>
</table>

- Heating coil
  The heating coil in the air handling unit is used to heat air under forced convection. The surface of the heating coil consists of a primary and secondary heat transfer surface. The primary surface is the external surface of the tubes, the secondary surface are the fins mounted on the primary surface to extend the heat transfer surface. The heating coil, used in the air handling unit in this research, is a hot-water heating coil. The heating medium in the tubes is in counter flow (flow of supply and return fluid in the opposite direction) arrangement. This is done to obtain the highest possible mean temperature difference, to achieve the highest heat transfer capacity of the coil.

In Figure 29 can be seen that the heating coil is located at the end of the air handling unit, after the cooling coil, humidifier and dehumidifier (cooling coil).
The power output of the heating coil depends on the mass flow of the hot supply water to the coil. The mass flow of the water is variable and the incoming water temperature depends on the outdoor temperature (heating curve). The mass flow is varied by an equiprocentual two-way valve. The hydraulic circuit is shown below. On the left, the situation in this research and the theoretical ‘module 6’ of the “ISSO 44: Ontwerp van hydraulische schakelingen voor verwarmen” [25] is displayed. On the right the hydraulic and thermal behavior of the circuit can be seen.

![Hydraulic circuit diagram](image)

**Figure 30.** Left: passive user ‘module 6’, Right: graph hydraulic and thermal behavior of circuit (without readjustment of air temperature).

Some properties of this circuit are: the volume flow ($q_{v1}$) is variable, the return temperature of the water decreases when the system operates at part load, the volume flow at full load is set by control valve ‘IRA1’, readjustment with valves is not possible and fixed premixing not (temperature transformation $\theta_{2,100} < \theta_{1,100}$) possible at full load.

The water temperatures during full load and the reference temperature (= constant) are determined by:

$$\varepsilon = \frac{\theta_{2,100} - \theta_{1,100}}{\theta_{2,100} - \theta_{ref}}$$

(3.1)

This temperature efficiency $\varepsilon$ determines the thermal and hydraulic behavior; see the graph on the right in Figure 30. Where the black lines indicate the ratio of the reference air temperature which enters the heating coil ($\theta_{ref}$=constant), the blue lines indicate the ratio of the volume flow of the water entering the heating coil ($q_{v1}$) and the red lines indicate the ratio of the water temperature before ($\theta_1, \theta_2$=constant) and after ($\theta_3$) the heating coil.
• Cooling coil (High temperature = HT)
  The cooling coil in the air handling unit is used to cool air under forced convection. Just like the
  heating coil, the surface of a cooling coil consists of a primary and secondary heat transfer surface. The primary surface is the external surface of the tubes, the secondary surface are the fins mounted on the primary surface to extend the heat transfer surface. The cooling coil, used in the air handling unit in this research, is a cold-water cooling coil. The cooling medium in the tubes is again in counter flow (flow of supply and return fluid in the opposite direction) arrangement. Because condensation can occur a drain pan with a tube to the sewer is applied.

In Figure 31 can be seen that the cooling coil is located at the beginning of the air handling unit, before the humidifier, dehumidifier (cooling coil) and heating coil.

![Diagram of air handling unit 5 showing location of cooling coil.](image)

**Figure 31. Air handling unit 5: location cooling coil.**

The power output of the cooling coil depends on the mass flow of the cold supply water to the coil. The mass flow of the water is variable and the incoming water temperature is assumed constant. The mass flow is varied by an equiprocentual two-way valve. The hydraulic circuit is shown below in Figure 32. On the left the situation in this research and the theoretical ‘module 6’ of the “ISSO 47: Ontwerp van hydraulische schakelingen voor koelen” [26] is displayed. On the right the hydraulic and thermal behavior of the circuit is pictured.

Some properties of this circuit are: the volume flow \((q_v)_1\) is variable, the return temperature \((\theta_4)\) of the water increases when the system operates at part load, the temperature of the supply water \((\theta_2)\) is constant, the temperature of the supply water \((\theta_2)\) remains the same when there is a possibility of dehumidification, the volume flow at full load is set by control valve ‘IRA1’, readjustment with valves is not possible and fixed premixing possible not possible (temperature transformation \(\theta_{2,100} > \theta_{1,100}\))

The graph (Figure 32) of hydraulic and thermal behavior of the circuit is based on the following parameters:

- \(\theta_{2,100} = 28 \ [^\circ C]\)
- \(x_{2,100} = 0.013 \ [kg/kg]\)
- \(\gamma_{2,100} = 54.7 \ [%]\)
- \(\theta_{3,100} = \theta_{\text{ref}} = 14 \ [^\circ C] \ \text{constant}\)
- \(x_{3,100} = 0.01 \ [kg/kg] \ \text{constant}\)
- \(\gamma_{3,100} = 100 \ [%]\)
\[ \theta_{1,100} = \theta_{2,100} = 6 \, ^{\circ}C \]
\[ \theta_{3,100} = 12 \, ^{\circ}C \]

No readjustment applied

Here, the black lines indicate the ratio of the power of the heating coil, the blue lines indicate the ratio of the water temperatures of the heating coil, the purple lines indicate the ratio of the absolute humidity of the air and the red lines indicate the temperature of the ingoing/outgoing air of the heating coil.

Figure 32. Left: passive user ‘module 6’, Right: graph hydraulic and thermal behavior of circuit (based on design requirements, \( \theta_{13} \) and \( x_3 \) constant).

- **Humidifier**
  The humidification is used to increase the (relative) humidity of the air. Steam is used to humidify the air. The steam is treated in such a way that the humidification process does not affect the collection of the museum. The humidifier, used in the air handling unit in this research, is an electrically heated direct steam injection humidifier. This is a humidification system where steam, produced locally with electricity, is supplied to a perforated tube after passing through a condensate separator. The perforations face into the airstream. The temperature of the air remains almost constant, so this is an isothermal process. Attention must be paid to the absorption distance and the possible influence of additional chemicals at the indoor environment.

Figure 33. Electrical heated direct steam injection humidifier.
An advantage of this system is that softened or demineralized water can be used, which extends the maintenance period. In Figure 34 can be seen that the electrical steam humidifier is located between the cooling coil and the dehumidifier (cooling coil).

![Figure 34. Air handling unit 5: location electrical steam humidifier.](image)

The steam controller receives its set point from a master-slave control (see section 2.5.2). In Figure 35 is the control of the humidifier shown according [27]. Here, A1 are humidity sensors, B1 is a lock of ventilator, B2 is a flow control, B3 is a maximum hygrostat, B4 are hygrostats, Pl is an internal PI-controller of the humidifier, Pl is an external PI-controller of the humidifier and Y is the input signal from A1. The difference between the (relative) humidity in the room and the return air (relative) humidity is an input signal for the external PI-control and for the internal PI-control. The hygrostats in the room and the return air duct monitor the (relative) humidity and this signal is sent to the controller of the humidifier. So based on the difference between the (relative) humidity in the room and the return air (relative) humidity and a monitoring signal of the hygrostats the humidifier is controlled. The production of steam is controlled in the range of 0 to 100 %.

![Figure 35. Control of humidifier (extracted from [27]).](image)
- **Dehumidifier (cooling coil, low temperature = LT)**
  The dehumidifier is used to decrease the (relative) humidity of the air. Dehumidification, in the air handling unit in this research, is done by a cooling coil as described above but with a lower cold water circuit temperature (4°/6°C - 10°/12°C). Dehumidification by cooling is a form of reducing the capacity of the air to hold moisture and attracting the water of the air. This is done by passing air over the surface of the cooling coil that has been cooled below the air’s dew point. The outdoor air entering an air handling unit is precooled before passing over the cooling coil to remove excessive moisture content. Disadvantages of this approach is the limitation based on the dew point at the cooling coil and reheating is necessary after dehumidification. Also the cooling system must be able to supply water that is cold enough.

In Figure 36 can be seen that the dehumidification cooling coil is located between the electrical steam humidifier and the heating coil. The cooling coil for dehumidification is provided with a bypass as can be seen below. The cooling coil has a relatively high air-side resistance. So when no dehumidification is needed most of the air is let through the bypass to avoid a high pressure loss. Also, when not the full capacity of dehumidification is needed, a part of the air is let through the bypass to provide less dehumidification.

![Figure 36. Air handling unit 5: location dehumidification cooling coil.](image)

For the properties of the control of the dehumidification cooling coil (low temperature), see the description of the high temperature cooling coil. The differences between the two cooling coils are the water temperatures and the set points (based on temperature, based on absolute humidity) that are supplied to the control valves for controlling the position of the two-way valves.
3.4.2 Modeling of the heating coil

In this paragraph the mathematical model which is used for modeling the heating coil is described. The theory of the model of the heating coil that is described below is all based on [28].

The amount of energy that is transported from the heating coil to the air ($Q_{hc}$) must be the same as the amount of energy that is transported from the water to the heating coil. In Figure 37 the principle of a heating coil and its variables can be seen. The model will need to calculate time-dependent values, therefore the thermal capacity must be used. This will deal with energy storage. The basic equation is:

$$\rho \cdot c \cdot \dot{V} \cdot \frac{dT}{dt} = \left( m_w \cdot c_w \cdot (T_{w,in} - T_{w,out}) \right) - \left( m_a \cdot c_a \cdot (T_{a,in} - T_{a,out}) \right)$$

(3.2)

Where

- $\rho$ = the density [kg/m$^3$]
- $c$ = the specific heat capacity [J/kg.K]
- $\dot{V}$ = volume flow [m$^3$/s]
- $T$ = temperature [K]
- $t$ = time [s]
- $m_w$ = mass flow of water [kg/s]
- $c_w$ = specific heat capacity of water [J/kg.K]
- $T_{w,in}$ = temperature of incoming water [K]
- $T_{w,out}$ = temperature of outgoing water [K]
- $m_a$ = mass flow of air [kg/s]
- $c_a$ = specific heat capacity of air [J/kg.K]
- $T_{a,in}$ = temperature of incoming air [K]
- $T_{a,out}$ = temperature of outgoing air [K]

Because the temperature of the outgoing air and water temperature is not known, the $\varepsilon$-NTU method is used [29]. This method calculates the power supplied to the air, as a function of the temperature of the incoming water and air. For this calculation, the effectiveness and fluid capacity rates are needed. If the supplied power (to the air) of the heating coil is known, the outgoing water and air temperature can be determined.

The actual heat transfer rate is:

$$q = \varepsilon \cdot q_{max}$$

(3.3)
Where $q$ = the actual heat transfer rate [W]
$\varepsilon$ = the heat exchanger effectiveness [-]
$q_{\text{max}}$ = the maximum possible heat transfer rate [W]

Where $q_{\text{max}}$ is determined by:

$$q_{\text{max}} = C_{\text{min}} \cdot (T_{hi} - T_{ci})$$

Where $q_{\text{max}}$ = the maximum possible heat transfer rate [W]
$C_{\text{min}}$ = the smallest of the hot and cold fluid capacity rates [W/K]
$T_{hi}$ = the hot fluid entering [K]
$T_{ci}$ = the cold fluid entering [K]

The hot fluid capacity rate is defined as:

$$C_h = (\dot{m} \cdot c_p)_h$$

Where $C_h$ = the (hot) fluid capacity rate [W/K]
$\dot{m}$ = the mass flow of the hot fluid [kg/s]
$c_p$ = the specific heat capacity of the hot fluid [J/kg.K]

The cold fluid capacity rate is defined as:

$$C_c = (\dot{m} \cdot c_p)_c$$

Where $C_c$ = the (cold) fluid capacity rate [W/K]
$\dot{m}$ = the mass flow of the cold fluid [kg/s]
$c_p$ = the specific heat capacity of the cold fluid [J/kg.K]

In case of a heating coil, the water has the highest temperature and the air has the lowest temperature.

The heat transfer effectiveness can be expressed as a function of the number of transfer units (NTU) and the capacity ratio $c_r$. For a counter flow heat exchanger, the heat transfer effectiveness can be calculated by:

$$\varepsilon = 1 - e^{[-NTU(1-c_r)]}$$

when $c_r \neq 1$ (3.7)

$$\varepsilon = \frac{1 - c_r \cdot e^{[-NTU(1-c_r)]}}{1 + NTU}$$

when $c_r = 1$ (3.8)

The capacity rate ratio can be calculated by:

$$c_r = \frac{C_{\text{min}}}{C_{\text{max}}}$$

Where $c_r$ = the capacity rate ratio [-]
$C_{\text{min}}$ = the smallest of the hot and cold fluid capacity rates [W/K]
$C_{\text{max}}$ = the largest of the hot and cold fluid capacity rates [W/K]

The number of transfer units (NTU) can be calculated by:

$$NTU = \frac{UA}{C_{\text{min}}}$$

(3.10)
Where

\( NTU \) = the number of transfer units  
\( UA \) = a characteristic value for the surface area and heat transfer of a heat exchanger  
\( C_{\text{min}} \) = the smallest of the hot and cold fluid capacity rates

In this research, the \( UA \)-value is considered as a constant. The \( UA \)-value of the heat exchanger is determined by the revised design conditions. When these conditions of the heating coil are known, the \( UA \)-value can be calculated with logarithmic temperature difference and the power of the heating coil.

\[
UA = \frac{q}{\Delta T_{\text{lm}}}
\]  (3.11)

Where

\( UA \) = a characteristic value for the surface area and heat transfer of a heat exchanger  
\( q \) = the heat transfer rate  
\( \Delta T_{\text{lm}} \) = the logarithmic temperature difference

The logarithmic temperature difference can be calculated by:

\[
\Delta T_{\text{lm}} = \frac{(T_{\text{whci}} - T_{\text{hco}}) - (T_{\text{whco}} - T_{\text{hci}})}{\ln \left( \frac{T_{\text{whci}} - T_{\text{hco}}}{T_{\text{whco}} - T_{\text{hci}}} \right)}
\]  (3.12)

Where \( \Delta T_{\text{lm}} \) = the logarithmic temperature difference  
\( T_{\text{whci}} \) = the temperature of incoming water heating coil  
\( T_{\text{hci}} \) = the temperature of incoming air heating coil  
\( T_{\text{whco}} \) = the temperature of outgoing water heating coil  
\( T_{\text{hco}} \) = the temperature of outgoing air heating coil

To determine the outgoing air and water temperature two ordinary differential equations (ODE), based on equation 1.1, are defined.

Air:

\[
c_{\text{hca}} \cdot \frac{dT_{\text{hco}}}{dt} = m_{a, hc} \cdot c_a \cdot (T_{\text{hci}} - T_{\text{hco}}) + \varepsilon \cdot C_{\text{min}} \cdot (T_{\text{whci}} - T_{\text{hco}})
\]  (3.13)

Where \( C_{\text{hca}} \) = the heat capacity of the air  
\( T_{\text{hco}} \) = the temperature of the outgoing air of the heating coil  
\( t \) = the time  
\( m_{a, hc} \) = the mass flow of the air through the heating coil  
\( c_a \) = the specific heat capacity of the air  
\( T_{\text{hci}} \) = the temperature of the incoming air of the heating coil  
\( T_{\text{hco}} \) = the temperature of the outgoing air of the heating coil  
\( \varepsilon \) = the heat exchanger effectiveness  
\( C_{\text{min}} \) = the smallest of the hot and cold fluid capacity rates  
\( T_{\text{whci}} \) = the temperature of the incoming water of the heating coil  
\( T_{\text{hci}} \) = the temperature of the incoming air of the heating coil
Water:
\[
C_{hcw} \cdot \frac{dT_{whco}}{dt} = \dot{m}_{w, hc} \cdot c_w \cdot (T_{whci} - T_{whco}) - \varepsilon \cdot \dot{C}_{\min} \cdot (T_{whci} - T_{hcl})
\] (3.14)

Where
- \(C_{hcw}\) = the heat capacity of the water [J/kg.K]
- \(T_{whco}\) = the temperature of the outgoing water of the heating coil [K]
- \(t\) = the time [s]
- \(\dot{m}_{w, hc}\) = the mass flow of the water through the heating coil [kg/s]
- \(c_w\) = the specific heat capacity of the water [J/kg.K]
- \(T_{whci}\) = the temperature of the incoming water of the heating coil [K]
- \(T_{whco}\) = the temperature of the outgoing water of the heating coil [K]
- \(\varepsilon\) = the heat exchanger effectiveness [-]
- \(\dot{C}_{\min}\) = the smallest of the hot and cold fluid capacity rates [W/K]
- \(T_{whci}\) = the temperature of the incoming water of the heating coil [K]
- \(T_{hci}\) = the temperature of the incoming air of the heating coil [K]

During the heating process no moist is added to or extracted from the air. Therefore, the incoming and outgoing moisture content of the air will remain the same.

\[
x_{hcl} = x_{hco}
\] (3.15)

Where
- \(x_{hci}\) = the absolute humidity of the incoming air of the heating coil [kg/kg]
- \(x_{hco}\) = the absolute humidity of the outgoing air of the heating coil [kg/kg]

The formulas in this paragraph shall be executed with an S-function in MATLAB. The Simulink model of the heating coil will call this S-function and calculate the outgoing air temperature and the outgoing water temperature of the heating coil. The S-function of the model is shown in Appendix 5.1. Below, the basic Simulink model of the heating coil can be seen. The S-function is behind the ‘mask’ of the heating coil block. Later on in this research also the control of the heating coil will be modeled.

![Simulink model of heating coil](image)

**Figure 38. Simulink model of heating coil.**
On the left, the input parameters of the model and their dimension can be seen. On the right the calculated output parameters and their dimensions are shown. Additional to the five input parameters, three other parameters must be defined by double clicking on the model of the heating coil. Here, $C_{\text{primary circuit}}$ is the heat capacity of the water circuit and $C_{\text{secondary circuit}}$ is the heat capacity of the air circuit.

![Image of model parameters](image)

**Figure 39. Additional parameters of the heating coil.**

### 3.4.3 Validation of Simulink model

In this chapter the Simulink model of the heating coil, based on the mathematical model described above, will be validated according data from the building specifications (static validation) and revised technical drawings of the climate control system. At this moment, the model will not be validated with data from the building management system because the control of the heating coil is not included. Therefore, it is not possible to make a useful validation. Later on in this research, data from the building management system is used to optimize the control of the heating coil and to determine a possible correlation between the different parameters of the heating coil.

The UA-value is determined with the formulas mentioned in paragraph 3.4.2. The data from Table 11 and Table 12 are used to calculate this value. Below, the calculation of the UA-value is shown.

\[
\Delta T_{\text{lm}} = \frac{(45-26) - (35-9)}{100240/22.3} = 22.3 \, [K]
\]

\[
UA = \frac{q}{\Delta T_{\text{lm}}} = \frac{100240}{22.3} = 4492 \, \left[\frac{W}{K}\right]
\]

In the table below the results of the static validation can be seen. From the results, one can conclude that the model of the heating coil correctly calculates the static values. The slight deviations can be caused by differences in volume flow of the air, mass flow of the water and total power of the heating coil.

<table>
<thead>
<tr>
<th>Air Handling Unit 5 – “Grote Zaal (Wisselexpositie) + Souterrain”: Heating coil</th>
<th>Building specification</th>
<th>Validation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature air in [°C]</td>
<td>9.00</td>
<td>9.00</td>
</tr>
<tr>
<td>Temperature water in [°C]</td>
<td>45.00</td>
<td>45.00</td>
</tr>
<tr>
<td>Absolute humidity air in [kg/kg]</td>
<td>0.0028</td>
<td>0.0028</td>
</tr>
<tr>
<td>Mass flow water in [kg/s]</td>
<td>2.34</td>
<td>2.38</td>
</tr>
<tr>
<td>Volume flow air in [m³/h]</td>
<td>16992</td>
<td>16416</td>
</tr>
<tr>
<td>UA [W/K]</td>
<td></td>
<td>4491.58</td>
</tr>
<tr>
<td>Temperature air out [°C]</td>
<td>26.00</td>
<td>26.94</td>
</tr>
<tr>
<td>Temperature water out [°C]</td>
<td>35.00</td>
<td>35.18</td>
</tr>
<tr>
<td>Absolute humidity air out [kg/kg]</td>
<td>0.0028</td>
<td>0.0028</td>
</tr>
</tbody>
</table>

**Table 12. Parameters and results of the validation of the heating coil.**
4 Results

This chapter describes the results obtained in this research. The results are divided into three parts. Firstly, the results according energy use and possible savings are shown. Finally, the results that deal with the control of the mass flow, of the heating coil model that is described in paragraph 3.4.2, are pointed out.

4.1 Energy: HAMBASE Model

In this section, the results according energy use and possible savings for the year 2011 are shown. To define the energy use of the exhibition area, the outputs $Q_{\text{plant}}$ and $Q_{\text{hum}}$ of the HAMBASE simulation software are used. $Q_{\text{plant}}$ is hourly energy use in Wh (positive for ‘heating’, negative for ‘cooling’). Actually it is the energy which is going to the heating/cooling device. $Q_{\text{hum}}$ is hourly energy use for latent cooling Wh (positive for ‘humidification’, negative for ‘dehumidification’). Energy consumption is determined by heating, cooling, humidification and dehumidification, which are combined using a system efficiency (useful energy delivered divided by the total energy consumed) of 100%. In practice, the energy use will also depend on the type of systems used and their efficiency [5].

The results in this chapter are represented in a certain way to be able to get a good insight in the possible energy savings, the preservation conditions and the comfort for staff and visitors. All these evaluation points are incorporated in a table. For the energy part of the representation, the results described above are used. For the representation of the conservation conditions the specific climate risk assessment method of Martens [5] is used, a short description of this method can be found in Appendix 8.2. The comfort part of the representation is evaluated with the thermal comfort function integrated in the HAMBASE simulation software. This function estimates the PPD (Predicted Percentage of Dissatisfied) and PMV (Predicted Mean Voted) according to NEN-EN-ISO 7730 [30] for each zone if the relative velocity is given together with some data about the clothing and activity. To assess the thermal comfort in the exhibition area, class B of the NEN-EN-ISO 7730 is used. Specifications for the indoor thermal comfort can be found in Appendix 8.1. In the sections below, only colors are used in the table to assess the conservation conditions and comfort for staff and visitors. Specifications for this values, like separated energy consumptions for heating, cooling, humidification and dehumidification, can be found in Appendix 8.4.

4.1.1 Basic model

The basic model is assessed in this paragraph. The set points for temperature and relative humidity are derived from Figure 21 of section 3.2.2, the average temperature and relative humidity are used as set points for the simulation of the basic model in HAMBASE. In the following sections, the energy consumption of the particular situation is compared with the energy consumption of the basic model. The simulation results are shown in Table 13.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
<td>Vs. Base case [%]</td>
</tr>
<tr>
<td>Base case</td>
<td>19.7</td>
<td>45.2</td>
<td>118</td>
</tr>
</tbody>
</table>

Table 13. Summarized results of the simulation of the basic model.

The results show that there might be no risk for deterioration of the collection and that the comfort of the visitors and staff stays within the prescribed limits. Most of the energy is consumed by the heating process, followed by the cooling and dehumidification process and finally the humidification process.
4.1.2 Fixed temperature set point

In the basic case, a temperature set point of 19.7°C is used for heating and cooling. So a fixed set point, without a bandwidth, is used during the whole year. The set point for the relative humidity is kept the same (45.2 %). Now, simulations are performed where these set points are decreased by 1°C and 2°C (18.7 and 17.7°C) and increased by 1°C and 2°C (20.7 and 21.7°C). The MatLab-code which is used to overrule the values for $T_{setu}$ and $T_{setumax}$ in this section can be found in Appendix 5.3 and a graphical representation of the set points can be found in Appendix 8.3. The results of these simulations are shown in Table 14.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>1</td>
<td>+1°C</td>
<td>20.7</td>
<td>45.2</td>
</tr>
<tr>
<td>2</td>
<td>-1°C</td>
<td>18.7</td>
<td>45.2</td>
</tr>
<tr>
<td>3</td>
<td>+2°C</td>
<td>21.7</td>
<td>45.2</td>
</tr>
<tr>
<td>4</td>
<td>-2°C</td>
<td>17.7</td>
<td>45.2</td>
</tr>
</tbody>
</table>

Table 14. Summarized results of the simulations with a fixed temperature set point.

Increasing the temperature set point has a negative effect on the energy consumption, more energy is probably consumed. This is caused by heating and humidification. The lifetime multiplier even drops below 1.0 when the temperature is increased by 2°C. In the other situations, there might be no risk for deterioration of the collection and the comfort of the visitors and staff stays within the prescribed limits. Lowering the temperature set point causes energy savings. Lowering the temperature set point might lead to less energy consumption. This is caused by less heating and humidification. By lowering the temperature set point, the estimated percentage of people that are dissatisfied increases.

In the simulations above, no bandwidth for the temperature set point is used. In the next simulations, the bandwidth of the temperature set point is adjusted. Simulations are performed with a bandwidth of respectively ± 1°C and ± 2°C (19.7°C ± 1°C and 19.7°C ± 2°C). The results of these simulations are shown in Table 15.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>5</td>
<td>+/-1°C</td>
<td>18.7 – 20.7</td>
<td>45.2</td>
</tr>
<tr>
<td>6</td>
<td>+/-2°C</td>
<td>17.7 – 21.7</td>
<td>45.2</td>
</tr>
</tbody>
</table>

Table 15. Summarized results of the simulations with an adjusted bandwidth of the temperature set point.

In Table 15 can be seen that, using a bandwidth in combination with a temperature set point, results in a large effect on the energy consumption compared to the basic model. Possible energy savings up to 40% can be seen. When the bandwidth is doubled (from ±1°C to ±2°C), also the energy savings are almost doubled. Only the energy consumption for dehumidification does not decrease. The reason for this is probably, that the heating period is much longer than the cooling period (temperature is low for a longer period). So this means that less dehumidification is needed because cold air can contain less moist. Risk for deterioration of the collection and a too low comfort for the visitors and staff might not be expected. By increasing the bandwidth, the estimated percentage of people that are dissatisfied increases.
4.1.3 Fixed relative humidity set point

In the basic case, a relative humidity set point of 45.2% is used. A fixed set point, without a bandwidth, is used during the whole year. The set point for the temperature is kept constant (19.7°C). Now, simulations are performed where these set points are decreased by 5% and 10% (40.2 and 35.2%) and increased by 5% and 10% (50.2 and 55.2%). The MatLab-code which is used to overrule the values for \(rv_{\text{minu}}\) and \(rv_{\text{maxu}}\) in this section can be found in Appendix 5.3 and a graphical representation of the set points can be found in Appendix 8.3. The results of these simulations are shown in Table 16.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>7</td>
<td>19.7</td>
<td>50.2</td>
<td>119</td>
</tr>
<tr>
<td>8</td>
<td>19.7</td>
<td>40.2</td>
<td>119</td>
</tr>
<tr>
<td>9</td>
<td>19.7</td>
<td>55.2</td>
<td>123</td>
</tr>
<tr>
<td>10</td>
<td>19.7</td>
<td>35.2</td>
<td>121</td>
</tr>
</tbody>
</table>

Table 16. Summarized results of the simulations with a fixed relative humidity set point.

In the table above is shown that the energy consumption is probably not lowered by increasing or decreasing the fixed set point for the relative humidity. In contrast, the energy consumption is increasing. The results of the simulations indicate that the energy needed for heating and cooling is equal to the basic model. The savings of energy needed for humidification and dehumidification cancel each other. For instance, when the set point is lowered, the energy for humidification decreases but the energy needed for dehumidification increases. The lifetime multiplier drops below 1.0 when the relative humidity set point is increased by 10%. No risks for the deterioration of the collection occurs. The comfort of the visitors and staff stays within the prescribed limits.

In the simulations above, no bandwidth for the relative humidity set point is used. In the next simulations, the bandwidth of the relative humidity set point is adjusted. Simulations are performed with bandwidths of ±2%, ±5%, and ±10%. The results of these simulations are shown in Table 17.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>11</td>
<td>19.7</td>
<td>43.2 – 47.2</td>
<td>113</td>
</tr>
<tr>
<td>12</td>
<td>19.7</td>
<td>40.2 – 50.2</td>
<td>108</td>
</tr>
<tr>
<td>13</td>
<td>19.7</td>
<td>35.2 – 55.2</td>
<td>101</td>
</tr>
</tbody>
</table>

Table 17. Summarized results of the simulations with an adjusted bandwidth of the relative humidity set point.

By implementing a bandwidth for the relative humidity set point, energy savings might occur. The energy savings are smaller than by the implementation of a bandwidth for the temperature set point. Possible energy savings up to 15% can be seen. Also here, the energy needed for heating and cooling is equal to the basic model. Most of the energy is saved by the humidification process. Risk for deterioration of the collection and a too low comfort for the visitors and staff might not be expected.
4.1.4 Seasonal change temperature set point

Instead of a fixed temperature set point like in section 4.1.2, a temperature set point that fluctuates with the season is investigated in this paragraph. Firstly, a fixed set point for the seasons is chosen. Secondly, a temperature set point that is based on a sine curve is researched. In Table 18, the results of the fixed temperature set points for the seasons can be seen. Here, a set point of 17 or 18°C is selected for the winter, a set point of 21°C is selected for the autumn/spring and a set point of 24 or 25°C is chosen for the summer. These two options are also simulated with a bandwidth of ± 1°C. The Matlab-code which is used to overrule the values for \( T_{setu} \) and \( T_{setumax} \) in this paragraph can be found in Appendix 5.3 and a graphical representation of the set points can be found in Appendix 8.3.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>14</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation A (step)</td>
<td>18-21-24</td>
<td>45.2</td>
<td>115</td>
</tr>
<tr>
<td>15</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation B (step)</td>
<td>17-21-25</td>
<td>45.2</td>
<td>113</td>
</tr>
<tr>
<td>16</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation C (step)</td>
<td>A ± 1</td>
<td>45.2</td>
<td>88</td>
</tr>
<tr>
<td>17</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation D (step)</td>
<td>B ± 1</td>
<td>45.2</td>
<td>92</td>
</tr>
</tbody>
</table>

Table 18. Summarized results of the simulations with a (seasonal) step change in the temperature set point, with and without a bandwidth.

In Table 18, can be seen that, by letting the indoor temperature set point depend on the season (outdoor conditions), possible energy savings can be obtained. In situation A the possible energy saving is small, because the amount of energy needed for heating is increased. Almost all situations lead to a decrease of the lifetime multiplier (below 1.0). When a low temperature set point (17°C) is used in the winter and a high temperature set point (25°C) in the summer, thermal comfort problems occur. When a bandwidth is added to the step change in temperature set point, more energy can possibly be saved. Only one situation meets the criteria for risk assessment and thermal comfort, also the highest energy savings are obtained with these set points (cause: less energy for heating and humidification).

The results below deal with a temperature set point that is based on a sine curve. The sine curve is based on outdoor conditions, so the indoor temperature set point fluctuates with the season. When the outdoor temperature is low, a lower indoor temperature is allowed and when the outdoor temperature is high, a higher indoor temperature is allowed. Firstly, a fixed temperature is used as an equilibrium position (19.7°C), so the maximum \( T_{setumax} \) and minimum \( T_{setu} \) temperature is not changed. The amplitude is changed with 1, 2 or 3°C. Hereafter, the equilibrium position is adjusted and the maximum \( T_{setumax} \) and minimum \( T_{setu} \) temperature are set to 18 and 23°C. Also here, the amplitude is changed with 1 or 2°C. The results are shown in Table 19.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>18</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation (sine)</td>
<td>19.7 ± 1</td>
<td>45.2</td>
<td>109</td>
</tr>
<tr>
<td>19</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation (sine)</td>
<td>19.7 ± 2</td>
<td>45.2</td>
<td>99</td>
</tr>
<tr>
<td>20</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation (sine)</td>
<td>19.7 ± 3</td>
<td>45.2</td>
<td>92</td>
</tr>
<tr>
<td>21</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation (sine)</td>
<td>18/23 ± 1</td>
<td>45.2</td>
<td>61</td>
</tr>
<tr>
<td>22</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal fluctuation (sine)</td>
<td>18/23 ± 2</td>
<td>45.2</td>
<td>55</td>
</tr>
</tbody>
</table>

Table 19. Summarized results of the simulations with a (seasonal) change, based on a sine curve, in the temperature set point.
The potential of energy saving, compared to the basic model, by letting the temperature set point fluctuate with the season (sine curve) is enormous. Also here, applying a bandwidth instead of a fixed temperature increases the possible energy savings. The energy consumption might be halved, compared to the basic model. The highest energy savings are achieved for heating and cooling. No risks for the collection or decrease of the thermal comfort occurs. By increasing the bandwidth, and thereby decreasing/increasing the temperature, the estimated percentage of people that are dissatisfied will increase.

4.1.5 Seasonal change relative humidity set point

Here, the relative humidity set point is varied over the seasons. Firstly, a fixed set point for the seasons is chosen. Secondly, a relative humidity set point that is based on a sine curve is researched. Below, a set point of 30 or 35% is selected for the winter, a set point of 45% is selected for the autumn/spring and a set point of 55 or 60% is chosen for the summer. These two options are also simulated with a bandwidth of ± 5%. The Matlab-code which is used to overrule the values for $T_{setu}$ and $T_{setumax}$ in this paragraph can be found in Appendix 5.3 and a graphical representation of the set points can be found in Appendix 8.3. The results of the simulations can be seen in Table 20.

<table>
<thead>
<tr>
<th>Set points</th>
<th>Energy</th>
<th>Risk assessment</th>
<th>Thermal comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>T [°C]</td>
<td>RH [%]</td>
<td>Total [MWh]</td>
</tr>
<tr>
<td>23 Seasonal fluctuation E (step)</td>
<td>19.7</td>
<td>35-45-55</td>
<td>112</td>
</tr>
<tr>
<td>24 Seasonal fluctuation F (step)</td>
<td>19.7</td>
<td>30-45-60</td>
<td>112</td>
</tr>
<tr>
<td>25 Seasonal fluctuation G (step)</td>
<td>19.7</td>
<td>E ± 5</td>
<td>102</td>
</tr>
<tr>
<td>26 Seasonal fluctuation H (step)</td>
<td>19.7</td>
<td>F ± 5</td>
<td>101</td>
</tr>
</tbody>
</table>

Table 20. Summarized results of the simulations with a (seasonal) step change in the relative humidity set point, with and without a bandwidth.

The results in the table above indicate a possible energy saving when the set points for relative humidity depend on the outdoor conditions (seasons). In contrast to the temperature settings, lower set points for relative humidity make no difference for the possible energy savings. By applying a bandwidth for the relative humidity set points, a lower energy consumption can be achieved. An equal amount of energy for heating and cooling is obtained by the simulations. Too high/low relative humidities could possibly lead to damage of the pictorial layer of paintings.

The results below deal with a relative humidity set point that is based on a sine curve. The sine curve is based on outdoor conditions, so the indoor relative humidity set point fluctuates with the season. In summer, when the outdoor relative humidity is high, the indoor relative humidity can also be higher. The opposite happens in the winter. Just as in section 4.1.4, firstly a fixed equilibrium position is used (45.2 %), so the maximum ($rv_{maxu}$) and minimum ($rv_{minu}$) relative humidity is not changed. The amplitude is changed with 5, 10 or 15%. Hereafter, the equilibrium position is adjusted and the maximum ($rv_{maxu}$) and minimum ($rv_{minu}$) relative humidity are set to 40 and 50%. Also here, the amplitude is changed with 5 or 10%. The results are shown in Table 21.
Table 21. Summarized results of the simulations with a (seasonal) change, based on a sine curve, in the relative humidity set point.

The difference in possible energy savings between a set change in relative humidity set point and a set point that fluctuates with the season (sine curve) is not that large as it is with the temperature set point. Still, up to 15% of possible energy savings can be achieved. The energy needed for heating and cooling is equal to the basic model. Almost no deviations occur in the results of the risk assessment and thermal comfort. So, there might be no risk for deterioration of the collection and the comfort of the visitors and staff stays within the prescribed limits.

4.1.6 Temperature set point based on outdoor conditions

Instead of a sine curve, actual weather conditions can be used to determine set points. This is a technique that already is widely used in office buildings in the Netherlands. Here, the average outdoor temperature of the last three days is calculated. When this average value reaches a temperature of 5°C or lower, a temperature set point of 18, 17 or 16°C is applied indoors. In case the average value reaches a temperature of 20°C or higher, a temperature set point is set to 22, 23 or 24°C. The set point increase is proportional to the average outdoor temperature increase. The MatLab-code (by Martens [5]) to determine the set point based on the real weather conditions can be found in Appendix 5.3 and a graphical representation of the set points can be found in Appendix 8.3.

Table 22. Summarized results of the simulations with a temperature set point, based on a the outdoor conditions.

Table 22 shows that possible energy savings up to 25% can be achieved when applying a temperature set point that depends on the actual weather conditions. The largest effect is noticeable on the cooling and dehumidification. The reason for this could be that the outdoor temperature drops less below 5°C than it rises above 20°C, so the heating set point is less adapted. It might be wise to adjust the parameters based on the outdoor climate and the building needs. By decreasing and increasing the temperature set point when a certain outdoor temperature is reached, the estimated percentage of people that are dissatisfied will increase. The PPD reaches values near the boundary of the thermal comfort limits. There might be no risk for deterioration of the collection.
4.1.7 Relative humidity set point based on outdoor conditions

In this section, the relative humidity set point is determined based on the outdoor conditions. When the outdoor relative humidity is lower than 40%, the indoor relative humidity set point is set to 30 or 35%. When the outdoor relative humidity is higher than 80%, the indoor relative humidity set point is set to 55 or 60%. The results of the simulation are shown in Table 23.

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<th>Thermal comfort</th>
</tr>
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<td>Total [MWh]</td>
</tr>
<tr>
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<td>19.7</td>
<td>&lt;40%=35%</td>
<td>&gt;80%=55%</td>
</tr>
<tr>
<td>Outdoor conditions</td>
<td>19.7</td>
<td>&lt;40%=30%</td>
<td>&gt;80%=60%</td>
</tr>
</tbody>
</table>

Table 23. Summarized results of the simulations with a relative humidity set point, based on the outdoor conditions.

It seems that, letting the relative humidity set point depend on the actual weather conditions does not lead to energy savings. The energy consumptions even rises. The reason for this could be that the energy that is needed for humidification is increased, because the outdoor relative humidity very often exceeds the upper boundary (80%). The energy needed for heating and cooling is equal to the basic model. There might also be risks for deterioration of the collection. The lifetime multiplier drops below 1.0 and too high/low relative humidities could possibly lead to damage of the pictorial layer of paintings. The comfort of the visitors and staff stays within the prescribed limits.

4.1.8 Combination of set points

Based on the conclusions of the sections above, the most promising settings are combined to investigate whether these combinations could lead to even a lower energy consumption and better indoor environment. Option 37 uses a temperature set point that is 2°C lower and a relative humidity set point that is 5% higher than in the basic model. Option 38 simulates the exhibition area with a fixed temperature and relative humidity set point that have a bandwidth of ±2°C and ±10%. Set points that depend on the seasons are used in option 39. A set point of 18°C/35% is selected for the winter, a set point of 21°C/45% is selected for the autumn/spring and a set point of 24°C/55% is chosen for the summer. All set points use a bandwidth of ±1°C/±10%. Option 40 and 41 combine set points for temperature and relative humidity that fluctuate with the seasons (sine curve), one with the same minimum and maximum temperature and one with a different minimum and maximum temperature. Both options have a bandwidth of ±2°C and ±10%. Option 42 uses a temperature set point based on the actual weather conditions and a relative humidity set points that fluctuates with the seasons. The results of these simulations can be seen in Table 24.

From the results in this table one can conclude that all the combinations might cause less energy consumption, that there may be no risks for deterioration of the collection and that the thermal comfort of the visitors and staff probably does not exceed the prescribed limits. When the results of the set point combinations are compared with the results of the separate set points it can be seen that only option 37 (-2°C / +5%) does not lead to higher energy savings. The highest energy savings might be achieved by option 41 (seasonal fluctuation based on a sine curve with min./max.: 18/23 ± 2°C and 40/50 ± 10%). However, this option has the lowest thermal comfort (still within the limits).
The second best option according possible energy savings, option 38 (± 2°C and ± 10%), has a better thermal comfort and the lifetime multiplier is also higher for all typical art objects. The next best options (option 6 and 3) all have a lower thermal comfort parameter.

Table 24. Summarized results of the simulations with a combination of most promising set points.

4.1.9 Discussion

In this section, the results that can be found in the paragraph above are discussed. Conclusions, based on this discussion, are drawn in the last chapter (5) of this report.

An average value for the PMV (Predicted Mean Vote) and the PPD (Predicted Percentage of Dissatisfied) is determined to judge the thermal comfort of the exhibition area. At certain moments in time, there exists higher and lower values than this average value. So it must be kept in mind that at some moments there could be thermal discomfort. Also, the exhibition area is a very high room. The temperature distribution in such a room is different from a normal room, so other local conditions can occur which can influence the thermal comfort.

Because the capacities for heating, cooling, humidification and dehumidification are almost always able to achieve the desired set points in the simulation, constant temperatures and relative humidities are obtained. Almost all mechanical risk are therefore negligible. One may wonder whether this simulations are approaching the real situation, because it seems almost as keeping under strict storage conditions.

With the aid of the bar charts below, the possible energy savings of the different set point optimizations is discussed. The energy consumption is expressed as a percentage. The colors of the bars indicate the level of energy consumption as a percentage, red means a high energy consumption, blue means a low energy consumption. In the tables above, there is a column with a number for each set point. This number can also be found on the x axis in the bar charts. Also the energy savings as a percentage, compared to the base case is shown on the x axis. A horizontal dotted line is displayed in the bar charts which indicates the energy consumption (100 %) of the base case.

From Figure 40 one can conclude that, lowering a fixed set point for temperature (2/3) could possibly lead to energy savings up to 8%. Lowering a fixed set point for relative humidity (8/10) probably doesn’t result in a lower energy consumption, this might even lead to a higher energy consumption. By using bandwidths, when controlling the temperature (5/6) and relative humidity (11/12/13), a large effect on the energy consumption compared to the basic model can be seen. Energy savings up to 41% (temperature) and 15% (relative humidity) might be achieved. The possible energy savings, and the areas of less energy consumption depend on the conditions inside the zone. For instance, when the...
heating period is much larger than the cooling period (temperature is low for a longer period) less dehumidification is needed and therefore the energy savings for dehumidification are the smallest.

Step change of temperature set points based on the seasons (14/15) might lead to a slightly lower energy consumption (2 to 4%). However, the risks for the deterioration of the collection and the estimated percentage of people that are dissatisfied will increase. This is not the case when implementing a step change of relative humidity set points (23/24). The possible energy savings caused by a fixed step change of the relative humidity set points is larger than the energy savings caused by fixed temperature set points. Energy savings of 5% might be achieved. When a bandwidth is applied this effect is reversed. An even lower energy consumption can be reached when a bandwidth is used. Energy savings up to 25% for changing temperature set points (16/17) and 14% for changing relative humidity set points (25/26) might be achieved.

The highest potential of possible energy savings, compared to the basic model, is achieved by letting the temperature set point fluctuate with the season based on a sine curve. When using fixed set points (18/19/20) possible energy savings of 8, 16 and 22% can be reached. When a bandwidth is applied almost half of the energy might be saved. Letting the relative humidity set point fluctuate with the season based on a sine curve (27/28/29) has a lower potential. Only energy savings of respectively 4, 6 and 8% might be obtained. The energy savings are doubled (12%, 15%) when a bandwidth is implemented (30/31). The difference in possible energy savings between a step change in relative humidity set point and a set point that fluctuates with the season (sine curve) is not that large as it is with the temperature set point. The highest energy savings are achieved for heating and cooling. By increasing the bandwidth, and thereby decreasing/increasing the temperature, the estimated percentage of people that are dissatisfied will increase.
Possible energy savings can be achieved when applying a temperature set point that depends on the actual weather conditions (32/33/34). It might be wise to adjust the parameters based on the outdoor climate and the building needs, to optimize the possible energy savings. The PPD reaches values near the boundary of the thermal comfort limits. The energy consumption is 14, 21 and 26% lower than in the base case situation. Letting the relative humidity set point depend on the actual weather conditions (35/36) might not lead to energy savings. There might also be risks for deterioration of the collection. Therefore, it might be not a good solution to let the relative humidity set point depend on the actual weather conditions.

All the combinations of set points might cause less energy consumption, there may be no risks for deterioration of the collection and the thermal comfort of the visitors and staff probably does not exceed the prescribed limits. Option 37 (-2°C/+5%) does not lead to high energy savings, only 8% compared to the base case. The most promising option uses temperature and relative humidity set points that fluctuate with the seasons based on a sine curve (41). The equilibrium position is set to 18/23°C and 40/50% and the amplitude is set to ±2°C and ±10%. A possible energy saving of 64% might be achieved. This option has the lowest thermal comfort (still within the limits). Therefore, the best option according energy savings, risk assessment and thermal comfort is the one where a bandwidth is applied to the set points for temperature and relative humidity (38). This option might cause a possible energy saving of 55% compared to the base case situation.

In Figure 41 till Figure 44 the possible energy savings for the separate processes (heating, cooling etc.) are pointed out. In these graphs can be seen that the largest savings (53%) on energy needed for heating, compared to the basic model, can be achieved by applying a set point that fluctuates with the season based on a sine curve (set point 22). This also applies to cooling (93 %). The largest savings on energy needed for humidification, compared to the basic model, can be achieved by applying a set point with a bandwidth of ±10% (set point 13) or by applying a set point that fluctuates with the season based on a sine curve (set point 31). A possible energy saving of 67 to 69% can be achieved. This also applies to dehumidification, where 55 to 57% less energy might be used compared to the base case. Also set point 26, a step change of relative humidity based on the season including a bandwidth, results in an energy saving of 55%. However, risks for the deterioration of the collection occur. For the combined set point settings, the largest savings (53%) on energy needed for heating, compared to the basic model, can be achieved by applying a temperature and relative humidity set point that fluctuates with the season based on a sine curve (set point 41). This also applies to cooling (93%).

The largest savings on energy needed for humidification, compared to the basic model, can be achieved by applying a bandwidth to the temperature and relative humidity set point (set point 38). A possible energy saving of 81% can be achieved. The largest savings on energy needed for dehumidification, compared to the basic model, can be achieved by applying a temperature set point that is based on the actual weather conditions and a relative humidity set point that fluctuates with the season based on a sine curve (set point 42). 77% less energy might be used compared to the base case.

When the energy consumptions for heating, cooling, humidification and dehumidification are compared to the real situation (calculated energy consumption based on measurement data) almost the same results are obtained.
Figure 41. Simulated energy consumption for heating with different set points for T and RH.

Figure 42. Simulated energy consumption for cooling with different set points for T and RH.
Figure 43. Simulated energy consumption for humidification with different set points for T and RH.

Figure 44. Simulated energy consumption for dehumidification with different set points for T and RH.
In the introduction in chapter 1, a research of Mecklenburg is discussed. He investigated several buildings at the Smithsonian Museum and found that the energy costs decreased significantly as a wider range of relative humidity was allowed (see Figure 1). Mecklenburg claims that increasing the relative humidity tolerance from ±2% RH to ±7% RH will reduce energy costs by 55%. In this research, the influence of the relative humidity fluctuation on the energy consumption, instead of the energy costs, is investigated. In contrast to the research of Mecklenburg, here only one building is researched.

![Graph showing energy consumption correlated to relative humidity control](image)

**Figure 45. Hermitage Amsterdam: Energy consumption correlated to relative humidity control.**

The correlation between the energy consumption and the relative humidity fluctuation is shown in Figure 45. It can be seen that there is an exponential relationship between the energy consumption and the level of control; as the variance of the relative humidity conditions increases, the energy consumption decreases exponentially. Mecklenburg suggested that increasing the relative humidity tolerance from ±2% RH to ±7% RH will reduce energy costs by 55%. From the graph above, one can conclude that this is not the case in the Hermitage Amsterdam. Here, only an energy saving of approximately 10% might be achieved (12 MWh). The orange lines, in different line styles, indicate that there might be a risk for the deterioration of the collection when certain relative humidity fluctuation is used. For instance, when a relative humidity fluctuation is used of ±16% than there might be a risk for the deterioration of the base layer of a sculpture.

New climate specifications (21°C ±2°C and 45% RH ±8% RH) were adopted by Mecklenburg, which led to enormous energy savings. These climate specifications could be a good starting point for the Hermitage Amsterdam, when the results from this chapter and the figure above are taken into account.
As said before in chapter 2.4, the primary energy for heating and cooling is a long term energy storage system. This system makes use of aquifers to extract and store energy. Aquifers are soil layers which include groundwater. During summer cold is extracted to cool the building and superfluous heat is stored in the soil layers to use it during the heating season. Two heat pumps are used to upgrade the energy during the cooling and heating season. To ensure the good working of the system, it is important that the energy balance between the hot and cold well remains intact. This means that after a while the amount of extracted energy should be the same as the amount of energy that is added to ground. This is also required by legislation. An imbalance can cause problems according efficiency of the heat pumps, thermal short circuit, thermal contamination of the soil etc.

Figure 46 shows the monitoring report of the thermal energy storage system of the Hermitage Amsterdam for the year 2011. From this report, one can conclude that there is a heat surplus in the system of 664 MWh (red rectangle). This means that there is more cold than heat extracted from the aquifers. A simple and inexpensive solution for this problem could be to lower the cooling needs of the building. The cooling needs of the building, which must be fulfilled by the thermal energy storage system, includes cooling and dehumidification.

For dehumidification, the water from the aquifer must be cooled by the heat pumps to achieve a lower temperature. So the water from the aquifer is not directly used in the system. In this case, the load becomes higher since the supplied cooling energy is often completely extracted from the thermal energy storage system. In addition there is additional heat supplied to the thermal energy storage system, which is released by the compressors of the heat pumps. Depending on the situation, this residual heat could be partially used for reheating. For dehumidification by making use of a cooling coil (dew point cooling) is relatively a large amount of cooling energy needed. The air is cooled considerably, but a small amount of latent cooling occurs. When this residual cooling cannot be used, extra energy is needed for reheating of the air. This depends heavily on the type of system, the generating systems, cooling water temperature, cooling coil characteristics, building requirements etc. In general it can be said that in case of dew point cooling, a reduced need for dehumidification, yields a considerable saving on the energy consumption for cooling. In the example below, only the energy needed for cooling is pointed out.

The data in the table below makes not clear whether there is already a part of the energy regenerated. It is assumed that there is already an additional amount of cold energy stored in the aquifer in the winter by using the closed heat exchanger consisting of a boxcooler that is placed in the water of the Amstel. This could mean that the actual imbalance is even larger, and that all additional auxiliary energy is used for regeneration.

When the results of the cooling energy consumption is taken into account (Figure 42), it can be seen that, for example, set point 41 (and 22) yields a saving of 93% on cooling power. However, for the exhibition area cooling capacity is not that large (approximately 25 MWh) and the savings thus provides only a decrease of the cooling capacity of approximately 24 MWh. In this research only one exhibition area is investigated, so a larger effect on the balancing of the thermal energy storage system can be expected when the whole building is taken into account.
### Monitoringsrapport Provincie

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<th>Storen warmte levering</th>
<th>Storen koude levering</th>
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<td><strong>7,4</strong></td>
<td><strong>9,9</strong></td>
</tr>
</tbody>
</table>

**Figure 46. Monitoring report thermal energy storage system Hermitage Amsterdam (2011).**
4.2 Control: Simulink model(s)

In section 3.4.2 is the basic model of the heating coil described. This model is validated with constant parameters, so without implementing the control of the mass flow through the heating coil. In this chapter, an appropriate way to model this control is researched. Firstly, the mass flow is tried to be determined by using the control strategies and settings that are implemented in the actual situation and after that an alternative approach is adopted.

4.2.1 PI-control heating coil

The power output of the heating coil depends on the mass flow of the hot supply water to the coil. The mass flow of the water is variable and the incoming water temperature depends on the outdoor temperature (heating curve). Because no information is known after the temperature control action (mass flow of the supply water), the control must also be modeled. Firstly, the general principle of the control will be explained. Secondly, the operation of the heating curve of the hot supply water will be pointed out. At last, the complete control model of the mass flow of the heating coil will be described.

The temperature control of the heating coil consists of a master-slave control. A description can be found in paragraph 2.5. This means that there are two control loops, the master control loop which controls the slow process, in this case, the indoor air condition of the exhibition area and the slave control loop which controls the fast process, in this case, the supply air temperature.

Firstly, the set point value of the room air temperature is compared with the measured value of the room air temperature. By making use of a PI-controller (description, see paragraph 2.5) a set point for the supply air temperature is determined. This set point is compared with the measured value of the supply air temperature and the deviation between these values is used, by a second PI-controller, to determine the position of the two-way valve which controls the mass flow of hot water through the heating coil. The principle described above is shown in Figure 47.

The temperature of the supply water can be constant or can depend on the outdoor temperature. The control of the constant temperature is simple and yields under all circumstances an equal supply water temperature. When an outdoor temperature dependent control is used, a strategy must be determined which supply water temperature must be supplied at a certain outdoor temperature. Below, a strategy for an outdoor temperature dependent control is pointed out.

In order to determine the minimum and maximum water temperatures, the mechanical specifications are used. The desired central heating supply water temperature is determined by means of the following heating curve: -10°C/45°C, 20°C/35°C, 28°C/35°C. The heating curve is shown in Figure 48.
In the building management system a formula is found which determines the set point for the supply water temperature.

\[
Y = Y_1 + (X - X_1) \cdot \frac{(Y_2 - Y_1)}{(X_2 - X_1)}
\]  \hspace{1cm} (4.1)

Where

\begin{align*}
Y & = \text{supply water temperature} \\
Y_1 & = \text{maximum supply water temperature} \quad = 45°C \\
Y_2 & = \text{minimum supply water temperature} \quad = 35°C \\
X & = \text{outdoor temperature} \\
X_1 & = \text{minimum outdoor temperature} \quad = -10°C \\
X_2 & = \text{maximum outdoor temperature} \quad = 28°C
\end{align*}

A Simulink model (Figure 49) for the determination of the supply water temperature is made. The ability to choose a constant or outdoor compensated control is included in this model. If the option is selected for an outdoor temperature compensated control, the model determines the supply water temperature based on the outdoor temperature and the heating curve. The water temperature is limited by the saturation block (max. 45°C and min. 35°C).
The input, output and parameters of this model are:

- Input: outdoor temperature ($T_e$)
- Output: supply water temperature ($T_{w,supply}$)
- Parameters: maximum supply temperature ($T_{w,max}$), minimum supply temperature ($T_{w,min}$), outdoor temperature corresponding to the maximum supply temperature ($T_{e,max}$), outdoor temperature corresponding to the minimum supply temperature ($T_{e,min}$)

It is assumed that the hydraulic differential pressure is constant. The airflow over the heating coil is considered to be constant. If the amount of air is reduced by, for example a rotation frequency control of the fan, then the process is more difficult to control, and also the control range is increased. The determination of the control parameters should be done on the basis of the most unfavorable operating conditions.

The theory of the master-slave control has been incorporated into a Simulink model for controlling the mass flow of the heating coil. As said before, the master-control determines a set point for the supply air temperature and the slave-control determines the position of the two-way valve which controls the mass flow. The Simulink model can be seen in Figure 50. On the left, the determination of the desired supply air temperature can be seen with a block that applies a dead zone and the PI-controller. On the right, the determination of the position of the two-way valve is shown. Also here, a block that applies a dead zone and a PI-controller can be seen. The settings for the PI-controllers, that are shown in Table 25, are derived from the building management system and can be found in Appendix 7.3. By trial and error it was determined that, instead of a PI-controller to determine the supply air temperature, a P-controller was more appropriate to apply. When an integral action is used, the outgoing air temperature rises to and stays at a too high temperature. Also in practice, mostly a P-controller is applied because the process of the determination of the supply air temperature is a very slow process. The proportional parameter was set on the default value (0.3), which can be found in the table below.

![Figure 50. Simulink master-slave control mass flow heating coil.](image)

<table>
<thead>
<tr>
<th>Heating coil: control settings</th>
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</thead>
<tbody>
<tr>
<td></td>
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<tr>
<td><strong>Master (supply air temperature)</strong></td>
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<tr>
<td>----------------------------------</td>
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<td></td>
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<tr>
<td><strong>Slave (position valve)</strong></td>
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</table>

Table 25. Settings PI-controllers heating coil.

In the middle of Figure 50, a summation block can be found. There is a difference determined between a set point temperature of the exhibition area and the actual temperature in the area, the P-controller defines how to react on that deviation. This deviation is actually the correction of the incoming air temperature of the heating coil. Therefore, this temperature and the deviation are added and chosen as a ‘starting temperature’.
Below, an image is displayed of the Simulink model of the complete model of the heating coil with the control of the mass flow of the water. The heating coil itself is displayed in the red rectangle, the model to determine the supply water temperature can be seen in the yellow rectangle and the model of the mass flow control (which is described above) can be found in the blue rectangle.

![Simulink model](image)

Figure 51. Complete Simulink model of the heating coil (red) with the control of the supply temperature (yellow) and the mass flow of the water (blue).

To eliminate the uncertainties in the model, a mixed air temperature was calculated as the ingoing air temperature of the heating coil. This was done because the temperature transmitter TT02 was placed before the location where the dehumidified air and the air from the by-pass are combined. The S-function and the Simulink model that is used to calculated this mixed air temperature can be found in Appendix 5.8. Often, the fan in an air handling unit produces heat and therefore the air that is moved by the fan is warmed up. An additional 1.5°C is added to the mixed air temperature to take this heating into account.

The supply air temperature is simulated for the whole year of 2011, one week in January 2011, one day in July 2011 and one week in November 2011. The results of the simulated supply air temperature and the measured supply air temperature (TT04) are compared with each other. The graph of the whole year and of one week in November are shown in Figure 52 and Figure 53. The graphs of the other periods can be found in Appendix 9.1.

From the simulation of the whole year, one can conclude that the simulated temperature is almost the whole year a few degrees lower than the measured temperature. However, the graph of one week in November shows a good similarity between the simulated and calculated value. This might be the only period in the year (see graph of the whole year) which shows a good correspondence. From the results in the appendix one can conclude that, in general, the simulated temperature profile corresponds with the calculated temperature profile. Though, there exist some deviations. The largest difference are found in the summer period. At some moments, differences of around 3°C can be found.
Figure 52. Comparison simulated supply air temperature and the measured supply air temperature (TT04).

Figure 53. Comparison simulated supply air temperature and the measured supply air temperature (TT04).
4.2.2 Correlation heating coil

In the previous paragraph the mass flow was tried to be determined by using the control strategies and settings that are implemented in the actual situation. Because the desired result was not achieved another approach was adopted.

In the following section it is investigated if there is a correlation between the mass flow of the water and ingoing and/or outgoing temperature of the air of the heating coil. Firstly the correct mass flow is determined, after that the correlation is investigated and finally, when there is a correlation, the mathematical relation is derived.

To determine the mass flow, which reproduces the exact outgoing air temperature of the heating coil, a separate Simulink model is built in which a part of the controller from section 4.2.1 is integrated. In this model only the PI-controller which controls the mass flow (position of the valve) is included. As input parameter of the control, the measurement data of the temperature transmitter after the heating coil is inserted (TT04). In Figure 54 an image of the Simulink model is shown. In this image a constant ingoing water temperature is shown, but in order to determine the mass flow, there is a supply water temperature used which depends on the outdoor temperature (heating curve, see section 4.2.1).

![Simulink model to determine the mass flow for exactly reproducing Tair_out.](image)

The outgoing air temperature of the heating coil is reproduced for one week in January 2011 and one day in December 2011. As expected, almost no deviations occur, except for a couple of hours at the 17th of January. No reason is found for this deviation. The accompanying mass flow of a whole year is shown in Figure 55.
Four parameters were selected to search for a correlation between the mass flow of the water and ingoing and/or outgoing temperature of the air of the heating coil, namely the calculated ingoing air temperature ($T_{mix}$), the measured outgoing air temperature ($T_{T04}$), the temperature difference ($T_{T04} - T_{mix}$) and the simulated mass flow ($\dot{m}$) of the water through the heating coil. Only data of the month December was used to search for a correlation.

Correlation quantifies the strength of a linear relationship between two variables \[31\]. Two variables that are uncorrelated are not necessarily independent, they might have a nonlinear relationship. Before the relationship between pairs of quantities is modeled, a correlation analysis is performed to establish if a linear relationship exists between the quantities. Covariance quantifies the strength of a linear relationship between two variables in units relative to their variances. Correlations (coefficients) are standardized covariances, giving a dimensionless quantity that measures the degree of a linear relationship, separate from the scale of either variable.

The correlation coefficients range from -1 to 1, where
- Values close to 1 indicate that there is a positive linear relationship between the data columns.
- Values close to -1 indicate that one column of data has a negative linear relationship to another column of data (anti correlation).
- Values close to or equal to 0 suggest there is no linear relationship between the data columns.

A data model explicitly describes a relationship between predictor and response variables. Linear regression fits a data model that is linear in the model coefficients. The most common type of linear
regression is a least-squares fit. The method of least squares is used to find formulas to the so-called regression lines that are associated with the cloud of points of the relation.

By making use of MatLab, the correlation coefficients for the four parameters are determined. In the table below can be seen for which parameters the correlation coefficient is calculated and also the actual result of the correlation coefficient is shown. From these result, one can conclude that there is strong correlation between the temperature difference ($\Delta T$) of the air and the mass flow of the water ($\dot{m}$) through the heating coil. There is also a correlation between the outgoing air temperature of the heating coil (TT04) and the mass flow of the water ($\dot{m}$) through the heating coil. Below, a possible relation between the temperature difference ($\Delta T$) and the mass flow of the water ($\dot{m}$) through the heating coil is elaborated, because they have the strongest correlation.

<table>
<thead>
<tr>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>TT04-TT04</td>
<td>TT04-TT_mix</td>
<td>TT04-$\Delta T$</td>
<td>TT04-$\dot{m}$</td>
</tr>
<tr>
<td>0,2904</td>
<td>0,8280</td>
<td>0,8458</td>
<td></td>
</tr>
<tr>
<td>TT_mix-TT04</td>
<td>TT_mix-TT_mix</td>
<td>TT_mix-$\Delta T$</td>
<td>TT_mix-$\dot{m}$</td>
</tr>
<tr>
<td>0,2904</td>
<td>-0,2962</td>
<td>-0,2547</td>
<td></td>
</tr>
<tr>
<td>$\Delta T$-TT04</td>
<td>$\Delta T$-TT_mix</td>
<td>$\Delta T$-$\Delta T$</td>
<td>$\Delta T$-$\dot{m}$</td>
</tr>
<tr>
<td>0,8280</td>
<td>-0,2962</td>
<td></td>
<td>0,9935</td>
</tr>
<tr>
<td>$\dot{m}$- TT04</td>
<td>$\dot{m}$- TT_mix</td>
<td>$\dot{m}$-$\Delta T$</td>
<td>$\dot{m}$-$\dot{m}$</td>
</tr>
<tr>
<td>0,8458</td>
<td>-0,2547</td>
<td>0,9935</td>
<td></td>
</tr>
</tbody>
</table>

Table 26. Table of correlations coefficients calculated with MatLab.

Both variables are plotted in Figure 56. It can be seen that there exists a strong linear relation between the variables. To use this relation in the Simulink model of the heating coil it must be converted into a mathematical formula.
By making use of method that is being described in the Matlab Data Analysis guideline [31], the slope and the intercept of the linear predictor are determined. The Matlab-code (m-file) which is used to perform this method can be found in Appendix 5.5.

This results in the following linear formula for calculating the mass flow of the ingoing water of the heating coil based on the temperature difference between the ingoing and outgoing air temperature:

\[
m_w(t) = A \cdot \Delta T + B
\]  

Where \( m_w \) = mass flow of incoming water heating coil \([\text{kg/s}]\), \( A = 0.0512 \) \([\text{kg/s/K}]\), \( \Delta T \) = temperature difference incoming/outgoing air \([\text{K}]\), \( B = -0.0077 \) \([\text{kg/s}]\).

To verify that the linear formula above is correct the Simulink model used to determine the mass flow was adjusted. Now a function block was inserted in Simulink, which has the difference between the ingoing and outgoing air temperature of the heating coil as input parameter. The Simulink model is displayed in Figure 57. In this image a constant ingoing water temperature is shown, but in order to determine the mass flow, there is a supply water temperature used which depends on the outdoor temperature (heating curve, see section 4.2.1).

![Simulink model to calculate Tair_out with linear formula for mass flow.](image)

Also here the supply air temperature, for the whole year of 2011, one week in January 2011, one day in July 2011 and one week in November 2011, of the heating coil is simulated. The results of the simulated supply air temperature and the measured supply air temperature (TT04) are compared with each other. The graph of the whole year and of one week in November are shown in Figure 58 and Figure 59. The graphs of the other periods can be found in Appendix 9.2. Some slight deviations can be seen in the graphs, probably caused by a difference between the simulated ingoing water temperature and real ingoing water temperature and the increased ingoing air temperature caused by the ventilator in the air handling unit (which is not taken into account in the simulation).
Figure 58. Outgoing air temperature based on linear relation, one year (2011).

Figure 59. Outgoing air temperature based on linear relation, one week in November 2011.
4.2.3 Discussion

Possible reasons for the deviations in the results are pointed out below. It is not possible to validate the heating coil model with measurement data only. A reason for the deviations in the results of the simulation of the control of the heating coil could be that measurement data is used to determine the set point of the supply air temperature. The simulation of the supply air temperature, by the heating coil, causes deviations in the room air temperature. So when measurement data is used, the PI-control cannot anticipate on deviations in the room air temperature because these deviations are not observed by the controller. Therefore, the building model must be integrated in the validation model of the heating coil to obtain realistic values.

As already mentioned earlier in this report, the climate systems are controlled by the Priva building management system. Priva is based on a database of control modules. The control modules consist of flow diagrams and detailed descriptions which automatically adjust to the chosen configuration, so no detailed information about the program code and used algorithms of, for instance, the PID-controllers are known. Nothing can be said about the type of PID-controller and the configuration, for example options as described in section 2.5. Differences in the results might be caused by this lack of information, so a cooperation of Priva in future research, to obtain specific information about the control, could be an opportunity to improve the results.

Because the usage of a PI-controller caused problems during the determination of the desired supply air temperature set point, a P-controller was applied. This situation differs from the real situation and might explain the differences in the final results. The set point, that is chosen in the control part where the desired supply air temperature is determined, is a fixed set point. So, for instance, no bandwidth is applied. Also no maximum increase and decrease compared to the previous day is taken into account. However, this is the case in the real situation. This might explain differences in the results.

The calculation of the ‘starting temperature’ is not based on the real situation. This principle has been figured out by the writer of this research. Probably this ‘starting temperature deviates from the real situation and causes differences in the results.

No measurement data is available of the hydraulic (fluid) circuit. So for example, there is no information known about the supply and return water temperatures. Therefore, it is not possible to validate the supply water temperature based on the outdoor temperature (heating curve) as described above. Differences in this supply water temperature might cause deviations in the final result. When the supply air temperature is simulated, a low return water temperature is achieved. This result must be avoided, because a low return water temperature could lead to a low overall return water temperature which might influence the efficiency of the heat pumps in a negative way.

From the graphs above one can conclude that using a (linear) relation to calculate the mass flow through the heating coil, by using the temperature difference of the incoming/outgoing air, can be a good alternative for modeling the complete control of the mass flow. However, some remarks must be made. To be able to find correlations and calculate relations, several parameters must be available, namely: air temperature before coil, air temperature after coil and the mass flow of the water through the coil. In this research the mass flow (position of the valve) is simulated, but it is better to measure the mass flow (or position of the valve). The supply air temperature (temperature after the heating coil) does not longer directly depend on the air temperature in the exhibition area and the temperature set point of this area, so no direct influence of a set point is implemented. This can slow down the control and / or make it inaccurate.
5 Conclusion and Recommendations

This final chapter deals with conclusions that can be drawn from the discussions in the chapters above. Based on these conclusions, recommendations for future research are formulated.

5.1 Conclusions

Below, the results that are found in this research, and that answer the questions that are formulated in chapter 1, are discussed. The questions this research attempted to answer, and a brief answer to these questions, are described below.

1. Could the optimization of T/RH points lead to less energy consumption, while maintaining the right indoor climate conditions for the preservation of the collection and the thermal comfort of visitors and staff?

From the results in chapter 4.1 one can conclude that the optimization of temperature and relative humidity set points could possibly lead to less energy consumption while maintaining the right indoor climate conditions for the preservation of the collection and the thermal comfort of visitors and staff. Total energy savings in the range of 2 to 64% can be achieved. Eight set point settings (set point: 3, 9, 14, 15, 17, 26, 35 and 36) show possible risks for the deterioration of the collection and only one set point setting (set point 15) shows possible thermal discomfort for the visitors and staff.

2. Which optimization/combination of the temperature/relative humidity set points is the best according to energy consumption, conservation of cultural objects and comfort of staff/visitors?

Two best options for the optimization/combination of the temperature and relative humidity set points, according to energy consumption, conservation of cultural objects and comfort of staff/visitors, arise when the results in this research are analyzed. One option (set point 38) applies a bandwidth to the set points for temperature and relative humidity (± 2°C and ± 10%). This option might realize a total energy saving of approximately 55% (65 MWh). The other option is the one where temperature and relative humidity set points are applied that fluctuate with the seasons based on a sine curve (set point 41). The equilibrium position is set to 18/23°C and 40/50% and the amplitude is set to ±2°C and ± 10%. A possible total energy saving of 64% might be achieved (76 MWh). However, this option has the lowest thermal comfort (still within the limits).

3. Is it possible to reproduce the current indoor climate and air handling unit by making use of modeling software?

From the results in chapter 4.2 one can conclude that is not yet possible to exactly reproduce the current indoor climate and air handling unit by making use of modeling software. An exact validated HAMBASE model of the building is difficult to construct. Major problems occur when modeling the components of the air handling unit, especially when modeling the control of the components. However, progress is made on the field of modeling of climate control and the preliminary results look promising.

4. What are the issues, when modeling the indoor climate and air handling unit, that deserve more attention in future research?

In the final paragraph (5.2) the issues are mentioned that deserve more attention, when modeling the indoor climate and air handling unit, in future research.
5.2 Recommendation for future research

Based on the discussion and conclusion in the previous paragraph, in this section recommendations for future research are pointed out.

The implemented measures are only valid for the case study in this research, so further research should interpret the results for a wider range of museums and climate control systems.

In this research, the focus was only on the secondary climate control systems (components in the air handling unit). In future research also the primary systems, the systems that generate the heat and cold (for instance, the heat pumps), should be taken into account during the optimization. During the research it was attempted to build a complete model of an air handling unit. As a starting point, components from an older research [28] were used. While investigating and adjusting these components it was discovered that the model of the cooling coil might not function when condensation occurs. The mixing section and the humidifier were not validated because measurement data was missing. Only the heating coil and its control were investigated here. Therefore, future research could focus on the development and/or adjustment of the cooling coil, mixing section and humidifier.

While validating the HAMBASE model, the substantiation of the internal sources can be discussed. Future research may focus on other approaches, especially the method of ‘inverse modeling’, to validate the HAMBASE model and to substantiate these sources.

Future research should look at the initial costs of the climate control systems. When lower set points and/or smaller bandwidths are allowed, the climate control systems that must be installed can be less complex. Probably the initial cost will decrease.

During the simulation of the heating coil and its control, problems did occur with the PI-controllers which determine the supply air temperature and mass flow through the heating coil. A cooperation of Priva in future research, to obtain specific information about the control, could be an opportunity to improve the results. To be able to eliminate these problems measurement values of the hydraulic circuit must be made available. In order to carry out a good analysis of air handling unit 5 and the control of its components, some additional measurement data must be logged. Below a summary of the missing and useful measurement values and parameters is given:

- Temperature of the supply water of the HT cooling coil, the LT cooling coil and the heating coil
- Temperature of the return water of the HT cooling coil, the LT cooling coil and the heating coil
- Volume flow of the water through the HT cooling coil, the LT cooling coil and the heating coil
- Positions of the two-way valves of the HT cooling coil (43CV02), the LT cooling coil (43CV03) and the heating coil (43CV04)
- Percentage (0-100%) of humidity control or amount of water of humidifier (63BV05)
- Volume flow of outdoor air and or position of valve 42CD02
- Temperature and relative humidity after mixing section (outdoor air)
- Relative humidity after HT cooling coil (nearby temperature sensor 50TT05)
- Temperature and relative humidity between humidifier and dehumidifier
- Temperature and relative humidity after mixing section (after LT cooling coil)
No measurement data of the energy consumption, of the case study used in this research, are available. For a better comparison and validation of the simulation model with the real situation, energy consumption for heating, cooling, humidification and dehumidification must be made available.

In this research, only simulations are performed for the year 2011 in the Netherlands. It might be useful to perform simulations for other (climate) years and other countries (Meteonorm), because the energy consumption probably depends on the climate conditions the building is in.

Because the simulation of the complete air handling unit gradually shifted from the whole system to only the simulation of the heating coil, and eventually to the modeling of the control of the heating coil, no attention was paid to the optimizations of the control of the components. Therefore, future research could focus on the optimization of the control. One can think, for instance, of more complex and/or intelligent control applications (fuzzy logic, neural networks).

The set points are also maintained during closing hours, because the objects in the exhibition area must be exposed to the same climatic conditions the whole day. It can be taken into consideration to change the indoor climate during closing hours for saving energy.
6 References


Optimizing climate control systems for museums

M.P.E. Maas


