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BY
ALESSANDRO MACCARINI

DISSERTATION SUBMITTED 2017

AALBORG UNIVERSITY
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PREFACE

The work presented in this thesis was carried out at the Department of Energy Performance, Indoor Environment and Sustainability of Buildings, Danish Building Research Institute, Aalborg University, within the period July 1, 2013 to October 31, 2016. The study was financially supported by ELFORSK, a research and development program administrated by the Danish Energy Association.

Parts of this thesis have been accomplished during an external research stay at the Simulation Research Group, Lawrence Berkeley National Laboratory (LBNL), United States, within the period October 19, 2015 to March 18, 2016.

The opportunity of spending a period at the LBNL emerged from my participation in Annex 60, an international project conducted under the umbrella of the International Energy Agency (IEA) within the Energy in Buildings and Communities (EBC) Programme.

My sincerest gratitude goes to Professor Alireza Afshari for giving me the opportunity to pursue a doctoral degree. His trust, guidance and support truly helped me all along this journey. I would like to express my great appreciation to senior researcher Niels C. Bergsøe for always being extremely responsive in assisting me with any kind of issue. I thank everybody at the Danish Building Research Institute for a pleasant and supportive environment.

Special thanks go to Göran Hultmark from Lindab ICS, Indoor Climate Solutions. His vast engineering knowledge has been essential in conducting my research. I also would like to thank Anders Vorre, Daniel Bochen and Miroslav Dohnal from Lindab ICS for their technical support, especially during the experimental tests.

I deeply thank Michael Wetter from LBNL for hosting me and taking the time to provide me with invaluable and sharp inputs to my work. I thank also the working group of Annex 60 for all the fruitful meetings and discussions.

I thank all my friends around the world, but particularly my childhood friends. By growing up together, we have built strong bonds and I am glad I can always count on you.

I would like to thank my girlfriend Lonnie for her true love and encouragement during these years. Her smile makes every day a better day.

Last, I am forever grateful to my parents for teaching me the beauty of learning new things. Without their unconditional love, endless support and constant stimulation, I would not have been here. I thank also my grandmother for being always the happiest person in the world when I come back to Italy for holidays.

Copenhagen, 2017.

Alessandro Maccarini

ENGLISH SUMMARY

Anthropogenic greenhouse gas (GHG) emissions have increased since the pre-industrial era, driven largely by economic and population growth. According to the Intergovernmental Panel on Climate Change (IPCC), the emissions' effects are extremely likely (more than 95% probability) to have been the dominant cause of the observed warming of land and ocean surfaces since the mid-20th century. Continued emission of GHG will cause further warming and long-lasting changes in all components of the climate system, increasing the likelihood of severe, pervasive and irreversible impacts for people and ecosystems.

Energy-related activities in connection with buildings are responsible for 19% of GHG emissions worldwide. This energy use and related emissions may double or potentially even triple by mid-century. Therefore, research in the field of energy efficiency in buildings is crucial in order to reduce the global GHG emissions and mitigate the effects of the climate change.

In this thesis, a novel HVAC system for office buildings was studied. Active beams were used as terminal units for heating, cooling and ventilation. The main characteristic of the system is its ability to handle heating and cooling loads simultaneously by operating a single hydronic circuit with water temperatures of about 22 °C. This configuration shows two main benefits:

1) Waste heat from warm zones in a building can be transferred to cold zones through the hydronic circuit, reducing the total energy use.
2) Operating temperatures of about 22 °C in the hydronic circuit facilitate the use of sustainable energy sources. Therefore, savings in terms of primary energy use can be obtained with a consequent reduction of GHG emissions.

To analyze the energy performance of the system, a simulation-based research project was conducted. A detailed mathematical model of the system was developed with the programming language Modelica. This represents the first application of Modelica for HVAC systems integrating active beams as terminal units. Dedicated control strategies were designed in order to regulate the hydronic circuit. The model was exercised through a series of simulation experiments.

Simulation results illustrated that the energy savings due to heat transfer between building zones through the hydronic circuit varied between 7% and 27% depending on factors such as occupancy level and climate location.

Because the system works with water temperatures close to ambient temperatures, devices such as heat pumps and dry coolers operate more efficiently. Results showed that the heating seasonal coefficient of performance (COP) of a reversible air-to-water heat pump integrated into the novel system was 48% higher than the heating seasonal COP of the same heat pump when integrated in a

conventional system operating with standard water temperatures. In addition, when using a dry cooler to take advantage of free cooling conditions, cooling energy savings of approximately 70% occurred in the two-pipe system versus approximately 30% in a traditional four-pipe system. When considering the total annual primary energy use, savings of between 12% and 18% were achieved.

DANSK RESUME

Udledning af menneskeskabte drivhusgasser er steget siden den præindustrielle tidsalder, i hovedsagen på grund af vækst i økonomi og befolkning. Ifølge FN's klimapanel (IPCC, Intergovernmental Panel on Climate Change) er det yderst sandsynligt (> 95 % sandsynlighed), at virkningerne fra udledningen har været den primære årsag til den observerede opvarmning af jord- og havoverflader siden midten af det 20. århundrede. Fortsat udledning af drivhusgasser vil forårsage yderligere opvarmning og langvarige ændringer i alle dele af klimasystemet, og øge sandsynligheden for alvorlige, vidt udbredte og uoprettelige konsekvenser for mennesker og økosystemer.

Bygnings- og energirelaterede aktiviteter er ansvarlige for 19 % af verdens udledning af drivhusgasser. Denne brug af energi og den tilknyttede udledning kan blive fordoblet eller potentielt endog tredoblet omkring midten af dette århundrede. Af denne grund er forskning inden for bygningers energieffektivitet afgørende for nedbringelse af den globale udledning af drivhusgasser og modvirkning af klimaændringer

I denne afhandling, blev et innovativt varme-, køle- og ventilationssystem til kontorbygninger undersøgt. Aktive kølebafler blev brugt som armaturer til opvarmning, køling og ventilation. En væsentlig egenskab ved systemet er dets evne til at håndtere varme- og kølebelastninger simultant ved hjælp af kun et vandbaseret kredsløb med vandtemperaturer på ca. 22 °C. Denne udformning indebærer to primære fordele:

1) Overskudsvarme fra varme zoner i en bygning kan overføres til kolde zoner hvorved det samlede energibehov reduceres.
2) Driftstemperaturer på omkring 22 °C i det vandbårne kredsløb letter integrationen af vedvarende energikilder. Af denne grund kan der opnås besparelser i primært energiforbrug med deraf følgende reduktion af udledningen af drivhusgasser.

Med henblik på afdækning af systemets energimæssige ydeevne blev analyser baseret på simuleringer gennemført.

En detaljeret matematisk model af systemet blev udviklet med programmeringssproget Modelica. Modellen repræsenterer den første anvendelse af Modelica til HVAC-systemer, hvor aktive kølebafler anvendes som afslutningskomponent. Dedikerede reguleringsstrategier blev udviklet med henblik på regulering vandtemperaturen i vandbårne kredsløb. Modelica-modellen af det nye system blev bragt i anvendelse ved en række simuleringer.

Resultaterne af simuleringerne viste, at energibesparelserne hidrørende fra varmeoverføringen mellem rum varierede mellem 7 % og 27 %, afhængigt af faktorer som fx personbelastning og udeklima.

Som følge af at systemet arbejder med vandtemperaturer tæt på omgivelsernes temperatur, fungerer aggregater så som varmepumper og tørkølere mere effektivt. Simuleringsresultater viste, at den årlige COP-værdi af en reversibel luft-til-vand varmepumpe integreret i det nye system var 48 % højere, end når den samme varmepumpe var integreret i et konventionelt system med typiske vandtemperaturer. Endvidere, anvendelse af en tørkøler med henblik på udnyttelse frikøling viste energibesparelser til køling på ca. 70 % og ca. 30 % henholdsvis for det nye system og for et konventionelt system. Betragtes det totale årlige primære energiforbrug, blev der opnået besparelser på mellem ca. 12 % og 18 %.

LIST OF ORIGINAL PUBLICATIONS

This thesis has been submitted for assessment in partial fulfilment of the PhD degree. The thesis is based on the submitted and published scientific papers, which are listed below. Parts of the papers are used directly or indirectly in the extended summary of the thesis.

I. **Paper I.** A. Maccarini, G. Hultmark, A. Vorre, A. Afshari, and N. C. Bergsøe. (2015). *Modeling of active beam units with Modelica*. Building Simulation, vol. 8, no. 5, 543–550. doi:10.1007/s12273-015-0236-5.

II. **Paper II.** A. Maccarini, A. Afshari, G. Hultmark, N. C. Bergsøe, and A. Vorre. (2016). *Modeling of a novel low-exergy system for office buildings with Modelica.* In Proceedings of the 12th REHVA World Congress, CLIMA 2016, 23–26 May, 2016, Aalborg, Denmark.

III. **Paper III.** A. Maccarini, A. Afshari, G. Hultmark, N. C. Bergsøe, and A. Vorre. (2016). *Development of a new controller for simultaneous heating and cooling of office buildings.* In Proceedings of the 9th IAQVEC Conference, 24–26 Oct, 2016, Seoul, South Korea.

IV. **Paper IV.** A. Maccarini, G. Hultmark, A. Afshari, N. C. Bergsøe, and A. Vorre. (2016). *Transferring heat among building zones through a room-temperature water loop – Influence of climate and occupancy.* Submitted to Building Simulation. (Accepted). doi:10.1007/s12273-017-0358-z.

V. **Paper V.** A. Maccarini, M. Wetter, G. Hultmark, A. Vorre, A. Afshari, and N. C. Bergsøe. (2017). *Energy savings potential of a two-pipe system for simultaneous heating and cooling of office buildings.* Energy and Buildings 134, 234-247. doi:10.1016/j.enbuild.2016.10.051.

VI. **Paper VI.** A. Maccarini, G. Hultmark, A. Afshari, and N. C. Bergsøe. (2017). *Analysis of control strategies for a novel HVAC system equipped with a room-temperature water loop.* Submitted to the 15th Building Simulation Conference (BS2017) organized by the International Building Performance Simulation Association (IBPSA). (Accepted)

Publications that are part of the PhD study but not included in the thesis are:

VII. **Paper VII.** A. Maccarini, A. Afshari, N. C. Bergsøe, G. Hultmark, M. Jacobsson, and A. Vorre. (2014). *Innovative two-pipe active chilled beam system for simultaneous heating and cooling of office buildings*. In Proceedings of the 13th International Conference on Indoor Air Quality and Climate, Indoor Air 2014, 7–12 July, 2014, Hong Kong.

NOMENCLATURE

LATIN LETTERS

A	Coil heat transfer area [m^2]
A_{zone}	Thermal zone area [m^2]
B, c_1, c_2	Empirical coefficients
$c_{p,a}$	Specific heat capacity of air [J/kgK]
$c_{p,w}$	Specific heat capacity of water [J/kgK]
ED_{coo}	Energy demand (cooling) [J]
ED_{hea}	Energy demand (heating) [J]
EG_{coo}	Energy generated (cooling) [J]
EG_{hea}	Energy generated (heating) [J]
h_i	Enthalpy in the node i [J/Kg]
HL	Heat loss rate for transmission, infiltration and/or ventilation [W/K]
H_{ret}	Enthalpy flow rate entering the thermal plant [W]
$H_{ret,i}$	Enthalpy flow rate leaving the active beam [W]
H_{sup}	Enthalpy flow rate leaving the thermal plant [W]
$H_{sup,i}$	Enthalpy flow rate entering the active beam [W]
IC	Influence coefficient [-]
IP_{BC}	Input of base case [-]
k	Coil heat transfer coefficient [W/m^2K]
k_{coo}	Cooling temperature off-set [°C]
k_{flow}	Water flow coefficient [-]
k_{hea}	Heating temperature off-set [°C]
L	Coil length [m]
L_{coo}	Equivalent beam length needed to meet cooling peak loads
L_{hea}	Equivalent beam length needed to meet heating peak loads
L_{ven}	Equivalent beam length needed to provide enough outdoor air
L_{zone}	Equivalent beam length in a zone
$\dot{m}$	Mass flow rate [m^3/s]

$\dot{m}_{ai}$	Induced air mass flow rate [kg/s]
$\dot{m}_{ap}$	Primary air mass flow rate [kg/s]
$\dot{m}_{ap,e}$	Primary air mass flow rate per coil length [L/sm]
$\dot{m}_{i}$	Water mass flow rate in the node *i* [kg/s]
$\dot{m}_{nom}$	Nominal water mass flow rate [kg/s]
$\dot{m}_{w}$	Water mass flow rate [kg/s]
$\dot{m}_{vent}$	Outdoor air mass flow rate [kg/s]
N_{people}	Number of people [-]
OP_{BC}	Output of the base case [-]
p	Pressure in the air plenum [Pa]
$\dot{Q}$	Heat flow rate provided by heating/cooling system [W]
$Q_{heating}$	Space heating demand [W]
$\dot{Q}_{inf/ven}$	Heat losses through infiltration and/or ventilation [W]
$\dot{Q}_{int}$	Heat gains from people, lighting and equipment[W]
$\dot{Q}_{mech}$	Mechanical useful energy use [W]
$\dot{Q}_{a}$	Active beam heating/cooling capacity provided by primary air [W]
$\dot{Q}_{sol}$	Heat gains from solar radiation [W]
$\dot{Q}_{tot}$	Total active beam heating/cooling capacity[W]
$\dot{Q}_{tra}$	Heat losses through transmission[W]
$\dot{Q}_{w}$	Active beam heating/cooling capacity provided by water coil [W]
T_{a}	Primary air temperature [°C]
$T_{a,in}$	Induced air temperature entering the coil [°C]
$T_{a,out}$	Induced air temperature leaving the coil [°C]
T_{b}	Balance point temperature [°C]
T_{i}	Water temperature in the node *i* [°C]
T_{o}	Outdoor air temperature [°C] [K]
T_{r}	Room air temperature [°C] [K]
T_{ret}	Water temperature entering the thermal plant [°C]
$T_{ret,i}$	Water temperature leaving the active beam [°C]
T_{sup}	Water temperature leaving the thermal plant [°C]
$T_{sup,i}$	Water temperature entering the active beam [°C]

$T_{w,avg}$	Average water temperature [°C]
$T_{w,in}$	Water temperature entering the coil [°C]
$T_{w,out}$	Water temperature leaving the coil [°C]
$X_{heating}$	Exergy demand of a room [W]

GREEK LETTERS

ΔIP	Change of the input [-]
Δp	Pressure drop in water or air circuit [Pa]
ΔOP	Change of the output[-]
ΔT_w	Water temperature difference [K]

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CHAPTER 1. INTRODUCTION

Commercial and residential buildings account for approximately 40% of the total end-use of energy [1]. About half of this energy is used to operate heating, ventilation and air-conditioning (HVAC) systems [2]. Such systems aim to maintain a comfortable indoor environment with room air temperatures of about 20 °C, which is close to ambient temperature. Because of the low temperature level, the exergy demand for room conditioning is low [3].

For simplicity, consider the case of a room equipped with a heating system. The exergy demand of the room can be defined as

$$\boldsymbol{X_{heating} = Q_{heating}\left(1 - \frac{T_0}{T_r}\right)} \qquad (\mathbf{1.1})$$

Where $X_{heating}$ is the heating exergy demand [W], $Q_{heating}$ is the space heating demand [W], T_o is outdoor (environmental) air temperature [K], and T_r is the room air temperature [K] [4].

To be able to supply heat to the room, it is enough that the heating system operates with temperatures that are slightly higher than T_r. However, in most cases, heating systems operate with temperatures that are significantly higher than T_r. Such systems are commonly known as high-exergy systems. These require heat at high temperatures, which is produced by using high valued energy delivered by fossil fuels. Extensive usage of fossil fuels causes several environmental and health issues, such as global warming and pollution [5].

1.1. LOW-EXERGY SYSTEMS

Low-exergy building energy systems are defined as systems that provide heating and cooling at a temperatures close to room temperature [6]. This allows the employment of low valued energy, which can be delivered by sustainable energy sources such as waste heat, river/lake water, solar energy, geothermal applications and heat pumps with a high coefficient of performance (COP). Therefore, the use of low-exergy systems can reduce the environmental impact of buildings.

Various studies on low-temperature heating systems have been carried out in the past years. Such systems are mainly based on applications with a high fraction of radiative heat distribution. Hasan et al. [7] analyzed the performance of a heating system with nominal supply/return water temperatures of 45 °C/35 °C. This system included radiators in rooms and floor heating in bathrooms. Kazanci et al. [8] analyzed the exergy performance of different heating systems, including a floor heating system with a supply water temperature of 33 °C and a radiator heating system with a supply water temperature of 45 °C. Hesaraki et al. [9] conducted an experimental study in low-temperature hydronic systems. A ventilation radiator with required supply water temperature of 30 °C and a floor heating system with required supply water temperature of 33 °C were compared to a baseline system. Sakellari and Lundqvist [10] modeled a low-temperature heating system in which the heat pump operated with a supply water temperature of 28 °C.

Several works have also been carried out in relation to high-temperature radiative cooling systems. Bejček [11] simulated the performance of an absorption solar cooling system connected to radiant-cooled ceiling elements with chilled water temperature of 10-12 °C. Kazanci et al. [12] compared the exergy performance of different cooling systems, including water-based radiant cooling systems with supply water temperatures of about 17-19 °C. Zhao et al. [13] analyzed the performance of a radiant cooling system with supply water temperature of 18 °C in a large-space building in China. Lehmann et. al [14] presented application range and functionality of thermal-activated building systems (TABS) with supply water temperature of about 19 °C.

In the context of convective technologies, some works have studied the use of active beams. For these, water supply temperatures are usually in the range of 14-18 °C for cooling and 35-45 °C for heating [15].

1.2. ACTIVE BEAM SYSTEMS

Active beam systems have been used for more than 20 years in Europe, mainly for cooling purposes, and interest in these systems has increased in North America and Asia during the last decade. These systems incorporate active beams as terminal units. Active beams are devices able to provide outdoor air, sensible heating and sensible cooling to a space. Fig. 1-1 shows the schematic diagram of a typical active beam.

An active beam unit consists of a primary air plenum, a mixing chamber, a heat exchanger and several nozzles. Typically, an air-handling unit supplies primary air to the active beams. The primary air is discharged to the mixing chamber through the nozzles. This generates a low static pressure region which induces air from the room up through the heat exchanger, where hot or cold water is circulating. The conditioned induced air is then mixed with primary air, and the mixture is supplied to the space.

The main benefits of active beam systems are related to low energy use, high integration of sustainable energy sources, smaller floor-to-floor heights and quiet operation. To fully understand the performance of active beam systems, several research studies have been conducted in the past years. These studies mainly focused on the description of fundamentals performance, humidity control in hot and humid climate, air distribution in rooms and energy use.

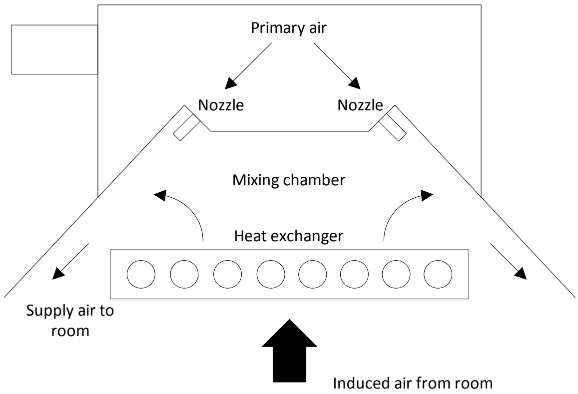


Figure 1-1 Schematic diagram of an active beam terminal unit

Virta et al. [15] published a comprehensive guidebook for chilled beam performance and applications. The book provides insights regarding theoretical aspects, system design and thermal comfort. Livchak et al. [16] introduced a parameter that represents the performance of active beam units. The so-called coil output to primary airflow ratio (COPA) represents the amount of cooling (or heating) produced by the active beam coil per volume of primary air used. The higher the COPA is, the more efficient the active beam design is, and thus the primary air is used more effectively. Guan et al. [17] performed a geometric optimization of an active beam unit in order to achieve a high induced ratio. By changing the geometry of the mixing chamber and lengthening the nozzles, the modified structure can increase the induced ratio by 30% with the same working conditions and primary air volume flow rate. Filipsson et al. [18] investigated how the induced ratio is influenced by operating conditions such as temperature and flow rate of chilled water, primary air flow rate and internal heat gains.

With regard to humidity control, Loudermilk et. al. [19] suggested alternative configurations for air handling units in humid climate. They showed that relaxing space design humidity level, or secondary moisture removal can successfully be used to control humidity and prevent condensation. Kosonen et al. [20] studied the feasibility of active beam systems in a tropical climate. It was concluded that it is possible to prevent condensation in beams and reach dry cooling by assuring proper supply air flow rates and low infiltration rates.

As for air distribution in rooms, Rhee et al. [21] investigated the thermal uniformity when an active chilled beam system is applied to an open-plan office. Experiments in a test bed showed that chilled beams can achieve an acceptable thermal uniformity, with less air flow rate than conventional air distribution systems. Koskela et al. [22] analyzed the air flow patterns in an open-plan room in Finland. The experiments were conducted with different internal heat loads assuming summer, winter, and spring/autumn conditions. It was shown that the internal heat loads had a significant influence on flow patterns and draught risks.

When it comes to the energy use, active chilled beam systems can be more efficient than traditional VAV systems, mainly because ventilation requirements are handled by the primary air, while most of the sensible cooling loads are treated by the chilled water [23]. In addition, with higher chilled water temperature (14 °C to 18 °C) than VAV systems (4 °C to 7 °C), the dedicated chiller in active beam systems can operate at higher COP. Roth et al. [23] argued that active beam systems can achieve energy-savings of 10-20% when compared to traditional HVAC systems. According to a report of the American Council for an Energy-Efficiency Economy, energy savings of about 20% can be estimated when comparing active chilled beam systems with traditional VAV systems [24]. Murphy et al. [25] conducted a simulation experiment to calculate the energy savings related to the application of chilled beam systems in the USA. Depending on the climate location, the active chilled beam system used 7-15% less energy than a conventional VAV system. Stein et al. [26] argued that active chilled beam systems with low primary air flow and medium temperature chilled water might be more efficient than VAV reheat systems in buildings with high sensible loads and located in extreme climates where outdoor air economizers are not effective.

In the abovementioned studies, when referring to active beam systems, it is implied that a standard hydronic circuit configuration is applied.

1.2.1. COMMON HYDRONIC CONFIGURATIONS

According to Fig. 1-2, the hydronic circuit of active beam systems consists of three main parts: generation, distribution and consumer loads. The distribution is typically available in a two-pipe or four-pipe configuration.

A two-pipe configuration (also known as change-over configuration) includes only one supply and one return pipe for distribution of water. This means that all building zones receive either cold water or hot water. Thus, the entire building is in either heating mode or cooling mode. Conversely, a four-pipe configuration includes two supply pipes and two return pipes for water. As a consequence, some zones can receive cold water while other zones receive hot water, meaning that heating and cooling can be provided simultaneously.

Generally, when using a configuration with two distribution pipes, the thermal comfort requirements cannot be always fulfilled, and for systems implementing four distribution pipes, the installation effort is high.

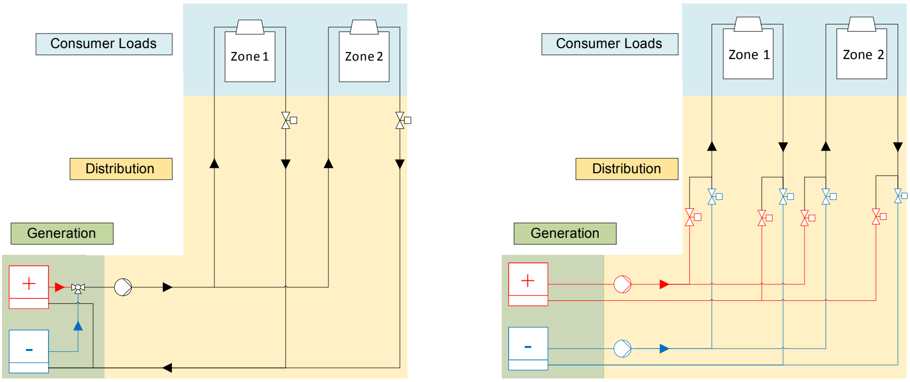


Figure 1-2 Common two-pipe (left) and four-pipe (right) hydronic configuration

1.2.2. INNOVATIVE HYDRONIC CONFIGURATION

This study aims to investigate the functionality of an innovative system that merges the installation simplicity of a traditional two-pipe configuration with the flexibility of a four-pipe configuration.

Fig. 1-3 shows the schematic diagram of the innovative system. The distribution circuit is analogous to the one of a traditional two-pipe configuration.

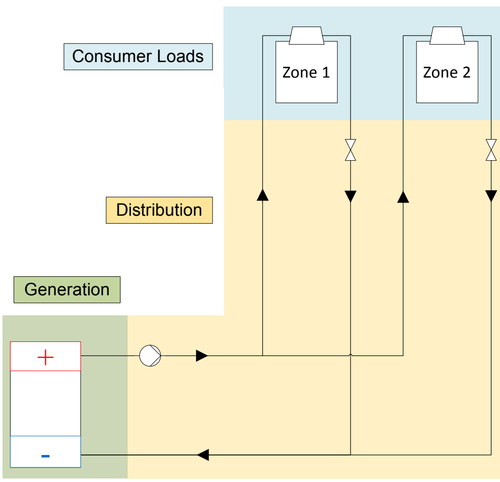


Figure 1-3 Innovative two-pipe system

However, the generation part differs in terms of operating temperatures. Usually, a traditional two-pipe system circulates supply water at a temperature of about 45 °C and 14 °C, respectively, in heating and cooling mode. Contrastingly, the innovative two-pipe system operates a supply water temperature of about 22 °C all year round. A room with an indoor temperature of 20 °C would be heated, while a room at 24 °C would be cooled. The return pipes from the single zones are mixed together and the common return pipe is sent to the central plant, which will operate in either heating or cooling mode, depending on the average resulting thermal loads in the building.

Beside the advantages in terms of exploitation of sustainable energy sources, operating such water temperatures opens opportunities for transferring heat among building zones when simultaneous heating and cooling occurs in a building. By mixing the return pipes from individual zones, excess heat can be transferred from warm to cold zones through the water circuit. This behavior leads to a reduction of the annual energy use when comparing the innovative two-pipe system with a traditional four-pipe system. This evidence will be explained by means of a stationary load situation.

1.2.3. A STATIONARY EXAMPLE

As an example, two office rooms are considered: one perimeter room[1] and one interior room. Each room is equipped with an active beam unit and occupied by one person. The supply water temperature is maintained at a temperature of 22 °C and delivered to both active beam units. Outdoor climate conditions correspond with a typical winter day in a temperate climate. It is reasonable to assume that the perimeter room requires heating while the interior room requires cooling. Fig. 1-4 illustrates an example of such temperature conditions.

It is assumed that in the perimeter zone, the water cools off by 1 K, while in the interior zone it warms up by 1 K. Assuming an equal water flow rate through the beams of 0.04 Kg/s, the common return water temperature corresponds to the mean of the two return water temperatures, i.e. 22 °C. This means that, neglecting heat losses from the pipes, no power is required by the central plant since return and supply water temperatures are equal.

Contrastingly, in a four-pipe configuration, a heater has to warm up the return water flow from the perimeter zone, while a cooler has to cool down the return water flow from the interior zone. The heater would require 167 W while the cooler would require -167 W.

[1] The term ”perimeter room” refers to a room of a building that has at least one vertical wall facing outdoor. The term “interior room” refers to a room of a building that does not have any vertical wall facing outdoor. See Figure 1-6.

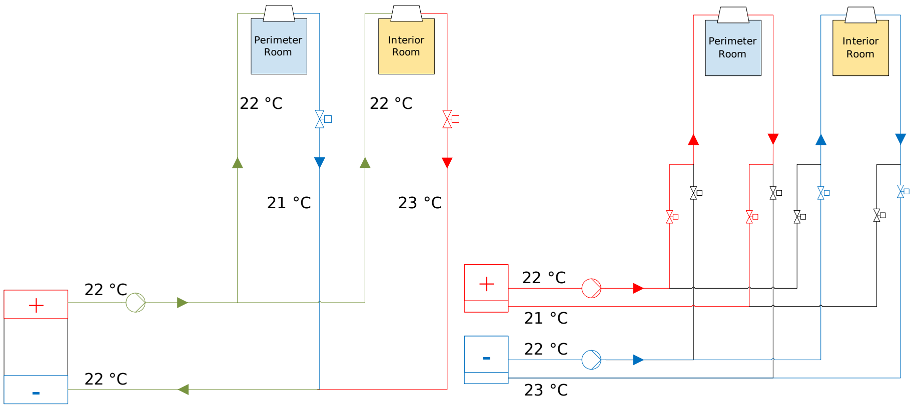


Figure 1-4 Stationary example: two-pipe system vs. four-pipe system

This example illustrates a case of perfect equilibrium between heating and cooling loads in the building. However, factors such as climate, distribution of rooms, occupancy and thermal properties of the building envelope lead to a building being likely to have a diversity of heating and cooling loads in its different rooms.

1.2.4. PILOT EXPERIMENTAL STUDY

In order to obtain initial indications of the basic functionality of the novel two-pipe system, a pilot experimental study was carried out by the Lindab ICS team in Farum, Denmark.

Two identical test rooms were set up to operate as typical office rooms. The rooms had dimensions: 4 m length, 4 m width and 3 m height. One room was equipped to operate in heating demand mode, while the other was equipped to operate in cooling demand mode. Both rooms were supplied with heat sources of 350 W representing internal heat gains from people, equipment and lighting. In addition, the room with cooling demand was furnished with a 400 W radiator. This was done to simulate solar gains flowing into the room.

The indoor air temperature surrounding the facility was about 21 °C, therefore it was not possible to passively establish a situation with heating demand. A cold surface with area of 7 m^2 and fixed temperature of about 13.5 °C was mounted behind one of the walls of the heating-demand room. One *Solus* active beam unit was mounted in each room. This unit was specially manufactured by Lindab A/S to operate with low temperature differences between water and room air in both heating and cooling mode [27].

Fig. 1-5 shows the values of the parameters measured in the experiment under steady-state conditions. The common supply water temperature was 21.9 °C. Return water temperature from the cooling-demand room was 22.8 °C while the return water temperature from the heating-demand room was 21.2 °C.

After the two return water flows mixed, the resulting total return water temperature was 22.1 °C, leading to a temperature difference between supply and return of about 0.2 K. With these conditions, the thermal power required by the plant was approximately 64 W (cooling power). If the same conditions were applied to a four-pipe configuration, the thermal power required by the heater would have been approximately 111 W, while the thermal power required by the cooler would have been 143 W.

Note that the room air temperature of the heating-demand room is higher than the return water temperature *Water OUT 1*. This is theoretically not possible, and might be explained by the accuracy of the temperature sensor, which is 0.2 °C. Another reason might be related to the fact that the room air temperature was measured only at one point on the horizontal plane, and its value might have been influenced by the location of the heat sources. Therefore, if air temperature had been measured at other points of the room, it might have been lower.

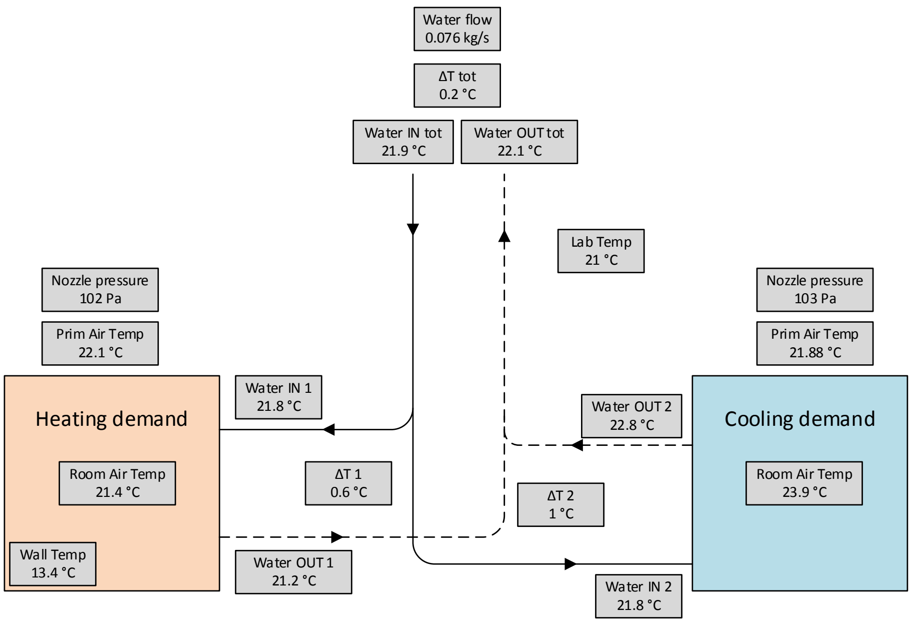


Figure 1-5 Experimental setup of the two-pipe system.

1.3. SIMULTANEOUS HEATING AND COOLING

In the previous sections it was shown that the innovative two-pipe system is able to reduce the total energy use by taking advantage of situations of simultaneous heating and cooling demand. It is therefore relevant to show why these situations may occur in a building.

A useful approach to present the reasons behind the simultaneous need for heating and cooling of buildings is the balance point temperature [28] [29], which is briefly introduced in the following.

1.3.1. BALANCE POINT TEMPERATURE

The balance point temperature is defined as the outdoor temperature at which total losses from conditioned spaces equal heat gains. In other words, this is the outdoor air temperature required for the indoor temperature to be comfortable without the use of any mechanical heating or cooling [28] [29]. The heat balance in a building can be expressed by:

$$\dot{Q}_{sol} + \dot{Q}_{int} + \dot{Q} - \dot{Q}_{tra} - \dot{Q}_{inf/\,vent} = 0 \qquad (1.2)$$

Where:

$\dot{Q}_{sol}$ is the heat gains from solar radiation [W]

$\dot{Q}_{int}$ is the heat gains from people, lighting and equipment [W]

$\dot{Q}$ is the heat provided by heating/cooling system [W]

$\dot{Q}_{tra}$ is the heat loss through transmission [W]

$\dot{Q}_{inf/vent}$ is the heat loss through infiltration and/or ventilation [W]

If we assume that no mechanical heating or cooling is needed, then $\dot{Q} = 0$. By revealing the outdoor and indoor air temperatures, Eq. (1.2) becomes:

$$\dot{Q}_{sol} + \dot{Q}_{int} - HL(T_r - T_0) \qquad (1.3)$$

Where:

HL is the heat loss rate for transmission, infiltration and/or ventilation [W/K]

T_r is the room air temperature [°C]

T_o is the outdoor air temperature [°C]

Finally

$$\boldsymbol{T_0 = T_b = T_i - \frac{\dot{Q}_{sol} + \dot{Q}_{int}}{HL}} \qquad (\mathbf{1.4})$$

Where T_b is the balance point temperature.

The balance point temperature can be calculated for each space in a building, and by comparing them, it is possible to highlight situations where there is a simultaneous need for heating and cooling in the whole building. The following example aims to provide a qualitative overview of this concept.

Fig. 1-6 illustrates the floor layout of a typical two-story office building [28].

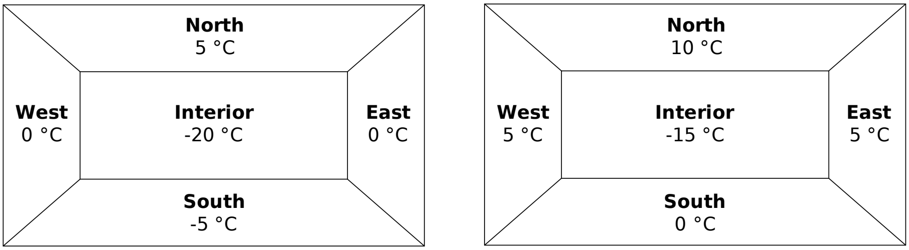


Figure 1-6 Two-story office building. Ground floor (left) and top floor (right). The balance point temperatures are shown for each zone. Adapted from [28]

Interior zones usually have low balance point temperatures, meaning that cooling is required even during the coldest day. The north-oriented zone on the top-floor has the highest balance point temperature, which is 10 °C. Therefore, whenever the outdoor temperature is between -20 °C and 10 °C, simultaneous heating and cooling demand occurs in the building, as illustrated in Fig. 1-7.

Generally, it can be concluded that interior zones of a building tend to overheat due to waste heat generated by internal factors (people, lighting and equipment), while perimeter zones require heating due to heat losses through windows, walls and infiltration. In reality, the continuous variation of factors such as solar gains and occupancy instantaneously affects the balance point temperature of each zone, making it difficult to provide a general value.

In large office buildings consisting of several single-office rooms, simultaneous heating and cooling demand might occur even among office rooms located on the same facade. This is because the occupancy patterns of office rooms have a stochastic behavior [30] [31]. For example, consider a typical spring day in a mild climate (outdoor air temperature of about 13 °C) and two office rooms located on the north facade. Both rooms are occupied by one person and they require cooling to keep the air temperature below the set-point (see Fig. 1-7). If one person leaves his/her office while the other one remains seated, a simultaneous need for heating and cooling might occur. Due to the absence of internal heat gains, the empty room increases its balance point temperature, most probably above 13 °C.

It is worth mentioning that the concept of balance point temperature was merely presented in this work to illustrate a simple mathematical explanation for why large office buildings may require simultaneous heating and cooling demand.

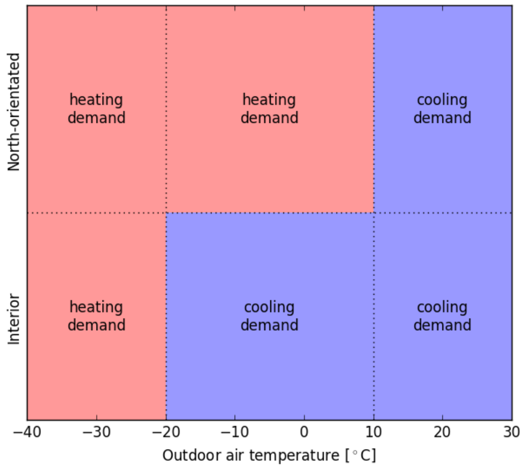


Figure 1-7 Simultaneous heating and cooling demand

1.3.2. SIMULTANOUES HEATING AND COOLING IN THE LITERATURE

Few works were found in the literature regarding the exploitation of simultaneous heating and cooling to reduce the energy use of buildings.

Byrne et al. [32] developed a heat pump able to carry out simultaneously heating and cooling with the same energy input. In the simultaneous mode, hot and cold water is produced using the water condenser and the water evaporator. A simulation study conducted by the same authors showed that the integration of the new heat pump into a HVAC system for a hotel building led to annual electricity savings of about 50% when considering space heating, space cooling and domestic hot water production. The high electricity savings were due to the fact that the domestic hot water production was predominant in the energy balance of the building and this was covered largely by the heat pump.

Karlsson [33] studied the possibility of transferring excess heat between a south-facing room and a north-facing room via a floor heating system. The two-room building model was located in Sweden and heating energy use decreased by about 3% during the months of March and April.

Le Dreau and Heiselberg [34], used capillary tubes embedded in the surface of walls to exchange heat between a south-facing room and a north-facing room. Simulation results showed that energy savings of about 5% occurred and an improvement in the indoor climate was observed due to thermal homogenization in the building.

No literature was found regarding the development of HVAC systems able to provide simultaneous heating and cooling through a single hydronic circuit and using active beams as terminal units.

1.4. OBJECTIVE

The objective of this work was to analyze the energy performance of a novel two-pipe system able to provide simultaneous heating and cooling with a water loop that is near the room temperature. Active beams are integrated in the system as terminal units. Fig. 1-8 shows a diagram of the system.

The work was mainly performed using mathematical modeling and simulation-based techniques. Experimental analysis and measurements were carried out in a laboratory environment to verify a mathematical model of an active beam unit developed within this thesis.

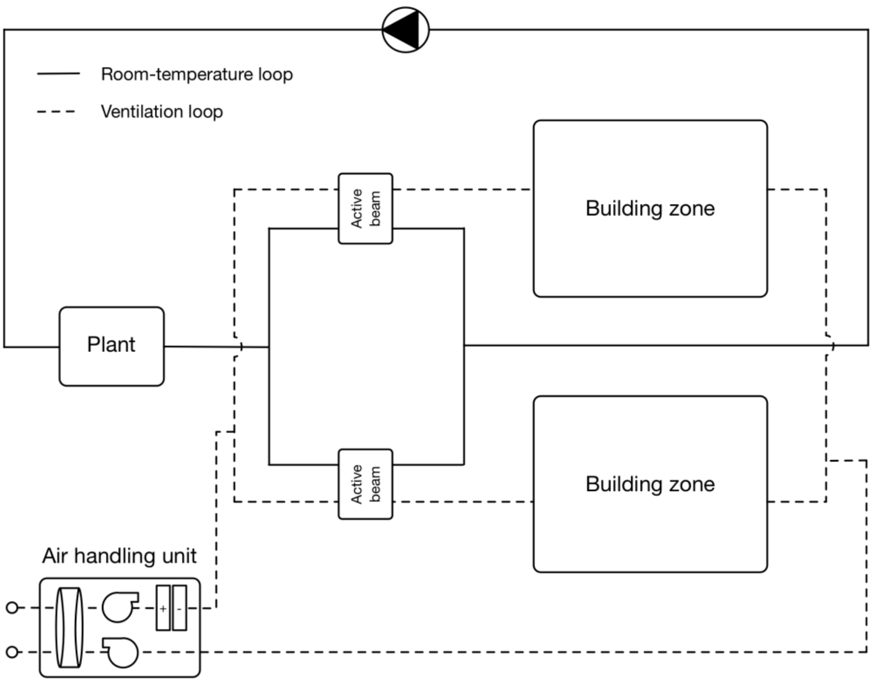


Figure 1-8 Diagram of the HVAC system investigated

1.5. RESEARCH QUESTIONS

This thesis aims to answer to the following research questions:

1) Is it possible to design a well-functioning active beam system that operates with only one water loop for both heating and cooling?
2) Is this system able to transfer useful heat among building zones through the water loop? How much useful heat can be transferred? Which are the main factors influencing the useful heat transfer?
3) How can this system be regulated in order to achieve comfortable indoor climate conditions?
4) Which kind of thermal plant could be integrated into the two-pipe system to take advantage of sustainable energy sources?
5) How does the two-pipe system perform in terms of annual primary energy use when compared with a traditional four-pipe system?

1.6. LIMITATIONS

When this PhD study started in the summer of 2013, no building was equipped with the proposed two-pipe system. Only recently (early 2016), has a full-scale two-pipe system started to operate in an office building in Jönköping, Sweden. Therefore, limitations of this thesis primarily relate to the fact that measurement data of a full-scale two-pipe system are not included.

Since the novelty of the two-pipe system is related to the water circuit, basic assumptions were made regarding the ventilation circuit. In particular, the simulations were performed by using a constant air volume (CAV) system. Advanced control strategies for the ventilation circuit (e.g. demand-controlled ventilation or variable air volume systems) might lead to further improvements of the general performance of the two-pipe system.

CHAPTER 2. METHODS

The first part of this chapter introduces the role of modeling and simulation techniques in the context of buildings and HVAC systems. In the second part, the development of the two-pipe system model with Modelica is presented. The third part gives an overview of the simulation experiments performed on the model.

2.1. COMPUTER MODELING AND SIMULATION IN ENGINEERING

One of the goals of engineering research is the analysis of complex systems. A system can be defined as a collection of interrelated objects forming a unified whole. A space shuttle, a car or a nuclear plant are all examples of systems.

The direct observation and analysis of a system through experiments is the most straightforward way of understanding its behavior. However, performing such experiments can be impractical or unfeasible. Experiments might be too expensive or the system might not yet exist.

Often, a model of the system can be constructed and used for investigation. There are different kinds of models, but, in engineering studies, mathematical models are usually adopted. A mathematical model is a description of a system where the relationships between the variables of the system are expressed in mathematical form. This often includes a combination of ordinary differential equations (ODEs) and algebraic equations (AEs). ODEs typically represent the governing equations, while AEs act as constraints. The resulting set of equations is known as a system of differential algebraic equations (DAEs) [35] [36].

Particularly useful are the mathematical models built with the aid of computer programs. The process of executing and performing experiments on the model is called simulation. Typically, the output takes the form of time trajectories which represent the system behavior. Thus, if the right model assumptions, parameters and inputs are used, simulations predict how the real system would behave if these very same experiments were performed on it. Fig. 2-1 illustrates the concept of modeling and simulation.

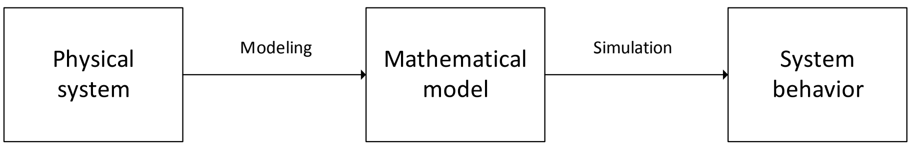


Figure 2-1 Modeling and simulation

2.1.1. BUILDING SIMULATION PROGRAMS AND PREVIOUS STUDIES

Over the past decades, a wide range of building simulation programs (BSPs) have been developed and used by the building research community and by building designers [37]. Based on a mathematical model that describes the interaction between a HVAC system and a building, these tools perform simulations and calculate outputs in terms of energy use, thermal comfort, daylighting etc. [38]. A comparison of twenty major BSPs can be found in [39]. When used appropriately, these tools have the potential to improve competitiveness, productivity, quality and efficiency in buildings as well as facilitating future innovation and technology [40].

Previous studies analyzed the energy performance of the two-pipe system by using BSPs. Afshari et al. [41] developed a model of the system with BSim [42], one of the most commonly used tools in Denmark. The use of BSim showed two key limitations. First, BSim does not include any terminal unit clearly defined as active beam. Therefore, the system was simplified by modeling fan coils for cooling and radiators for heating. Second, BSim treated heating and cooling as two separate processes. Therefore, the energy performance of the system could only be calculated by making some assumptions in a post-processing analysis. Maccarini et al. [43] investigated the possibility of modeling the system in EnergyPlus [44], a whole building energy simulation program. Simulations with EnergyPlus allowed a wider understanding of the energy behavior of the system primarily because EnergyPlus includes a specific terminal unit defined as active beam. However, also EnergyPlus considered heating and cooling as two separate processes. In addition, in both studies, limitations were found in respect to the modeling of the controller for the regulation of the room-temperature water loop.

It is worth highlighting that both BSim and EnergyPlus can be defined as traditional BSPs. Traditional BSPs are usually written using an imperative language, such as FORTRAN, C and C++ [45]. In these programs, a developer writes sequences of computer instructions for algebraic equations, differential equations and numerical solution algorithms. Thus, code that describes the physical behavior of components is mixed with code for numerical solution methods. Tight coupling of numerical solution methods with model equations makes it difficult to add new models to these programs and support new use cases [46].

In the present study, it became clear that traditional BSPs were not suitable, and a more flexible modeling and simulation tool was necessary for a comprehensive investigation of the two-pipe system.

2.1.2. EQUATION-BASED MODELING - MODELICA

The use of equation-based languages in the buildings research community has its origin in the ’80s with the birth of projects that led to the development of programs such SPARK [47] [48] and IDA ICE [49] [50].

In 1997, Mattson and Elmqvist reported on an international effort to design Modelica [51]–[54], a freely available, object-oriented equation-based language for modeling large, complex, and heterogeneous physical systems. In the last two decades, Modelica has been used especially in the design of multi-domain engineering systems such as mechatronic, automotive and aerospace applications involving mechanical, electrical, hydraulic and control subsystems. The use of Modelica has only recently been extended to the building energy research community, because of the increasing need for analysis of more complex and efficient systems.

An important distinction between Modelica and other building simulation programs (DOE-2, TRNSYS, EnergyPlus, ESP-r) is that Modelica is an open language, and not a dedicated computer program [55]. Equations in this language are encapsulated into models, which can be graphically assembled through connectors in order to define the architecture of larger and more complex models, as illustrated in Fig. 2-2. To assembly models and perform simulations, a simulation environment (e.g. Dymola [56] and OpenModelica [57]) is needed.

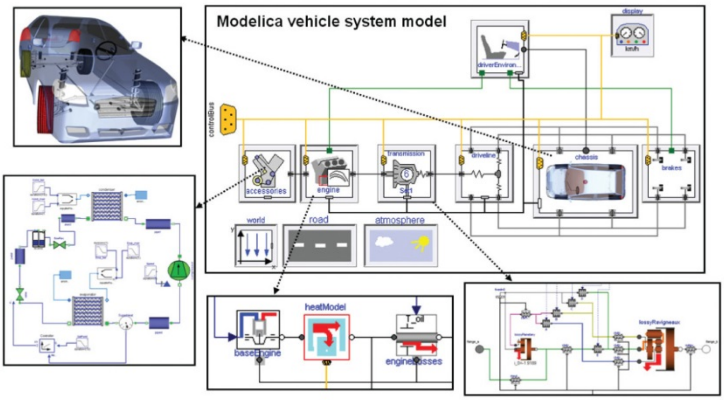


Figure 2-2 System architecture of a Modelica vehicle model

The Modelica code behind a graphical model is automatically converted into executable code. Therefore, a separation exists between the code defining the physical equations and the executable code. This separation makes it easier to implement new component and system models than in traditional tools.

Modelica models are typically structured into libraries. There are numerous libraries available, both commercial and free, ranging from thermodynamics and chemical processes to automotive and space applications. Currently, several

Modelica libraries exist for building components and HVAC systems, and these are continuously being upgraded [58]–[61]. Moreover, the International Energy Agency (IEA) has undertaken a large-scale international project (IEA ECB Annex 60 [62]) with the aim to develop a new generation of computational tools for building energy systems based on Modelica.

The Modelica *Buildings* library [58], developed by the Lawrence Berkeley National Laboratory (LBNL), was used in this work in the Dymola simulation environment.

2.1.3. MODELICA BUILDINGS LIBRARY

The *Buildings* library contains dynamic and steady-state models for building energy and control systems. In particular, it includes models of water-based systems, building thermal zones, controls, heat transfer among rooms and multi-zone airflow. A detailed description of all models can be found online [63]. As stated in Wetter et al. [58], some of the main features of the library are:

- *Support for rapid prototyping*: Users can rapidly add new component and system models by modifying existing models or extending basic.
- *Modeling of arbitrary HVAC system topologies*: Users can model HVAC systems with non-conventional piping or ducting layout, and user-defined control algorithms.

The models in the *Buildings* library are organized in the following main packages:

Airflow: This package provides models to compute the air flow between different rooms and between a room and the exterior environment. The physics of the models in this package is described in [64].
Boundary Conditions: This package contains models to read TMY3 weather data and to compute boundary conditions, such as solar irradiation and sky temperatures.
Controls: This package contains blocks that model continuous time and discrete time controllers (e.g. PID controllers).
Electrical: This package contains models for both DC and AC electrical systems.
Fluid: This is the largest package of the library. It contains component models for air-based and water-based HVAC systems. The package includes models such as chillers, boilers, cooling towers, heat exchangers, solar collectors, valves, mass flow boundary conditions, pressure boundary conditions, pumps, fans, sensors and energy storage. Simple component models are typically based on first principles, whereas models of more complex equipment typically use steady-state performance curves.
HeatTransfer: This package contains models for steady-state and dynamic heat transfer through building constructions such as walls and windows.
Media: This package contains implementations of various media such as models for water and moist air.

Rooms: This package contains models for heat transfer in rooms and through the building envelope. Air exchange between rooms can be computed with models from the package *Airflow*. The room models can also be linked to models of HVAC systems that are composed of the components in the package *Fluid*. A detailed model description can be found in [65].
Utilities: This package contains utility models for thermal comfort calculations, input/output, co-simulation, psychrometric calculations and various functions that are used throughout the library.

2.2. DEVELOPMENT OF THE MODEL WITH MODELICA

Fig. 2-3 shows the graphical layout of the two-pipe system model developed with Dymola. With the aim of providing a readable picture and highlighting all the components needed to form the system, Fig. 2-3 illustrates the two-pipe system when connected to only two rooms. The layout of the system when connected to several rooms/zones would be structurally identical.

Most of the components needed to build the model were already included in the various packages of the *Buildings* library. Others were missing and had to be created. The component models already included in the *Buildings* library are briefly described below, while the component models created within this thesis are explained in separated sections later on. In particular, the active beam model is described in 2.2.1, the control system in 2.2.2 and the Air Handling Unit (AHU) in 2.2.3.

Thermal zone. This is a model of a room with completely mixed air. The room models the following physical processes:

- Transient heat conduction through opaque surfaces.
- Heat transfer through glazing systems, taking into account solar radiation, infrared radiation, heat conduction and heat convection.
- Convective heat transfer between the outside air and outside-facing surfaces.
- Solar and infrared heat transfer between the room enclosing surfaces and convective heat transfer between the room enclosing surfaces and the room air.
- Temperature, pressure and species balance inside the room volume.

Constructions. These models define the thermal properties of the building elements used in the thermal zone model such as walls, roof, floor and windows.

Internal heat gains. These blocks define the schedule for the heat gains injected to the thermal zone model by occupants, lighting and equipment. The model allows distinguishing between convective, radiant and latent heat.

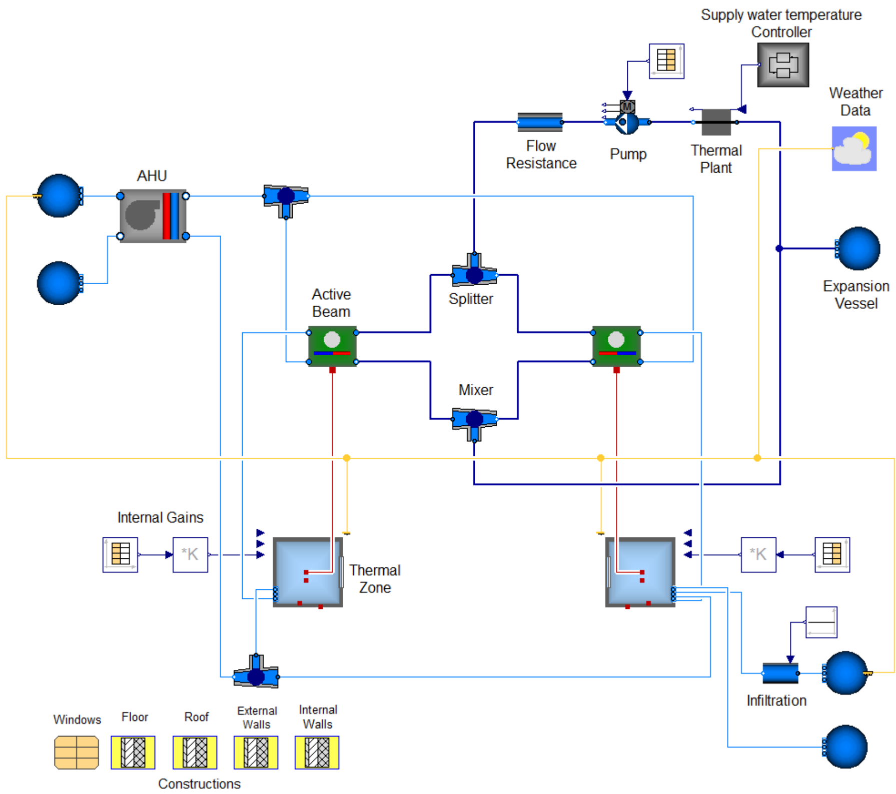


Figure 2-3 Modelica model of the two-pipe system. Light-blue lines represent air streams, dark-blue lines represent water streams, red lines represent convective heat exchange and yellow lines represent weather data.

Weather data. The weather data format is the Typical Meteorological Year (TMY3) as obtained from the EnergyPlus website at https://energyplus.net/weather. These data, which are in the EnergyPlus format, need to be converted into Modelica format through a script included in the library.

Infiltration. This model supplies a prescribed outdoor air mass flow rate to the thermal zone.

Pump. This model represents a water pump. The input of this model is the desired mass flow rate. The pressure rise is computed from the flow resistance of the piping network. A detailed description of the model is provided in [66].

Thermal Plant. This is a model for an ideal heater or cooler with a prescribed outlet temperature. The model forces the outlet temperature to be equal to the temperature of the input signal. This means that the heat delivered to the fluid is computed with the formula:

$$\dot{Q} = \dot{m}_w \, c_{p,w} \, (T_{ret} - T_{sup}) \qquad (2.1)$$

Therefore, when integrated in an HVAC system, this model calculates the useful thermal energy, which can be defined as the energy in the form of direct heat that is used, in place of fuel or electricity, for the production of heating, cooling or other end-use requirements [67]. A more realistic plant for the two-pipe system is described in section 3.5 and Paper V, when discussing primary energy use.

Splitter / Mixer. This model represents a flow splitter or mixer. In particular, the mixer placed at the valley of the two active beam models illustrates the main characteristic of the two-pipe system. Note that this connection is quite easy to model in Dymola, but it is quite cumbersome to model when using traditional BSPs.

Expansion Vessel. This is a model acting as an expansion vessel. When the fluid model in a closed loop models an incompressible flow, such as the water model used, then the density is constant. Consequently, there is no equation that can be used to compute the pressure based on the volume. To avoid this singularity, a model that imposes a pressure reference is needed.

Flow resistance. This model simulates the pressure drop in a piping network. The system curve is based on nominal conditions.

As mentioned in section 2.1.3, one of the main features of the Modelica *Buildings* library is the possibility of adding new components or extending basic ones. This feature was employed to develop three key component models of the two-pipe system model:

- *Active beam model:* Starting from empirical equations, a model that simulates the thermal behavior of the active beam unit was modeled and verified against measurements.
- *Room-temperature water loop controller*: Standard control blocks included in the library were used to develop customized control strategies able to regulate the water loop.
- *Air Handling Unit*: An AHU model was developed by using existing models in the library such as heat exchangers, fans and sensors.

These three models are further described in the next sections.

2.2.1. ACTIVE BEAM MODEL

Generally, component models used in HVAC system simulations are classified as first-principle models or empirical models. First-principle models are derived from fundamental physical laws [38]. However, certain processes associated with typical HVAC components can be too complex to be described entirely by physical laws, or more commonly, the effort to generate a detailed first-principles model is not justified. Therefore, empirical models are frequently used. Empirical models are constructed by using equations derived from experiments, and they usually take the form of a polynomial curve-fit of the component manufacturer's data [38].

In this work, a model of the active beam *Solus*, manufactured by Lindab [27], was developed in Modelica by means of empirical equations. This is a common approach when modeling active beam units in BSPs [16] [68] [69] .

A system of equations describing the heat transfer behavior of active beams suitable for BSPs is given by Livchak and Lowell [16]. The total capacity of an active beam unit is the sum of capacities provided by the primary air and the water.

$$\dot{Q}_{tot} = \dot{Q}_w + \dot{Q}_a \qquad (2.2)$$

The following equation calculated the capacity provided by the primary air

$$\dot{Q}_a = \dot{m}_{ap}\, c_{p,a}\, (T_r - T_a) \qquad (2.3)$$

Heat transfer through the coil is described by the following system of equations under steady-state conditions and assuming no condensation on the coil surface:

$$\dot{Q}_w = \dot{m}_w\, c_{p,w}\, (T_{w,out} - T_{w,in}) \qquad (2.4)$$

$$\dot{Q}_w = k\, A\, (T_r - T_{w,avg}) \qquad (2.5)$$

$$\dot{Q}_a = \dot{m}_{ai}\, c_{p,a}\, (T_{a,in} - T_{a,out}) \qquad (2.6)$$

It is noted that two variables must be defined: k and $\dot{m}_{ai}$. For a given active beam unit, the heat transfer coefficient k is a function of several parameters including primary air and water mass flow rate, and water and air temperature. The empirical equation for k presented in [16] depends on six coefficients that, according to the author, can be derived from manufacturers' capacity tests.

However, common data sheets of active beam products showed that manufacturers do not provide such coefficients. Typically, manufacturers' data sheets only describe the capacity of the coil as a function of primary air flow rate. Correction factors can be found to take into consideration the influence of water flow rate and water temperature difference.

An alternative empirical equation to [16] for the heat transfer coefficient k is provided in this work. The presented equation is based on coefficients derived from the performance data sheet of the *Solus* unit [27]. However, other active beam units can be modeled by adjusting the equation with coefficients directly acquired from their respective performance data sheets. The following equation describes the coefficient *k*:

$$\boldsymbol{k = B\, c_1\, c_2\, \frac{L}{A}} \qquad (2.7)$$

The parameter *B* is a cubic polynomial function of the primary air mass flow rate per active length. Five different polynomials were developed for five different levels of pressure in the primary air plenum. Linear interpolation was assumed between the five polynomials. Table 2-1 shows the polynomials.

Table 2-1 Equations for the parameter B

Pressure	Equation
40 Pa	$(0.0005\, m_{ap,e}{}^3 - 0.0771\, m_{ap,e}{}^2 + 4.0232\, m_{ap,e} + 9.0793) * 1.1329$
60 Pa	$(0.0005\, m_{ap,e}{}^3 - 0.0782\, m_{ap,e}{}^2 + 4.0232\, m_{ap,e} + 12.738) * 1.1329$
80 Pa	$(0.0005\, m_{ap,e}{}^3 - 0.0777\, m_{ap,e}{}^2 + 4.0373\, m_{ap,e} + 15.616) * 1.1329$
100 Pa	$(0{,}0005\, m_{ap,e}{}^3 - 0{,}0767\, m_{ap,e}{}^2 + 3{,}9885\, m_{ap,e} + 18{,}528) * 1.1329$
120 Pa	$(0.0005\, m_{ap,e}{}^3 - 0.0763\, m_{ap,e}{}^2 + 3.9765\, m_{ap,e} + 20.796) * 1.1329$

Modifier c_1 is a cubic polynomial function of the water temperature difference between outlet and inlet. Modifier c_2 is a quartic polynomial function of the water mass flow rate. Table 2-2 displays c_1 e c_2. Length and area of the active beam are represented respectively by L and A.

Table 2-2 Equations for the modifiers c_1 and c_2

Coefficient	Equation
c_1	$-0.0014\, \Delta T_w{}^3 + 0.0154\, \Delta T_w{}^2 - 0.0248\, \Delta T_w + 0.9428$
c_2	$113612\, m_w{}^4 - 15884\, m_w{}^3 + 540.08\, m_w{}^2 + 9.5737\, m_w + 0.4903$

The induced air $\dot{m}_{ai}$ is expressed by the following equation as a function of pressure in the plenum and primary air mass flow rate per coil length.

$$\boldsymbol{\dot{m}_{ai} = 1.05\,(0.047\,p + 20.043)\dot{m}_{ap,e}{}^{-0.635}\dot{m}_{ap}} \qquad (2.8)$$

The active beam model was built by using base elements of the Modelica Standard Library [54] and the Modelica Buildings library. Fig. 2-4 shows the graphic layout of the model.

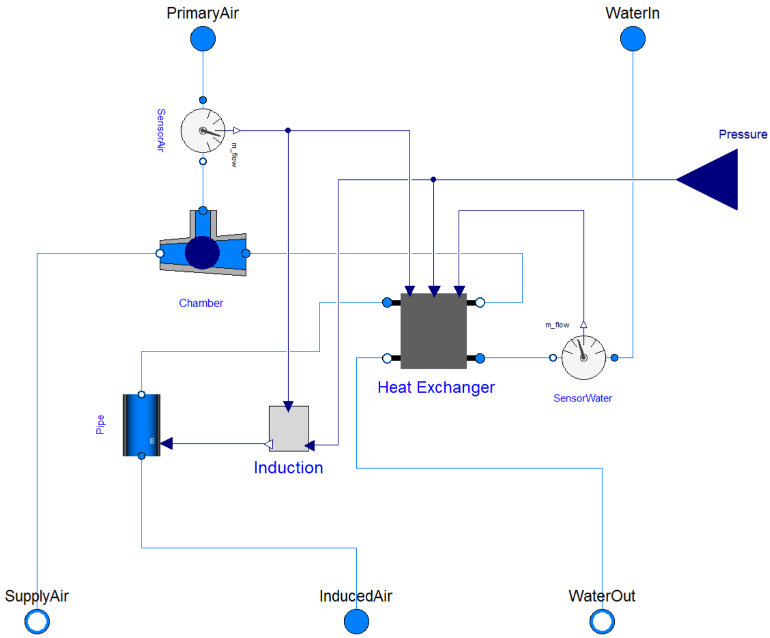


Figure 2-4 Modelica model of the Solus active beam

To verify the thermal behavior of the Modelica active beam model, an experimental data collection was conducted at Lindab ICS laboratories in Farum. Various instruments were used in order to capture experimental data. Temperatures were measured with temperature resistance sensors connected to a data logger. The accuracy was 0.2 °C. Mass flow rates were determined by pressure transmitters with an accuracy of 2%. Pressure was measured by differential pressure transducers. The accuracy was between 0.5% and 4% depending on the level of pressure.

The model was evaluated by comparing predicted and experimental outlet water temperature for four case studies. Fig. 2-5 shows the predicted vs. experimental outlet water temperature for one of the case studies analyzed. Generally, the simulation results show that the model corresponds closely with the actual operation. More details can be found in Paper I.

In this work, an enhanced version of this active beam model has been developed in collaboration with the Simulation Research Group at the Lawrence Berkeley National Laboratory (LBNL). This enhanced model was included in the latest release of the *Buildings* library (v4.0.0) [63] and in the first release of the *Annex 60* library [70], which is one of the main outcomes of the IEA EBC Annex 60 project. Further information can be found in Appendix A.

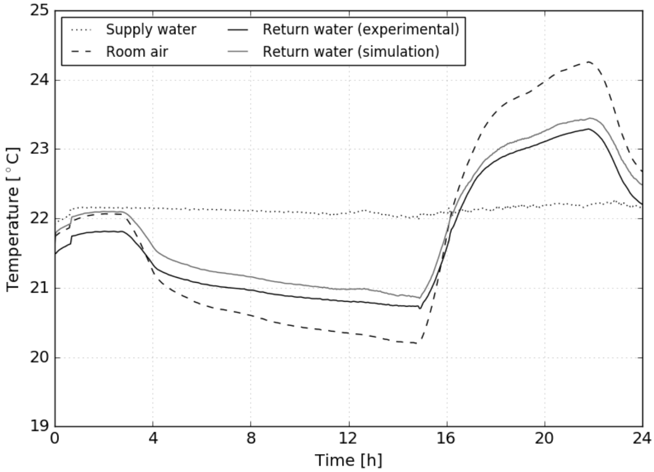


Figure 2-5 Outlet water temperature: comparison between simulated and experimental data

2.2.2. ROOM-TEMPERATURE WATER LOOP CONTROLLER

The capacity of HVAC systems is typically designed for extreme conditions. However, due to the continuous variation of variables such as ambient temperature, solar gains, occupancy etc., most of the operation scenarios require smaller capacities. Therefore, without an appropriate control system, the HVAC system would almost always operate at a greater capacity than the actual loads. Consequently, situations of overheating or overcooling might occur, leading to elevated energy use and probably discomfort in spaces.

There are two main configurations for control systems: Open-loop and closed-loop [71]. An open-loop control does not have a direct feedback link between the value of the controlled variable and the controller. In HVAC systems, generally, the controlled variable is the room air temperature. An example of open-loop control consists of an outdoor thermostat arranged to control heat to a building in proportion to the changes in outdoor air temperature. In this case, the designer presumes a fixed relationship between outdoor air temperature and the heating demand in the building. The actual room air temperature has no effect on the controller. Open-loop systems have the advantage of being relatively simple and consequently cheap with generally good reliability. However, they are often inaccurate since there is no correction for errors in the controlled variable which might result from disturbances. The elements of an open-loop control system are shown in Fig. 2-6.

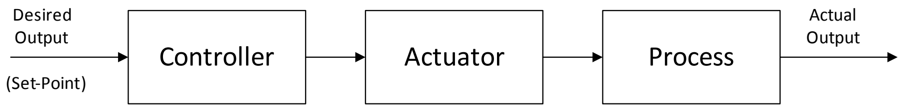


Figure 2-6 Open-loop control system

In a closed-loop or feedback control a signal indicating the actual value of the controlled variable is fed back to the input where it is compared with a reference value. The difference between the actual and reference value activates the controller.

Therefore, the closed-loop aims to keep a variable at a desired value (set-point) despite potential disturbances. Closed-loop systems have the advantage of being relatively accurate in matching the actual values to the desired values. However, they are more complex and costly. The elements of a closed-loop control system are illustrated in Fig. 2-7.

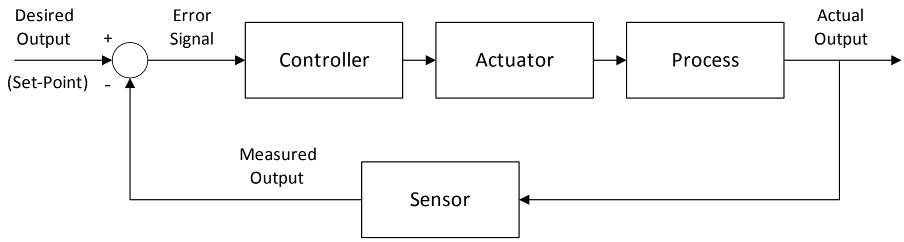


Figure 2-7 Closed-loop control system

In traditional four-pipe active beam systems, a closed-loop control system is usually actuated at zone (or room) level. This means that the air temperature of each room can be controlled individually. In most cases the water flow has a constant supply temperature (45 °C for heating and 14 °C for cooling), and according to the error between actual room air temperature and its set-point, the mass flow rate delivered to the active beam is adjusted through a control valve.

Conversely, the design of the two-pipe system design does not allow for individual control of rooms. Control valves are not contemplated and the supply water temperature is the same for all the zones in the building.

Two main control systems have been developed and analyzed in this thesis in order to regulate the water circuit of the two-pipe system: one open-loop control system and one closed-loop control system. These are described below.

Open-loop controller for the two-pipe system

In the open-loop controller, a constant water mass flow rate is circulated, while the supply water temperature is adjusted based on the outdoor air temperature. The relationship between the outdoor air temperature and the supply water temperature is illustrated in Fig. 2-8.

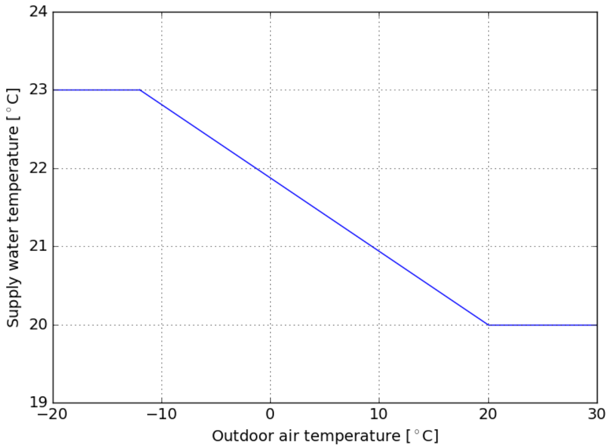


Figure 2-8 Supply water temperature as a function of outdoor air temperature

At extreme cold temperatures, a maximum supply water temperature of 23 °C is set. As shown in section 1.3.1, in the cold season, interior zones of office buildings require cooling while perimeter zone require heating. Since the inherent characteristic of the two-pipe system is to have the same supply water temperature in all the zones, this must be within the range of 20-25 °C, which is the typical room air temperature in buildings. By using a maximum supply water temperature of 23 °C, a perimeter room with 20 °C would be heated while an interior zone with 25 °C would be cooled.

At extreme warm conditions, a minimum supply water temperature of 20 °C was set. As shown in section 1.3.1, in the summer, both interior and perimeter zones generally require cooling. Since there is no direct control on each room, a minimum temperature of 20 °C ensures that the air temperature in every room will be at least at around 20 °C, even if the room becomes unoccupied. A lower supply water temperature would lead to overcooling of unoccupied rooms.

Open-loop control systems are particularly effective in low-exergy systems, where the heating and cooling response is largely self-regulating. As mentioned in section 1.1, low-exergy systems present a small temperature difference between water and room air temperature. Therefore, the rate of heat transfer is very sensitive to changes in room temperature (see Paper V) [72].

Closed-loop controller for the two-pipe system

In the closed-loop controller, a constant water mass flow rate is circulated in the loop, while the supply water temperature is adjusted based on the actual room air temperatures. The supply water temperature can be expressed by the following equation:

$$\boldsymbol{T_{sup} = T_{ret} + k_{hea} - k_{coo}} \qquad (2.9)$$

Where T_{ret} is the return water temperature and k_{hea} and k_{coo} are offsets able to adjust the return water temperature based on current air temperatures in the rooms and set-point temperatures. Fig. 2-9 shows the Modelica model of the controller for a generic five-zone building.

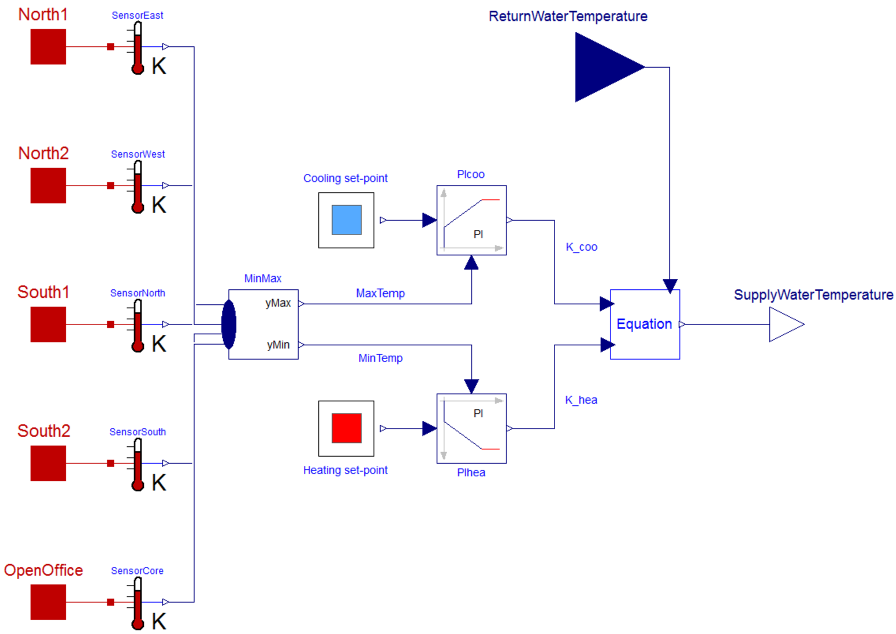


Figure 2-9 Modelica model of the closed-loop controller for the regulation of the supply water temperature in the water loop

The controller is fed with the signals of actual air temperatures in the rooms and return water temperature. The block *MinMax* evaluates the minimum and maximum air temperature among the rooms. The minimum temperature is an input for the block *PIhea*, where it is compared with the heating temperature set-point. If the minimum air temperature is above the set-point, k_{hea} is equal to 0. Otherwise, the PI controller evaluates the value of k_{hea} to be added to T_{ret} to meet the heating set-point. The maximum temperature is an input for the block *PIcoo*, where it is compared with the cooling temperature set-point. If the maximum air temperature is below the set-point, k_{coo} is equal to 0. Otherwise, the PI controller evaluates the value of k_{coo} to be deducted from T_{ret} to meet the cooling set-point. As a consequence, whenever all the zone air temperatures are within the heating and cooling set-point range, k_{hea} and k_{coo} are equal to 0 and, therefore, the supply water temperature is set equal to the return water temperature, thus no energy is required in the thermal plant.

If the minimum room air temperature is below the heating set-point, and the maximum room air temperature is above the cooling set-point, priority is given depending on the season. In winter, priority is given to the cooling set-point. This assures that the maximum room air temperature will never exceed the cooling set-point, while it is allowed for the minimum room air temperature to fall below the heating set-point. In summer, priority is given to the heating set-point. Therefore, the minimum room air temperature will never fall below the heating set-point, while it is allowed for the maximum room air temperature to exceed the cooling set-point. This choice reflects the adaptive thermal comfort model, which is based on the idea that outdoor climate influences indoor comfort. In winter, it is most likely that occupants perceive an indoor climate as more satisfactory if it has air temperature levels below 21 °C rather than above 24 °C.

The output signal leaving the controller is an input for the thermal plant which sets the supply water temperature in the loop. Both the open-loop and the closed-loop controllers were used within this thesis, depending on the simulation experiment.

It should be noted that two additional control strategies were developed in relation to Paper VI. More details can be found in section 3.6.

2.2.3. AIR HANDLING UNIT (AHU)

The ventilation system is a CAV system with constant supply air temperature. Air is delivered to the active beams by an AHU comprising supply and return fans, heating and cooling coils and a heat recovery unit.

Heating and cooling coils are supplied by a hot and a cold water circuit respectively. PI controllers acting on the pumps throttle the water flow rate in the coils. The reference value for the PI controllers is the supply air temperature set-point, while the measured value comes from a temperature sensor placed after the coils. Heating and cooling energy is provided by a heat pump and a chiller respectively.

The heat recovery unit is modeled as a rotary heat exchanger where the speed of rotation of the wheel is adjusted by a PI controller with a reference value equal to the supply air temperature set-point and a measured value coming from a temperature sensor placed after the unit. By varying the speed, the heat exchanger effectiveness is modified according to the actual need.

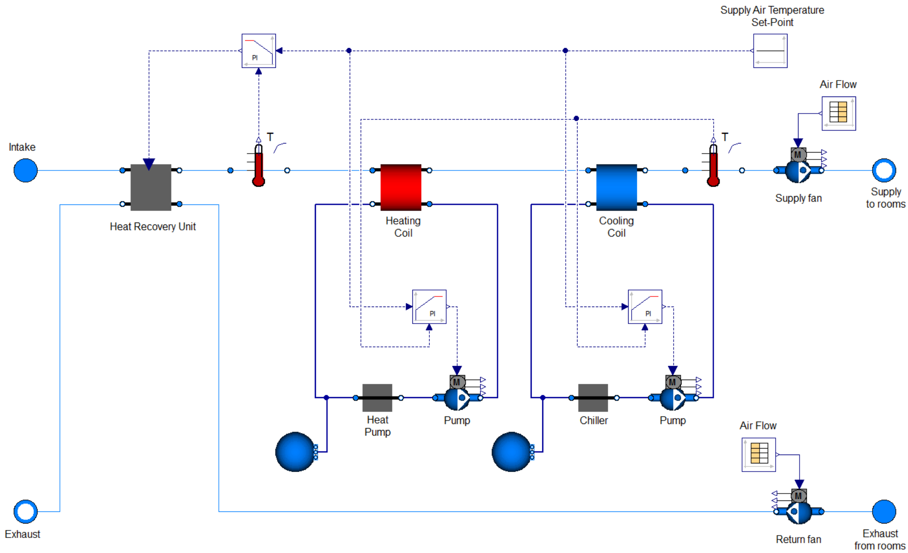


Figure 2-10 Modelica model of the AHU

2.3. SIMULATIONS

2.3.1. SIMULATION INPUT PARAMETERS

In section 2.2, all the components needed to develop the Modelica model of the two-pipe system were described. To analyze the performance of the two-pipe system, the Modelica model was exercised by running a series of simulation experiments under different input conditions. The list of the simulation experiments carried out within this thesis is provided in section 2.3.2.

In order to be simulated, the model had to be defined by setting input parameters such as room dimensions, thermal properties of the building envelope, water and air temperature in the circuit etc. Note that some of these input parameters were kept constant for all the simulation experiments performed while some others were slightly changed. Below, some general input parameters are illustrated. More details are provided for each simulation experiment in chapter 3 and in the corresponding papers.

All the input parameters related to the building model (U-values, infiltration rate, internal loads etc.) were selected according to the medium prototype model as described in the report, U.S. Department of Energy Commercial Reference Building Models of the National Building Stock [73]. The report characterizes 16 prototype buildings for 16 climate zones covering the majority of the US commercial building stock. These building models have been developed to serve as a starting point for energy efficiency research, as they represent fairly realistic buildings and typical construction practices. Since the publication of the report, the Pacific Northwest National Laboratory (PNNL) has made numerous enhancements to the original prototype models that are now compliant with 2004, 2007, 2010 and 2013 editions of AHSRAE standard 90.1 [74].

With regards to ventilation requirements, these were selected according to the EN 15251 standard for category II low-polluting building [75] and were calculated according to

$$\dot{m}_{vent} = 0.0084\ N_{persons} + 0.00084\ A_{zone} \qquad (2.10)$$

where $\dot{m}_{vent}$ is the outside air mass flow rate in [kg/s], $N_{persons}$ is the number of persons in the zone and A_{zone} is the area of the zone in [m^2].

The design values used for dimensioning the system were selected according to the REHVA chilled beam guidebook [15] and manufactures recommendations [27] and these are shown in Table 2-3.

Each zone was thermally connected to one active beam model where the capacities of multiple active beams were lumped together by using an equivalent total beam length. To lump active beams into one model does not affect simulation results since the total capacity of the lumped model is equal to the sum of the capacities of the single active beams. To calculate the total beam length required by each thermal zone (L_{zone}), the following expression was used:

$$L_{zone} = max(L_{hea}, L_{coo}\ , L_{ven}) \qquad (2.11)$$

Where L_{hea} is the beam length needed to meet heating peak loads, L_{coo} is beam length needed to meet cooling peak loads and L_{ven} is beam length needed to provide enough outdoor air. L_{hea} and L_{coo} were calculated by performing simulations, while L_{ven} was simply calculated by using Eq. (2-10).

Table 2-3 Design parameters for sizing the two-pipe system

Design parameters	**Heating**	**Cooling**
Room air temperature	20 °C	24 °C
Primary air temperature	19-22 °C	19-22 °C
Supply water temperature	23 °C	20 °C
Primary air mass flow rate	0.026-0.03 kg/s	0.026-0.03 kg/s
Water mass flow rate	0.038 kg/s	0.038 kg/s
Length	1.8-3 m	1.8-3 m
Total capacity	200-250 W	450-500 W

Both the water and the air circuits operated at constant full flow rate during operating hours. To reduce the energy use of fans and pumps, the total air and water flow rates were multiplied by a factor 0.5 and 0.25, respectively, during non-operating hours. In some simulation experiments, the water circuit was completely turned off during non-operating hours.

When it comes to pressure losses, at the full flow rate, the total pressure drop in the water loop was assumed to be 35 kPa, and the total pressure drop in the ventilation loop was assumed to be 500 Pa. For lower flow rates, e.g. during night operation, these values were reduced, as the simulation model computes flow friction as a function of the flow rate.

The thermal losses of hydronic parts, such as pipes and valves, were assumed to be small in comparison to the total energy transferred to the active beams. Therefore, thermal losses of hydronic parts were not included in the model.

2.3.2. SIMULATION EXPERIMENTS

An overview of the six simulation experiments is provided below:

- Simulation experiment 1 – Test case (Paper II)
- Simulation experiment 2 – Open-loop vs. closed-loop controller (Paper III)
- Simulation experiment 3 – Sensitivity analysis
- Simulation experiment 4 – Parametric analysis (Paper IV)
- Simulation experiment 5 – Primary energy savings (Paper V)
- Simulation experiment 6 – Control strategies for the water loop (Paper VI)

Simulation experiments 1, 2, 4, 5 and 6 resulted in the writing of five scientific papers, while simulation experiment 3 is presented only in this thesis. Results from the six simulation experiments are provided in chapter 3.

As recommended by the *Buildings* library developers [63], all simulations were run with the Radau solver and a tolerance of 1E-6. These settings generally lead to faster and more robust simulations for thermo-fluid flow systems.

CHAPTER 3. RESULTS AND DISCUSSION

In this chapter, the main results from the simulation experiments briefly mentioned in section 2.3.2 are presented and discussed.

The results of the simulation experiments are mainly presented with the aim of illustrating the difference between energy generated and energy demand. The energy demand is defined as the sensible energy added (heating mode) or removed (cooling mode) by the active beam to the space. The energy generated has the same meaning of the useful thermal energy defined in section 2.2. This is defined as the energy in the form of direct heat that is used, in place of fuel or electricity, for the production of heating, cooling or other end-use requirements. The term energy generated is introduced only for convenience along the comparison with energy demand. The difference between energy generated and energy demand is shown in Fig. 3-1.

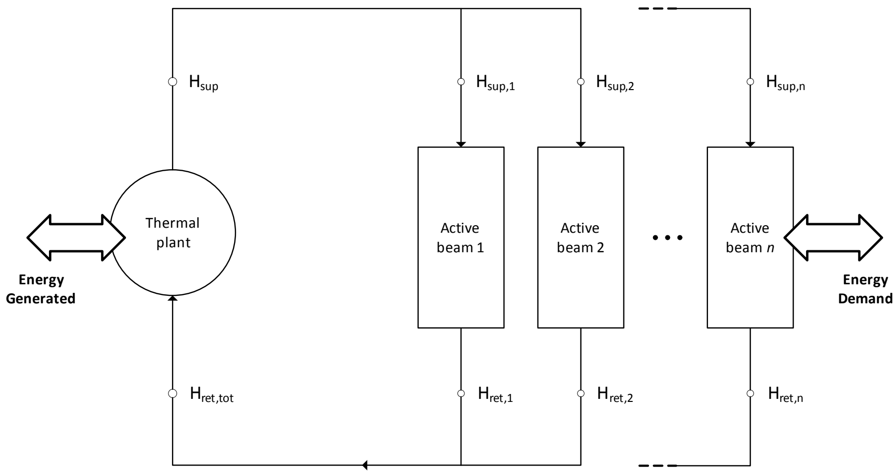


Figure 3-1 Energy generated vs. energy demand

Mathematically, the energy generated can be expressed by:

$$EG_{hea} = \int_{t_1}^{t_2} \left(H_{sup} - H_{ret,tot}\right)dt \qquad for\ H_{sup} > H_{ret,tot} \tag{3.1}$$

$$\boldsymbol{EG_{coo}} = \int_{t_1}^{t_2} \left(\boldsymbol{H_{sup}} - \boldsymbol{H_{ret,tot}}\right)\boldsymbol{dt} \qquad \boldsymbol{for\ H_{sup} < H_{ret,tot}} \tag{3.2}$$

While the energy demand:

$$\boldsymbol{ED_{hea}} = \int_{t_1}^{t_2} \sum_{i=1}^{n} \left(\boldsymbol{H_{sup,i}} - \boldsymbol{H_{ret,i}}\right)\boldsymbol{dt} \qquad \boldsymbol{for\ H_{sup,i} > H_{ret,i}} \tag{3.3}$$

$$\boldsymbol{ED_{coo}} = \int_{t_1}^{t_2} \sum_{i=1}^{n} \left(\boldsymbol{H_{sup,i}} - \boldsymbol{H_{ret,i}}\right)\boldsymbol{dt} \qquad \boldsymbol{for\ H_{sup,i} < H_{ret,i}} \tag{3.4}$$

Where the enthalpy flow rate *H* in correspondence of a generic node *i* is

$$\boldsymbol{H_i} = \boldsymbol{\dot{m}_i\, h_i} = \boldsymbol{\dot{m}_i\, c_{p,w}\, T_i} \tag{3.5}$$

Note that the difference between energy demand and energy generated represents not only the heat transfer among zones through the water loop, but also the energy savings achieved by the two-pipe system when compared with a four-pipe system running with exactly the same operating conditions and reaching the same indoor thermal climate.

In a four-pipe system, neglecting heat losses from pipes, the energy exchanged between the active beam and the space (energy demand) is equal to the energy used by the thermal plant (energy generated). Therefore, it is expected that $EG_{hea} = ED_{hea}$ and $EG_{coo} = ED_{coo}$. In the two-pipe system, due to the piping layout, the energy exchanged between the active beam and the space (energy demand) is equal to or greater than the energy used by the thermal plant (energy generated). Thus, it is expected that $EG_{hea} \leq ED_{hea}$ and $EG_{coo} \leq ED_{coo}$.

Therefore, by running a simulation of the two-pipe system model, it is possible to calculate the energy savings achieved by the two-pipe system in comparison to a fictional four-pipe system running with exactly the same operating conditions and reaching the same indoor thermal climate. The energy use of the two-pipe system would be represented by the energy generated (Eq. 3.1 and 3.2), while the energy use of the four-pipe system would be represented by the energy demand (using Eq. 3.3. and 3.4).

A realistic model of the four-pipe system was developed only for the simulation experiment 5, where the focus was on primary energy use. In this case, parameters such as the COP of the thermal plant were also included in the energy calculation.

With regards to indoor thermal climate, most of the results are presented in terms of room air temperature. Values of relative humidity are shown only for simulation experiment 5.

3.1. SIMULATION EXPERIMENT 1 - TEST CASE

This simulation experiment aimed to study a basic Modelica model and highlight some general principles of the two-pipe system function. A perimeter and an interior office room were considered as the building model. More details about the simulation input parameters can be found in Paper II.

A full-year simulation was run and results are presented for three typical weeks in three seasons: winter, spring and summer. Figs. 3-2a, 3-2c and 3-2e show the weekly temperature profiles of room air temperatures and supply water temperature. Figs. 3-2b, 3-2d and 3-2f show the daily profiles of room air temperature, supply water temperature, total return water temperature, return water temperature from the perimeter zone and return water temperature from the interior zone. The water loop is regulated by the open-loop controller described in section 2.2.2. Generally, comfortable levels of indoor air temperature were achieved in all the three periods.

In winter, during occupied hours, the supply water temperature always lies between the air temperature in the interior room and the air temperature in the perimeter room. This means that the two-pipe system is simultaneously providing heating to the perimeter room and cooling to the interior room. As a result, the return water temperature from the interior zone is higher than the supply water temperature while the return water temperature from the perimeter is lower than the supply water temperature. Therefore, after they mix, the total return water temperature has values quite close to the supply water temperature.

In spring, the supply water temperature lies alternately below and between the two room air temperatures. During occupied hours, when the internal gains are high, the supply water temperature is mostly below both room air temperatures, meaning that cooling is provided to both rooms. However, there are still periods where the supply water temperature lies between the two air temperatures, especially in the morning hours. Note that during occupied hours (daytime) the supply water temperature typically decreases as a consequence of higher outdoor air temperatures (see Fig. 2-8).

In summer, the supply water temperature always lies below both room air temperatures. This is because the sum of internal and external heat gains is very high, and cooling is required at any time during operating hours.

The difference between the energy generated and the energy demand can be illustrated by plotting the weekly profiles of the power generated and the power demand, as shown in Fig. 3-3. Power generated and power demand can be defined similarly to Eq. (3-1 to 3-4) by neglecting the integral functions.

In winter, at the beginning of the day, due to the absence of internal heat gains, the building only needs heating. Therefore, power generated and power demand have the same profile.

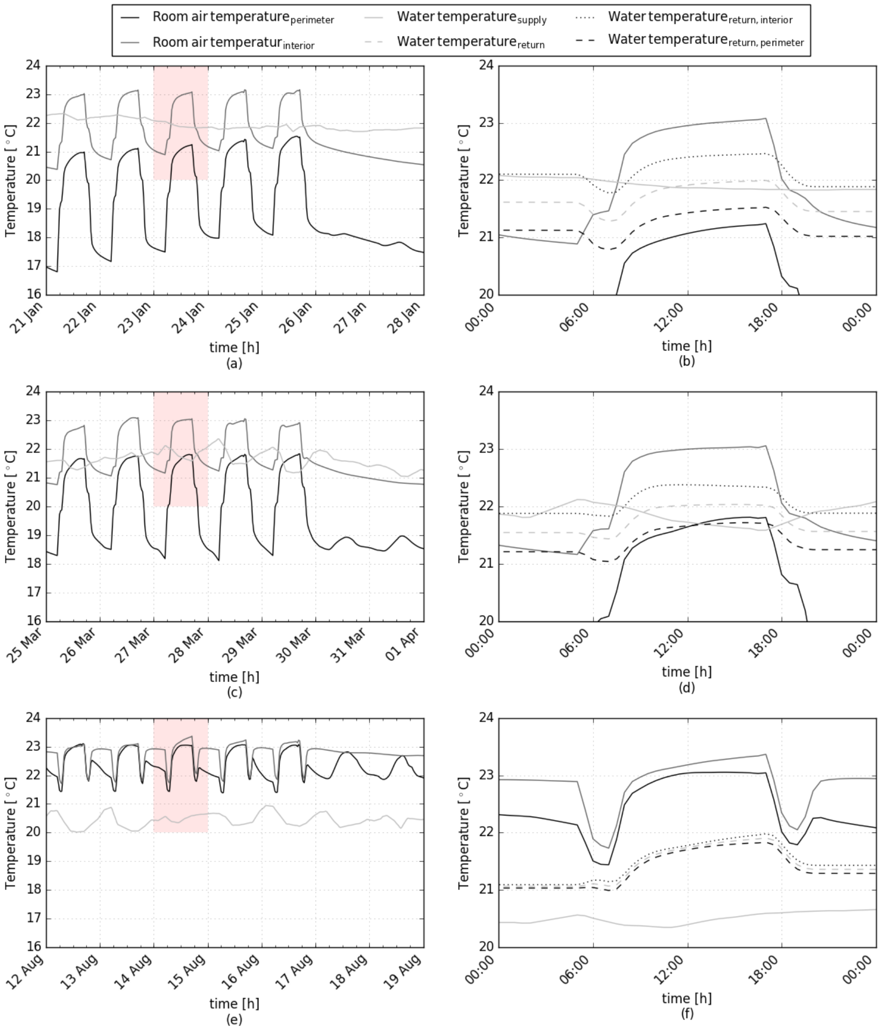


Figure 3-2 Weekly and daily temperature profiles: (a) winter week, (b) winter day, (c) spring week, (d) spring day, (e) summer week, (f) summer day. Note that the figures on the right column are the enlargement of the shaded areas in the figures on the left column.

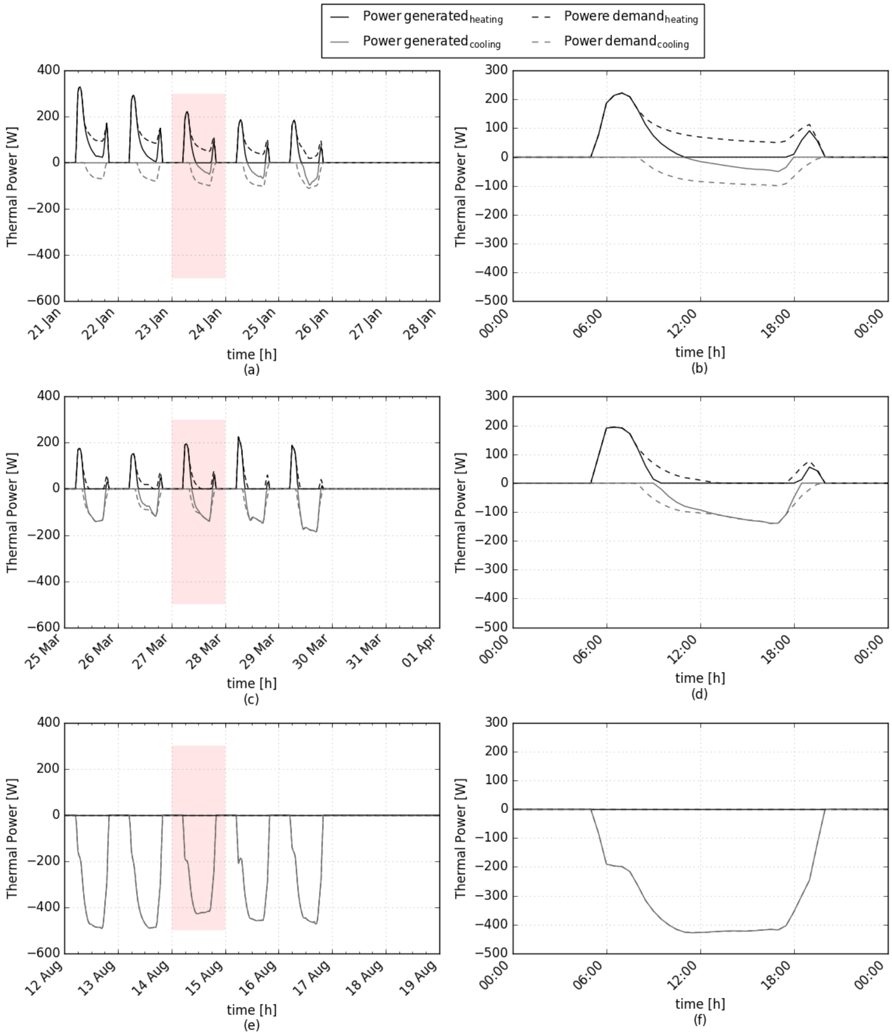


Figure 3-3 Thermal power profiles: generated vs. demand (a) winter week, (b) winter day, (c) spring week, (d) spring day, (e) summer week, (f) summer day. Note that the figures on the right column are the enlargement of the shaded areas in the figures on the left column.

When the internal gains increase, the heating power generated goes to zero and the cooling power generated begins to increase. Note that the heating power demand profile also decreases, but does not reach zero. On the other hand, the cooling power demand starts to rise a couple of hours before the cooling power generated, and assumes higher absolute values.

In spring, this behavior is less accentuated. In the middle of the day, both the heating power generated and the heating power demand reach zero. With regards to cooling, both the power generated and the power demand have the same profile. No difference is noticed between power generated and power demand during the summer week.

The mismatch between the power generated and the power demand results in energy savings due to heat transfer between zones through the water loop. Fig. 3-4 shows the energy generated and the energy demand for the three typical weeks. It can be noticed that the energy generated is lower than the energy demand for the winter and spring week. In particular, the energy generated and the energy demand during the winter week were 6.4 kWh and 9.92 kWh, respectively. Therefore, energy savings of approximately 35% were achieved. Energy savings of about 7% occurred during the spring week. In this case, the energy generated and the energy demand were 8 kWh and 8.64 kWh, respectively. As expected, simulations for the typical summer week show exactly the same value for the energy generated and energy demand.

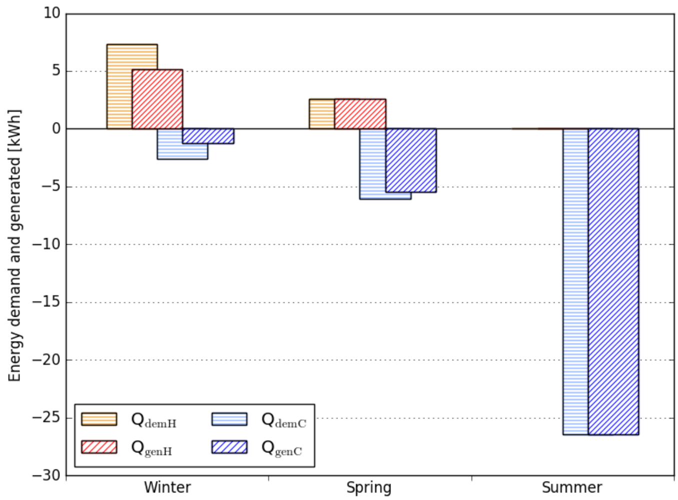


Figure 3-4 Weekly energy generated and energy demand – winter, spring and summer

Fig. 3-5 shows the cumulative energy generated and energy demand of the two-pipe system over the year. In winter, the slope of the curve representing the energy generated is less steep than the curve representing the energy demand. During this period, opportunities to transfer heat among rooms occur. In the summer, the two curves have the same slope, meaning that the energy generated and the energy demand are equal. The annual energy generated was 23.7 kWh/m^2, while the annual energy demand was 25.8 kWh/m^2. Therefore, annual energy savings of approximately 8% were achieved thanks to the innovative hydronic configuration.

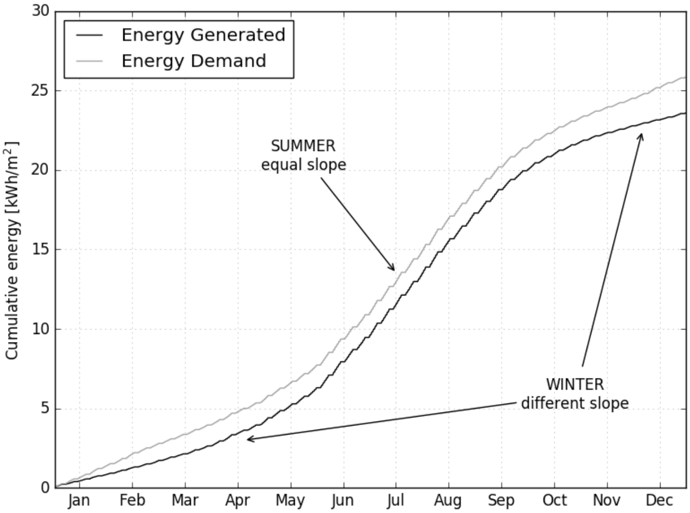


Figure 3-5 Cumulative annual energy generated and demand

3.2. SIMULATION EXPERIMENT 2 - OPEN-LOOP VS. CLOSED-LOOP CONTROLLER

This simulation experiment aimed to present the development of the closed-loop controller described in section 2.2.2. To evaluate the benefits provided by the closed-loop controller, a comparison was made with respect to the open-loop controller.

As in the previous study, the building model was represented by a perimeter room and an interior room. The simulation input parameters used for the thermal properties of the envelope, infiltration rate, internal heat gains and HVAC system are described in Paper III. Two full-year simulations were run, one for each control strategy. Results are presented for a typical winter and summer day. Since the indoor air temperatures obtained when using the open-loop controller are not the same as the ones obtained when using the closed-loop controller, the energy use for ventilation was also computed.

Fig. 3-6 shows the daily temperature profiles of the perimeter and the interior room. In winter, as already shown in section 3.1, the open-loop controller presents supply water temperature almost always above the return water temperature. Therefore, heating energy is required by the thermal plant for most of the hours. In the middle of the day, a small amount of cooling is provided. Conversely, the closed-loop controller is able to set the supply water temperature equal to the return water temperature for all the operating hours, expect for a few hours at the beginning of the day. This means, that little energy is required by the thermal plant. Generally, the air temperatures obtained with the open-loop controller are slightly higher than the ones obtained with the closed-loop controller.

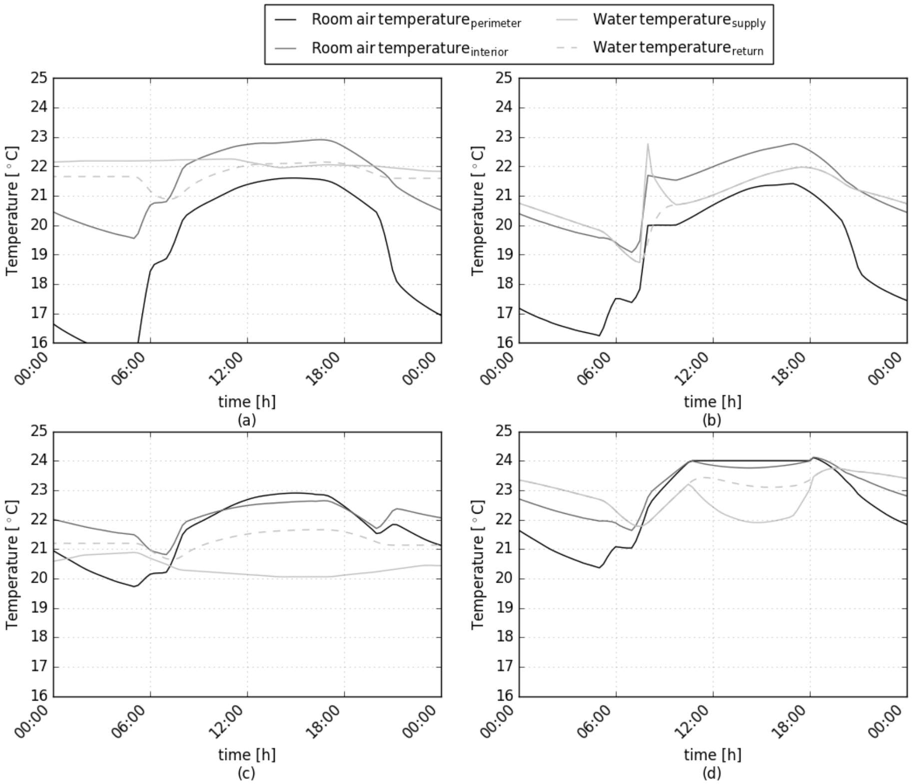


Figure 3-6 Temperature profiles, (a) winter day open-loop, (b) winter day closed-loop, (c) summer day open-loop, (d), summer day closed-loop

In the summer day, the open-loop controller has the attitude to set the supply water temperature at a value close to the minimum of 20 °C. This is because the open-loop controller sets the supply water temperature as a function of the outdoor air temperature. There is no track of the actual thermal needs of the rooms. As a result, the open-loop controller is over-cooling the rooms and wasting energy. Conversely, the closed-loop controller sets the supply water temperature by adjusting the return water temperature just enough to meet the set-point cooling temperature.

Figure 3-7 shows the energy use of the two-pipe system for both controllers for the typical winter day and summer day. When the system integrated the open-loop controller, the energy use was 2.5 kWh and 6.3 kWh, respectively, for the winter and summer day. When the system integrated the closed-loop controller, the energy use was 1.4 kWh and 3.5 kWh, respectively, for the winter and the summer day. This means that the integration of the closed-loop controller lead to energy savings of approximately 44% for both the winter and summer day.

When considering the entire year, the two-pipe system used 38 kWh/m^2 and 18 kWh/m^2, respectively, for the open-loop and the closed-loop control system.

As a result, energy savings of about 46% occurred. In conclusion, with the development of the closed-loop controller, it was possible to increase the energy performance of the two-pipe system.

Note that the energy use for the AHU heating coil is higher for the closed-loop controller. This is because the indoor air temperatures are usually lower, and therefore less heat can be transferred in the heat recovery unit. It is worth mentioning that the energy use for pumps and fans is intended to be as useful mechanical energy (similar to useful thermal energy), and can be expressed by:

$$\dot{Q}_{mech} = \dot{m}\,\Delta p \tag{3.6}$$

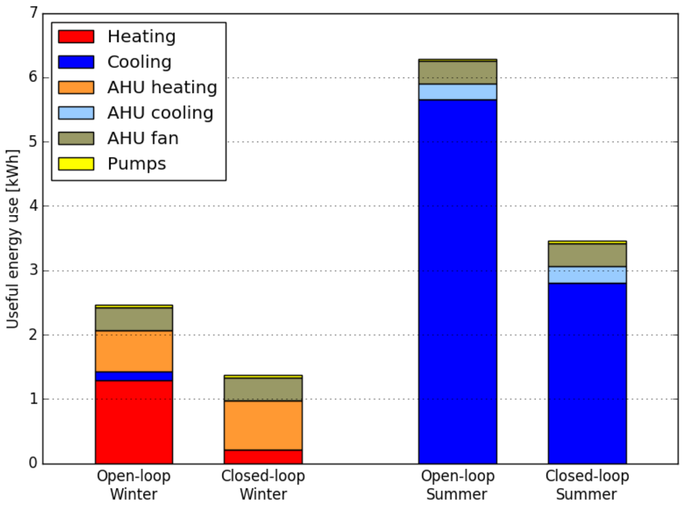


Figure 3-7 Daily energy use for the open-loop and closed-loop controller

3.3. SIMULATION EXPERIMENT 3 - SENSITIVITY ANALYSIS

The goal of this simulation experiment was to reveal the factors affecting the energy savings (in terms of difference between energy generated and energy demand) of the two-pipe system. A sensitivity analysis was carried out by making successive alterations to a base model and executing simulations after each step. The factors were selected based on their potential influence on increasing or decreasing the simultaneous need of heating and cooling in the base model.

In Dymola, the time required to perform a simulation grows almost cubically with the number of states [76], which is strongly influenced by the number of thermal zones used in the model. Therefore, in order keep a reasonable computational time, but still capture the behavior of the system in detail, three thermal zones were considered.

These three thermal zones represent a portion of a typical middle floor of a multi-story office building. Fig. 3-8 shows the layout of the floor plan. This portion consists of two perimeter office rooms and one interior office space. If the length is much larger than the width, the specific energy use of this portion of floor can accurately predict the specific energy use of the entire floor.

The left and the right internal walls, the ceiling and the floor are considered as adiabatic surfaces. The internal walls between the perimeter rooms and the open space are thermally connected. Also, a door with height of 2.1 m and width of 0.9 m is placed on both these internal walls. The door is assumed to be always half-open. The model of the door allowed a value between 0 and 1 to represent the degree of opening. Therefore, a value of 0.5 was chosen for the base model.

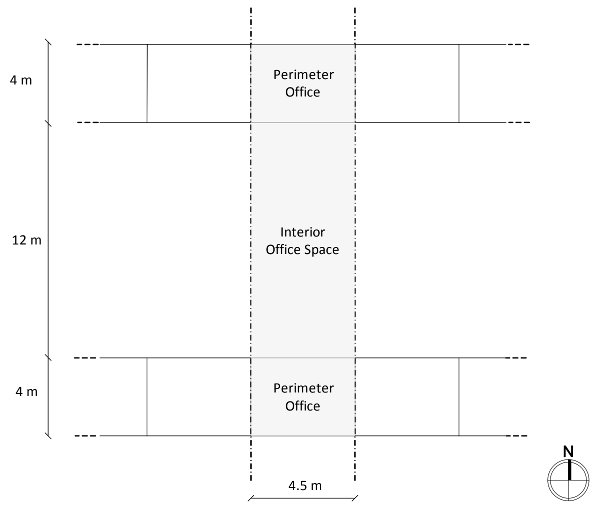


Figure 3-8 Typical floor layout in a multi-story office building

Base model and alterations

The base model represents the starting point for the sensitivity analysis. Table 3-1 shows the simulation input parameters used. The annual heating energy generated and demand was 0.4 kWh/m^2 and 1.9 kWh/m^2, respectively. The annual cooling energy generated and demand was 14.2 kWh/m^2 and 15.7 kWh/m^2, respectively. Therefore, in the base model, energy savings of approximately 17% occurred thanks to heat transfer between building zones through the water loop.

Table 3-1 Simulation input parameters for the base model

	Parameter	Value
Building		
	Climate	London
	Number of zones	3
	Total floor area	90 m^2
	U-value external wall	0.35 W/m^{2}K
	U-value window	2.37 W/m^{2}K
	Infiltration	0.08 ACH
	Internal gains	25 W/m^2
	Occupancy hours	8-18 h
	Door opening	0.5
HVAC system		
	Supply water temperature	23-20 °C
	Water mass flow rate per beam	0.038 kg/s
	Supply air temperature	19 °C
	Air mass flow rate per beam	0.026 kg/s
	Operating hours	6-22 h
	Set-point temperatures	21 °C / 24 °C
Control		
	Room-temperature loop	Closed-loop
	Ventilation	CAV ideal

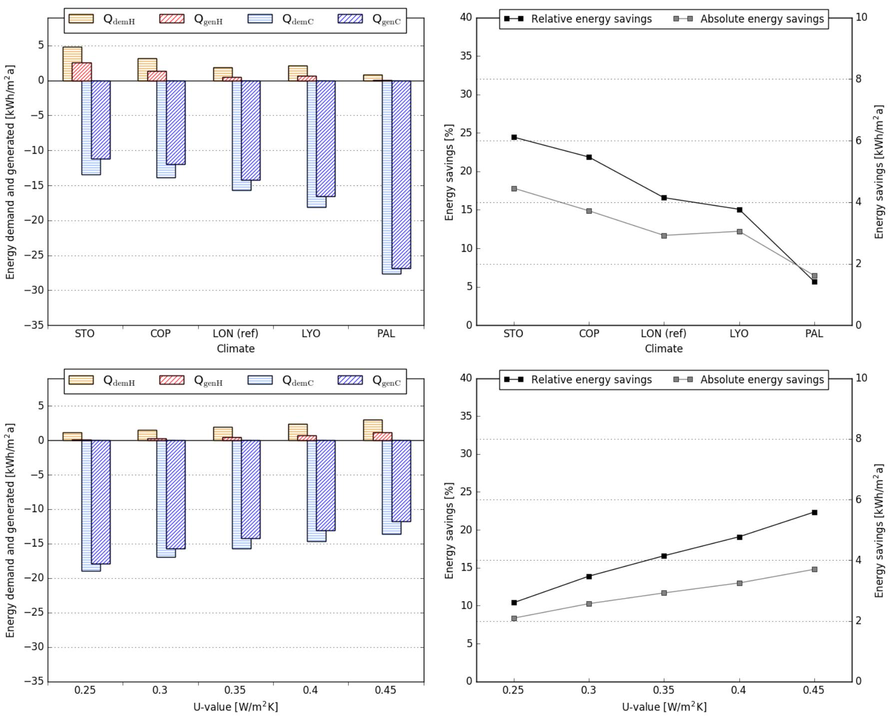
Changes to the base model were made to explore factors affecting the energy saving potential of the two-pipe system. In table 3-2 a complete list of the factors used for the sensitivity analysis is displayed. The analysis included 8 factors and 2 to 4 alterations for each. Beside the base case, the total number of simulation cases was 25.

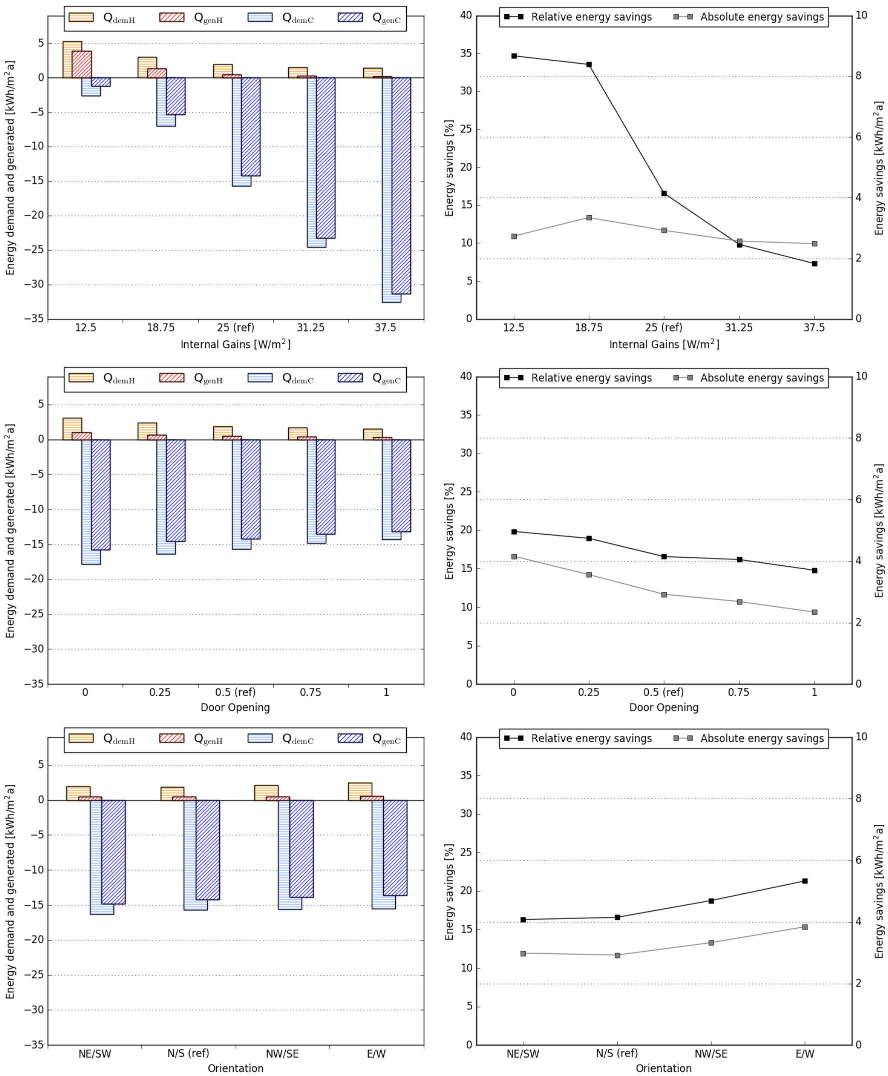
Table 3-2 Alterations to the base model. Thermal mass is represented by the thickness of the concrete slabs, while interior ratio is represented by the area of the interior zone divided by the area of the perimeter zones.

	Ultra-low	Low	Reference	High	Ultra-high
Climate	Stockholm	Copenhagen	London	Lyon	Palermo
U-value [W/m^{2}K]	0.25	0.3	0.35	0.4	0.45
Heat Gains [W/m^{2}K]	12.5	18.75	25	31.25	37.5
Door opening [-]	0	0.25	0.5	0.75	1
Orientation	-	NE/SW	N/S	NW/SE	W/E
Thermal Mass [m]	-	0.1	0.2	0.3	-
Interior ratio [-]	-	0.5	1.5	2.5	
Location	-	Floor	Middle	Roof	-

Results

Fig. 3.9 illustrates the results of the sensitivity analysis. Figures on the left show the energy generated and the energy demand while figures on the right show the relative and the absolute energy savings achieved.




Q_{demH}
Q_{genH}
Q_{demC}
Q_{genC}
Energy demand and generated [kWh/m^{2}a]
Internal Gains [W/m^2]
12.5
18.75
25 (ref)
31.25
37.5
Relative energy savings
Absolute energy savings
Energy savings [%]
Energy savings [kWh/m^{2}a]
Door Opening
0
0.25
0.5 (ref)
0.75
1
Orientation
NE/SW
N/S (ref)
NW/SE
E/W

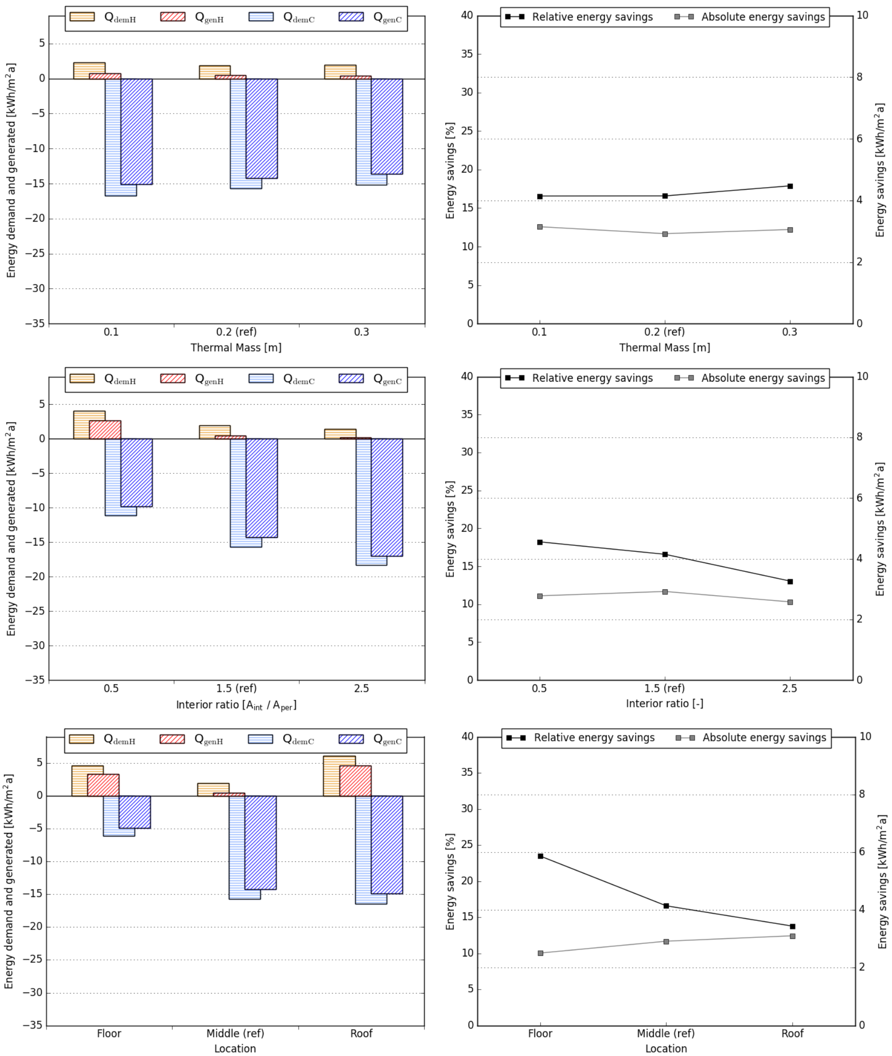


Figure 3-9 Energy demand vs. energy generated (left) and relative vs. absolute energy savings (right)

Generally, the maximum and minimum values of absolute energy savings were both achieved by altering the climate. The maximum was 4.2 kWh/m^2 and occurred for the Stockholm climate, while the minimum was 1.8 kWh/m^2 and occurred for the Palermo climate. The maximum value of relative energy savings was 34.8% and was obtained with internal gains of 12 W/m^2. The minimum value of relative energy savings was 6.1% and occurred for the Palermo climate.

With regards to the geography, cold climates present higher opportunities for transferring heat among zones. By having longer periods with low outdoor temperatures, simultaneous need of heating and cooling occurs more frequently than in warm climates.

As expected, by increasing the insulation level of the building envelope, the need for heating decreases, while the need for cooling increases. In addition, the balance point temperature of perimeter zones decreases, limiting the values of outdoor temperatures suitable to take advantage of simultaneous heating and cooling demand.

A similar concept applies to the internal heat gains. High heat gains lead to high cooling demand and low balance point temperatures. As a result, all the zones in the building mainly require cooling, limiting opportunities for simultaneous need of heating and cooling. Note that the high relative energy savings for low internal gains are mainly due to the low total annual energy use. Absolute energy savings have similar values.

Inter-zone air flow is another factor affecting the energy saving potential of the two-pipe system. The need for simultaneous heating and cooling of thermal zones also depends on the air exchange between zones. In the limiting case of no air exchange (closed doors), the differences in room temperature can be large, leading to higher opportunities for transferring heat through the room-temperature water loop. Conversely, if there is very high air flow between zones, such as through open doors, all rooms would be at a similar temperature, limiting the heat transfer potential of the water loop.

When it comes to the orientation, the lowest relative energy savings are achieved for the orientation SW/NE, while the highest relative energy savings are achieved for the orientation W/E.

By increasing the thermal mass of the building, the heating and cooling energy use was slightly reduced. The values of absolute energy savings are similar. Therefore, the higher relative energy savings for the heavy building are mainly due to the low total annual energy use.

The interior ratio is defined as the area of the interior zone divided by the area of the perimeter zones. Higher values of the interior ratio lead to situations where cooling demand largely overcome heating demand in the building. Also in this case, the absolute energy savings present similar values; therefore the higher relative energy savings achieved for low interior ratios are obtained thanks to a general reduction in energy use.

The base model was assumed to be located in a middle floor of a multi-story office building. If the floor was located at the top-floor (roof), higher heating and cooling energy use is seen. This is because a larger surface is exposed to the outdoor environment. If the floor was located at the bottom-floor (ground level), higher heating energy use is seen due to the heat losses through the ground. However, these heat losses also have the beneficial effect to reduce the cooling energy use.

The influence of each factor in relation to the others can be highlighted by introducing the influence coefficient [77]:

$$\boldsymbol{IC} = \frac{\boldsymbol{change\ in\ output}}{\boldsymbol{change\ in\ input}} = \frac{\Delta \boldsymbol{OP}/\boldsymbol{OP_{BC}}}{\Delta \boldsymbol{IP}/\boldsymbol{IP_{BC}}} \tag{3.7}$$

Where *ΔOP* is the change of the output, *ΔIP* is the change of the input, and subscript *BC* represents the base case. Using Eq. (3.7), the average influence coefficient of each parameter was calculated as shown in Table 3-3. A larger absolute value of the influence coefficient represents a more sensitive relation between the relative energy savings achieved and the parameter. Note that the influence coefficient can only be determined if the input perturbations are quantifiable. Therefore, the climate was expressed in terms of annual average outdoor air temperature, while the location was ignored in this analysis as it is not possible to directly express this parameter in a quantifiable form.

Table 3-3 shows that the most influencing parameter is the heat gains with an influence coefficient of -1.913. The least influencing parameter is the thermal mass with an influence coefficient of 0.072.

The plus and minus signs of the influencing coefficient in Table 3-3 express the direction of the change of the output. The plus sign means that an increment of the influencing parameter leads to an increment of the relative energy savings. The minus sign means that an increment of the influencing parameter leads to a decrement of the relative energy savings.

Table 3-3 Influence coefficient of the studied parameters

Studied parameter	**Influence coefficient**
Climate	-0.953
U-value	1.219
Heat Gains	-1.913
Door opening	-0.152
Orientation	0.402
Thermal Mass	0.072
Interior ratio	0.230

3.4. SIMULATION EXPERIMENT 4 - PARAMETRIC ANALYSIS

A parametric (or factorial) analysis involves choosing a given number of samples for each parameter and running simulations for all combinations of the samples [78]. Such a technique can reveal aspects not directly noticeable when performing a sensitivity analysis. Since it is not practical to conduct a full parametric analysis of all the parameters shown in Table 3-2 (this would require 67500 simulation runs), a limited number of parameters were chosen. In particular, three climate locations and five occupancy levels were selected.

The simulation conditions of this experiment were almost identical to the ones used in section 3.3. However, a significant change was made in respect to the modeling of the occupancy. Several studies showed that office rooms are occupied for 50-60% of the working hours. To model this situation, and therefore reproduce situations where an office room is occupied while others may be vacant, a simple probabilistic model was developed in Modelica.

The main contribution provided by this parametric analysis is illustrated in Fig 3-10. The sensitivity analysis previously described showed that the relative energy savings in warm climates are generally lower than in cold climates. However, Fig. 3-10 shows that in correspondence of low occupancy levels, the relative energy savings achieved in warm climates are higher than the energy savings achieved in cold climates. More details can be found in Paper IV.

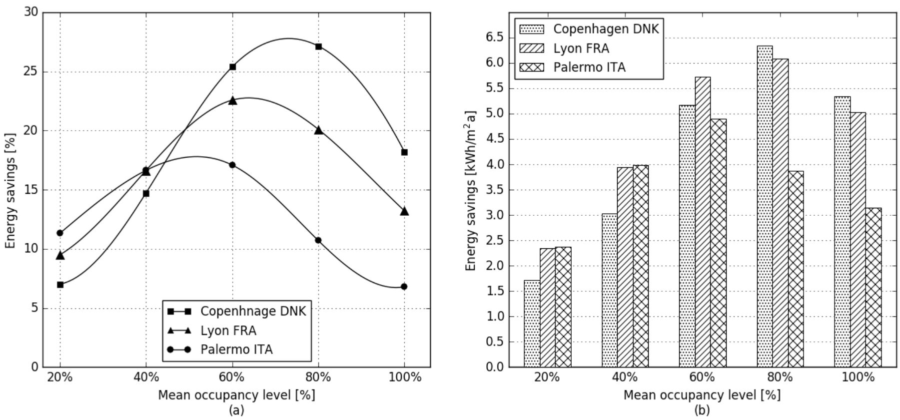


Figure 3-10 Relative and absolute energy savings for the fifteen simulation cases considered.

3.5. SIMULATION EXPERIMENT 5 - PRIMARY ENERGY SAVINGS

This simulation experiment aims to estimate the primary energy savings achieved by the two-pipe system. In the previous simulation experiments, the focus was on calculating the energy savings obtained by considering the difference between energy generated and energy demand. However, an additional benefit of the two-pipe systems is related to the use of operating water temperatures close to ambient temperatures. Therefore, a reduction of primary energy use is expected when comparing the two-pipe system with a four-pipe system operating with conventional water temperatures.

The energy performance of the two-pipe system was evaluated through its integration in a reference building model. The geometry of the building model is illustrated in Fig. 3-11. It consists of four perimeter thermal zones and one interior thermal zone. The total floor area is 1660 m^2 with an aspect ratio of 1.5. This five-zone model is representative of one floor of the medium office building prototype, as described in the report, U.S. Department of Energy Commercial Reference Building Models of the National Building Stock [73]. Simulations were run for two construction sets of the building envelope and two conditions related to inter-zone air flows, as shown in Table 3-4. The weather conditions of Copenhagen (Denmark) were used. For details regarding the simulation input parameters, please see Paper V. The graphic layout of the system modeled in Dymola is shown in Fig. 3-12.

Table 3-4 Description of the cases simulated

Case	Construction set	Inter-zone air flow
1	ASHRAE 90.1 2004 (High U-value)	No doors
2	ASHRAE 90.1 2013 (Low U-value)	No doors
3	ASHRAE 90.1 2004	With doors open
4	ASHRAE 90.1 2013	With doors open

The four cases previously described were analyzed for three configurations of the system. Each configuration aimed to highlight a different aspect of the energy savings of the two-pipe system. The three configurations were defined as:

- Ideal configuration
- Ideal configuration with dry cooler
- Real configuration

The configurations are described below.

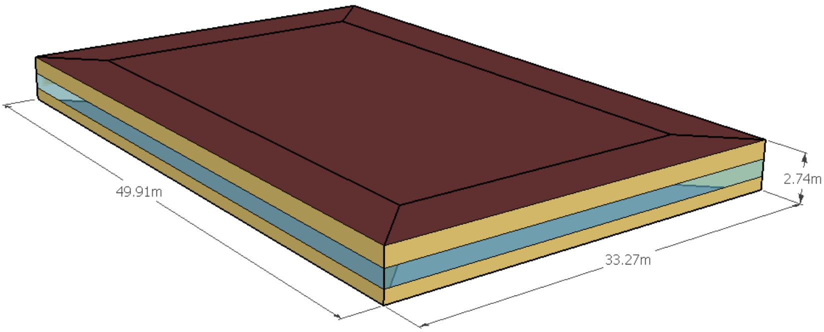


Figure 3-11 Geometry of the typical office building

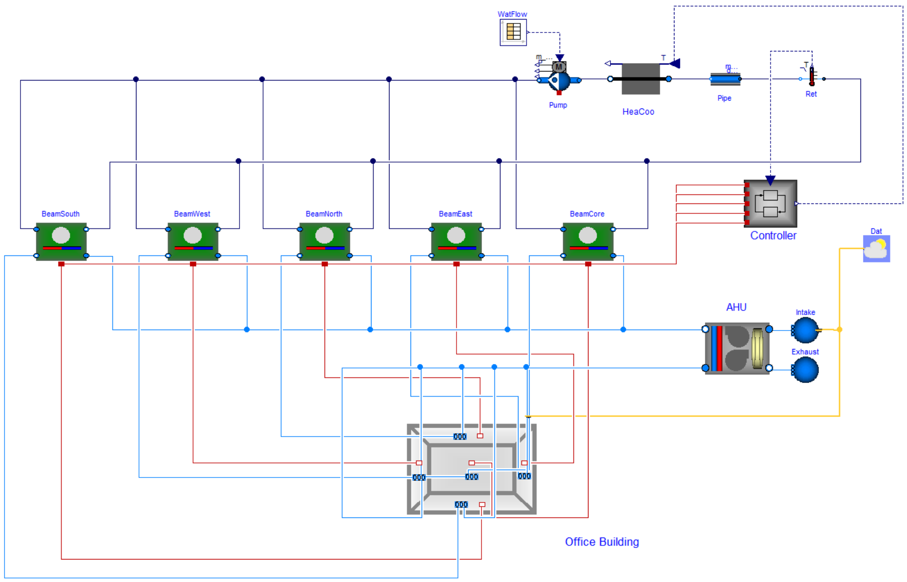


Figure 3-12 Layout of the two-pipe system model in Modelica. Light-blue lines represent air streams, dark-blue lines represent water streams, red lines represent convective heat exchange and temperature signals, and dashed blue lines represent control signals.

The first configuration included Modelica models of ideal plants for both the water and air loops (useful thermal energy). This ideal configuration allowed prediction of the actual energy savings related to the useful heat transferred from warm to cold rooms through the room-temperature loop when simultaneous heating and cooling occurred.

In the second configuration, a dry cooler was added to take advantage of free cooling. The dry cooler was dimensioned to be able to cool the return water to the design temperature condition of 20 °C with a temperature difference of 6 K between water and outside air. Therefore, whenever the outside air temperature was below 14 °C, no cooling energy was required by the thermal plant in the room-temperature water loop.

In the third configuration, the ideal models were replaced by more realistic components. In particular, a reversible air-to-water heat pump model was integrated into the room-temperature water loop. It is worth mentioning that the current version of the Modelica *Buildings* library does not include any model specifically defined as an air-to-water heat pump. Therefore, a new model was developed. The model was based on performance curves related to the Maroon 2 unit by Swegon [79]. More details regarding the heat pump model are provided in Paper V.

The AHU heating coil was supplied by a heat pump with a supply water temperature of 45 °C, while the cooling coil was connected to a chiller with a supply water temperature of 7 °C. Average efficiencies for pumps and fans were set to 0.8.

Energy savings

Fig. 3-13a shows the comparison of the annual heating and cooling useful energy use for the first system configuration. As illustrated, the two-pipe system (2PS) required less useful heating and cooling energy than the four-pipe system (4PS) in all four cases. This means that the room-temperature loop was able to remove heat from the warm core zone and release it to the cold perimeter zones. In particular, energy savings of approximately 17%, 21%, 4% and 6% were achieved, respectively, for cases 1, 2, 3 and 4. Note that the cases with doors open present a significantly lower potential for energy savings when compared with the cases with closed doors (solid surfaces).

Fig. 3.13b illustrates the annual heating and cooling useful energy use for the second system configuration. In this configuration, a dry cooler was added to the two systems. The heat removed by the dry cooler is also shown in Fig. 3-13b for each simulation case. Due to a higher supply water temperature than the 4PS, the 2PS was able to take better advantage of free cooling conditions. The 2PS presents a significantly higher value of heat removed in all four cases. In particular, the dry cooler in the 2PS removed approximately 67%, 70%, 65% and 69% of cooling demand versus 31%, 33%, 16% and 18% in the 4PS, respectively, for cases 1, 2, 3 and 4.

Fig. 3-13c shows the annual heating and cooling primary energy use for the third system configuration. Here, the ideal plants were replaced by heat pump and chiller models. A factor of 2.5 was assumed for the conversion of electricity into primary energy [80]. Additional energy savings were achieved thanks to the lower temperature difference between evaporator and condenser. The 2PS used approximately 46%, 52%, 40% and 45% less primary energy than the 4PS, respectively, for cases 1, 2, 3 and 4.

Fig. 3.13d shows the total annual primary energy use for the four cases. Energy use for ventilation and pumps was added to the values obtained in Fig. 3.13c. When comparing the total primary energy, the 2PS used approximately 18%, 17%, 13% and 12% less energy than the 4PS. As illustrated, fans account for a large share of the total energy, reducing the relative energy savings achieved due to the room-temperature water loop. Since the 2PS circulated water continuously, pumps have higher energy use than the 4PS.

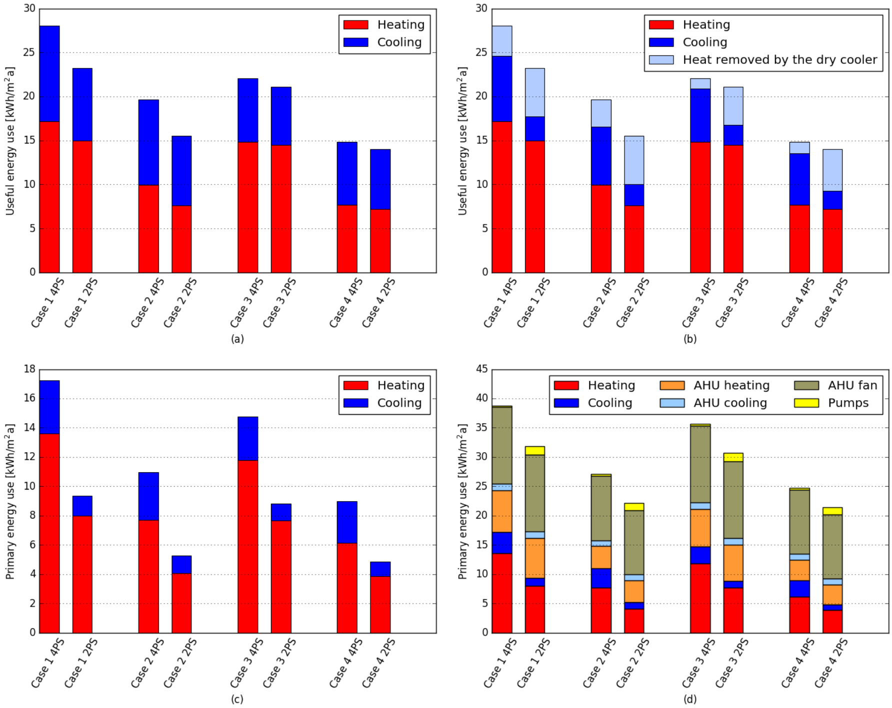


Figure 3-13 Energy use: two-pipe system vs. four-pipe system; (a) ideal configuration, (b) ideal configuration with dry cooler, (c) real configuration – only heating and cooling, (d) real configuration – total.

Indoor climate

Fig. 3-14 illustrates the indoor air temperatures of the five rooms for a typical winter and summer day for simulation case 3. More details and figures regarding the other simulation cases can be found in Paper V.

The results confirm the findings obtained in section 3.2: the controller was able to maintain the air temperatures within the set-points in all the zones.

Note the behavior of the air temperatures during the summer day. It is worth remembering that the supply water temperature is adjusted by taking into account only the air temperature corresponding to the maximum temperature among all the zones in the building at the current time. Therefore, in the morning, when the sun rises, the supply water temperature is set by considering the east-orientated zone, which is the warmest zone in the building and reaches the cooling set-point of 24 °C. In the middle of the day and in the afternoon, the warmest zone is the south-orientated and the west-orientated, respectively.

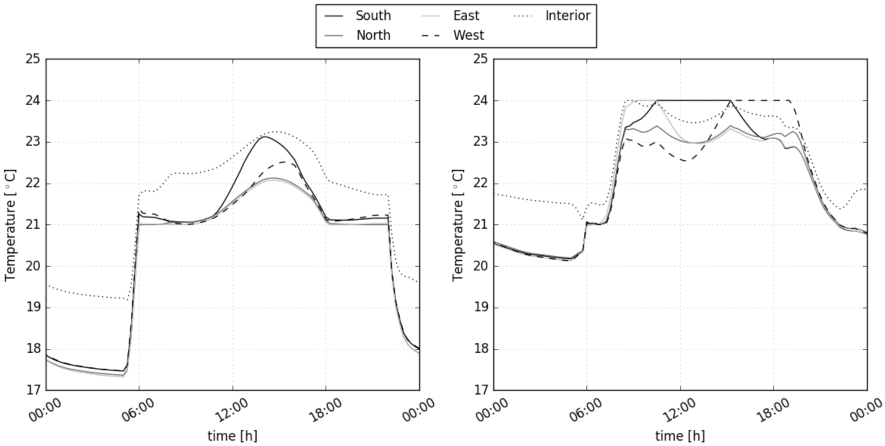


Figure 3-14 Air temperatures for a typical winter (left) and summer day (right)

Fig. 3-15 shows the relative humidity obtained in the five zones. Generally, similar values of relative humidity are seen in the zones. In winter, the relative humidity is about 25-30%, while in summer the relative humidity is about 55-60%.

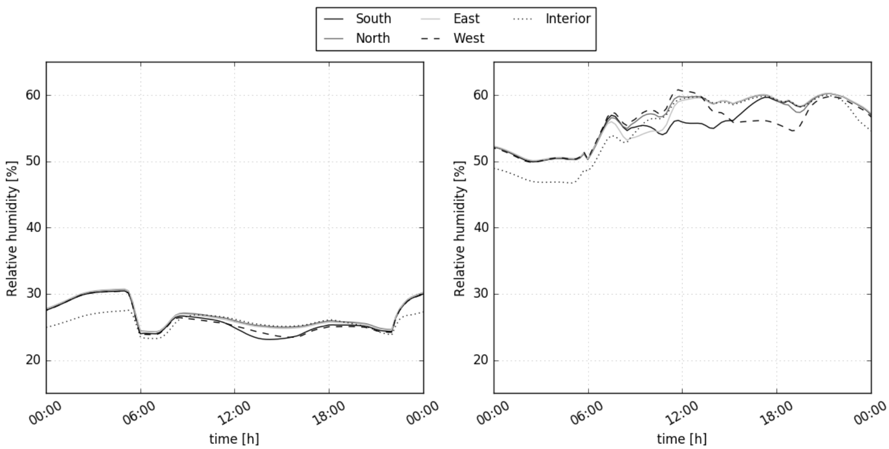


Figure 3-15 Relative humidity for a typical winter (left) and summer day (right)

3.6. SIMULATION EXPERIMENT 6 - CONTROL STRATEGIES FOR THE WATER LOOP

This simulation experiment investigated the energy performance of four control strategies able to regulate the room-temperature water loop. In addition to the two control strategies described in 2.2.2 and used in experiment 3.2, two other control strategies were developed and tested on a typical three-story office building model in two different climates, Chicago (USA) and Copenhagen (Denmark).

Table 3-5 summarizes the four control strategies analyzed. These were named according their most distinctive features. Their level of complexity spans from linear SISO (single-input, single-output) with standard inputs to PI feedback controllers with detailed information about the disturbances acting on the building. All the four strategies were designed to operate with a supply water temperature range between 20 °C and 23 °C.

Table 3-5 Summary of the control strategies analyzed

Name	Water mass flow rate	Supply water temperature
Outdoor air temperature strategy (OATS)	Constant	$f(T_{out})$
Exhaust air temperature strategy (EATS)	Constant	$f(T_{exh})$
Feedback water temperature strategy (FWTS)	Constant	PI controller
Feedback water flow strategy (FWFS)	PI controller	$f(T_{out})$

The *outdoor air temperature strategy* (OATS) and the *feedback water temperature strategy* (FWTS) are, respectively, the open-loop and the closed-loop control strategy previously described in 2.2.2 and used in 3.2.

The OATS presents constant water mass flow rate and variable supply water temperature. The value of the supply water temperature is a function of the outdoor air temperature, as shown in Figure 3-16. The maximum supply water temperature of 23 °C was set in correspondence with the coldest design outdoor air temperature (-20 °C and -10 °C respectively for Chicago and Copenhagen). The minimum supply water temperature of 20 °C was set in correspondence with the warmest design outdoor air temperature (35 °C and 25 °C respectively for Chicago and Copenhagen). Note that the value of the minimum supply water temperature of 20 °C (for Copenhagen climate) was set differently than in section 2.2.2 and 3.2. The *exhaust air temperature strategy* (EATS), similar to the OATS, presents constant water mass flow rate and variable supply water temperature. However, in this case, the supply water temperature was set based on the exhaust air temperature, as illustrated in Figure 3-17. This represents an average of the air temperatures in the zones. The maximum supply water temperature of 23 °C was set in correspondence with an exhaust air temperature of 20 °C, which is the heating design room air temperature. The minimum supply water temperature of 20 °C was set in correspondence with an exhaust air temperature of 24 °C, which is the cooling design room air temperature.

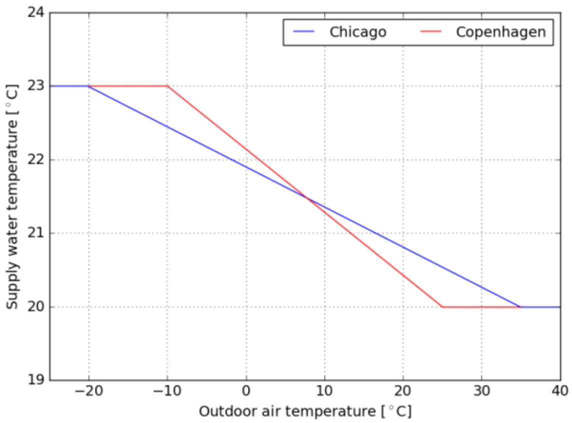


Figure 3-16 Supply water temperature vs. Outdoor air temperature for the OATS

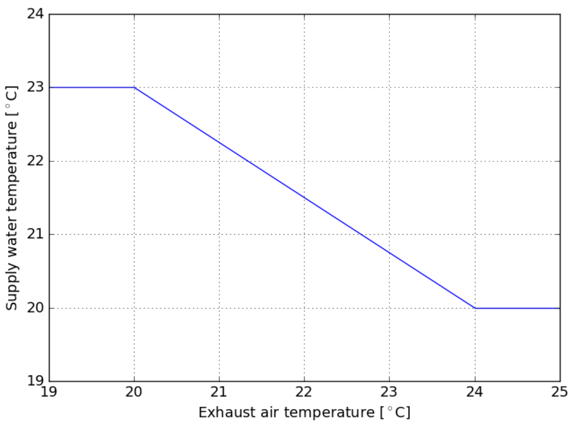


Figure 3-17 Supply water temperature vs. Exhaust air temperature for the EATS

The *feedback water flow strategy* (FWFS) presents variable water mass flow rate and supply water temperature adjusted using the relationship described in Figure 3-16. Due to the absence of valves at zone levels, the water mass flow rate must be prescribed at system level. The actual water mass flow rate can be expressed by the formula

$$\dot{m}_w = k_{flow}\,\dot{m}_{nom} \qquad (3.8)$$

where $\dot{m}_w$ is the actual water mass flow rate, $\dot{m}_{nom}$ is the nominal mass flow rate and k_{flow} is a coefficient in the range of 0.25 and 1. The value of 0.25 was chosen as the minimum water mass flow rate allowed in the circuit plant during operating hours. As in the FWTS, the controller is fed by the signals of the actual air temperatures in the fifteen zones. Whenever the minimum and the maximum zone air temperature are within the heating and cooling set-points, k_{flow} is equal to 0.25, and therefore, the minimum amount of energy is required. Otherwise, the value of k_{flow} is increased just enough to meet the set-points.

The results were calculated in terms of annual electricity use for the air-to-water heat pump (space heating and space cooling), heating and cooling AHU coils, fans and pumps. In Figure 3-18, the energy use of the two-pipe system is presented for the Chicago and Copenhagen climate respectively. Generally, due to more extreme climate conditions, the two-pipe system in Chicago requires more energy than the two-pipe system in Copenhagen.

The ranking provided by sorting the alternative strategies according to their energy performance was consistent for the two climate locations. That is, the FWTS has the highest energy performance with annual electricity energy use of 25.2 and 11.3 kWh/m^2 respectively for Chicago and Copenhagen, subsequently followed by the FWFS (25.4 and 11.4 kWh/m^2 respectively), EATS (26.7 and 12.5 kWh/m^2 respectively) and OATS (27 and 12.6 kWh/m^2 respectively).

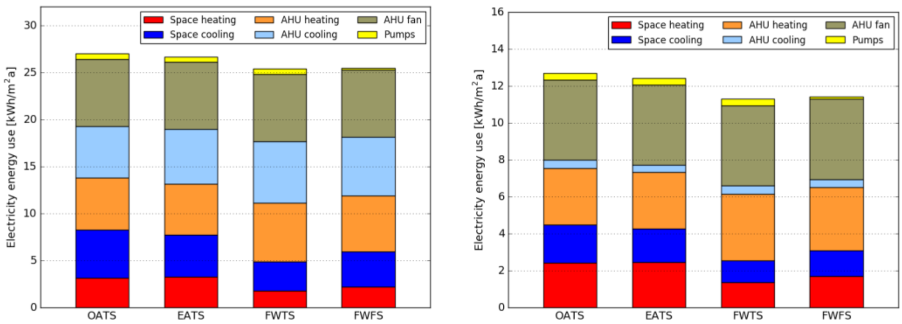


Figure 3-18 Annual electricity use for Chicago (left) and Copenhagen (right)

No significant difference is noticed when comparing the two SISO strategies (OATS and EATS). The relative difference is approximately 1% for both climates.
Similarly, the two PI feedback control strategies (FWTS and FWFS) present analogous values of total annual electricity use. Also in this case, the relative difference is approximately 1% for both climates.

Larger differences are noticed between the SISO strategies and the PI feedback strategies. In particular, when comparing the most efficient strategy (FWTS) with the least efficient strategy (OATS), energy savings of approximately 7% and 10% were achieved for Chicago and Copenhagen, respectively.

Note that these results differ from the results obtained in simulation experiments 3.2 (Copenhagen climate). In the previous simulation experiment, it was found that the implementation of the closed-loop control system (FWTS) led to a reduction of the annual energy use by 46% when compared with the open-loop control system (OATS). In this case, the reduction was only 10%.

Beside the differences in the building model, one of the reasons for the difference between the two simulation experiments is related to the curve representing the supply water temperature in the OATS. In section 3.2, this function was represented by Figure 2-8, while in this case by Figure 3-16. Therefore, the curve used in this simulation experiment generally provided less cooling effect than the curve in section 3.2, leading to smaller cooling energy use.

A second reason is related to fact that in section 3.2, no dry cooler was integrated in the thermal plant. Free cooling potential is expected to be higher for the OATS, where more cooling energy is generally used. For example, the cooling energy required in winter for the OATS (see Fig. 3-7) would be completely provided by the dry cooler, with no energy use by the cooling machine.

A third reason refers to the difference in presenting the results. In section 3.2, the energy use is intended as useful energy use. Here the energy use is intended as electricity use. When considering electricity use instead of useful energy use, parameters such as COPs (for heat pumps and chillers) and efficiencies (for pumps and fans) are considered.

CHAPTER 4. CONCLUSION

In this work, a novel two-pipe system integrating a room-temperature water loop was studied. A detailed computational model of the system was developed by using Modelica. Several dynamic energy simulations were carried out in order to study the behavior of the system under different conditions.

Based on the results achieved in this work, the following conclusions can be drawn:

- It is possible to design a well-functioning two-pipe system that operates a room-temperature water loop, together with active beams, to provide heating, cooling and ventilation to buildings.

- The innovative hydronic layout of the two-pipe system causes a difference between the energy delivered by the thermal plant (energy generated) and the energy transferred by the active beams to the spaces (energy demand). This is evidence of the ability of the two-pipe system to transfer energy from warm to cold zones through the room-temperature water loop. The difference between energy generated and energy demand not only represents the energy transfer among zones, but also the energy savings achieved by the two-pipe system if compared to a fictional four-pipe system running with exactly the same operating conditions and reaching exactly the same indoor thermal climate. Note that in a four-pipe system, the energy generated equals the energy demand.

- The amount of energy savings (in terms of difference between energy generated and energy demand) depends on several factors. The base model used for the sensitivity analysis showed that annual energy savings of about 17% can be achieved. A series of alterations was made to the base model in order to reveal the influence of eight factors on the amount of energy savings. It is seen that, depending on the scenario considered, the relative energy savings range between approximately 6% and 35%. The most influential factor resulted in internal heat gains with an absolute influence coefficient of about 1.9.

- The water circuit can be regulated by using open-loop and closed-loop control systems. Generally, both strategies are able to keep the room air temperatures within desired values. An open-loop control system is simple and cheap (does not need temperature sensors in rooms). On the other hand, a closed-loop control system requires the installation of temperature sensors in each zone, leading to higher installation costs. However, room air temperatures are kept within desired set-points by minimizing the energy use in the thermal plant through PI controllers. The integration of closed-loop controllers into the two-pipe system made it possible to reduce the annual energy use by approximately 7-10% when compared to the open-loop controllers, depending on the climate conditions.

- By operating water temperatures close to ambient temperature, the two-pipe system can more easily take advantage of sustainable energy sources, with a consequent reduction of primary energy use. In particular, due to the higher supply water temperature in cooling mode, a dry cooler can significantly reduce the cooling energy use. Simulation results showed that the dry cooler in the two-pipe system can remove between 65% and 70% of cooling demand against between 16% and 33% in the four-pipe system. Due to the lower supply water temperature in heating mode, a heat pump integrated in the two-pipe system can achieve a value of the heating seasonal COP 48% higher than the heat pump in the four-pipe system. This allowed for a significant reduction of primary energy use for space heating. When comparing the total annual primary energy use, the two-pipe system used approximately 12% to 18% less total primary energy (including ventilation) per year than the four-pipe system.

- The design of the two-pipe system does not allow individual control of the air temperature in the thermal zones. The supply water temperature is adjusted by taking into account only the zone temperature corresponding to the maximum or minimum temperature among all the zones in the building at the current time. However, proper dimensioning and control of the system ensures that air temperatures are always within the desired set-point values. Note that, in some countries, the possibility to individually control the air temperature in rooms might be a requirement imposed by policy makers. Therefore, the two-pipe system presented in this thesis would not be allowed. In some regions, for example in Basel Stadt, Switzerland, automatic temperature control for each room is only required if the supply water temperature at design conditions is higher than 30°C [81].

- Due to lower temperature differences between room air and water in the active beams, the two-pipe system requires approximately four-times more heat transfer area than a four-pipe system operating with conventional water temperatures. This means that a larger number of active beam units have to be installed. On the other hand, the two-pipe system needs only one water pump, fewer pipes and no control valves.

Future work

Based on the work carried out in this study, recommendations for future research are presented below.

I. The two-pipe system was recently implemented in a newly constructed office building in Sweden. Real-life monitoring and analysis of the system is a current task under a new scientific project funded as a direct consequence of the results achieved in this work. This new scientific project started in fall 2016.

II. The Modelica model developed within this thesis could be refined in order to resemble the real system installed in Sweden. This model would provide a base for further improvements of the energy performance of the real system. For example, the regulation strategy currently adopted in the real system is similar to the open-loop controller described in section 2.2.2. Therefore, the refined Modelica model could be used to predict the actual energy and cost savings achieved thanks to the integration of the closed-loop controller.

III. Fig. 3-13d showed that fan energy use is a large share of the total annual primary energy use of the system. Therefore, strategies such as VAV or demand-controlled ventilation should be studied.

IV. In this thesis, the two-pipe system was studied only when coupled to office buildings. However, other building types such as hotels or hospitals should be considered.

V. The efficacy of other sustainable energy sources in connection to the two-pipe system should be investigated. Geothermal heat pumps, solar collectors and phase change materials seem promising technologies to further reduce the primary energy use of the two-pipe system.

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APPENDIX A. ENHANCED ACTIVE BEAM MODEL

As mentioned in section 2.2.1, an enhanced active beam model was developed in collaboration with LBNL within the IEA EBC Annex 60 project. The participation in the IEA EBC Annex 60 project was financially supported by the Danish Energy Agency, under the Energy Technology Development and Demonstration Program (EUDP).

In particular, two active beam models were developed: a model for cooling only, and a model for heating and cooling. The graphic layout of the model for heating and cooling is illustrated in Fig A-1. More details can be found in [63].

The main difference between the model described in section 2.2.1 and this model is related to the fact that in the latter there is no explicit modeling of the rate at which room air is induced over the convector (heat exchanger). The rationale for this implementation is that the amount of induced air is generally not known from manufacturers' catalog data. This also avoids having to add an extra air flow path for the air induced from the room. Therefore, this model does not heat/cool the supply air between the ports *air_a* and *air_b*. The heat flow rate from the convector is transmitted to the room through the heat port *heaPor*.

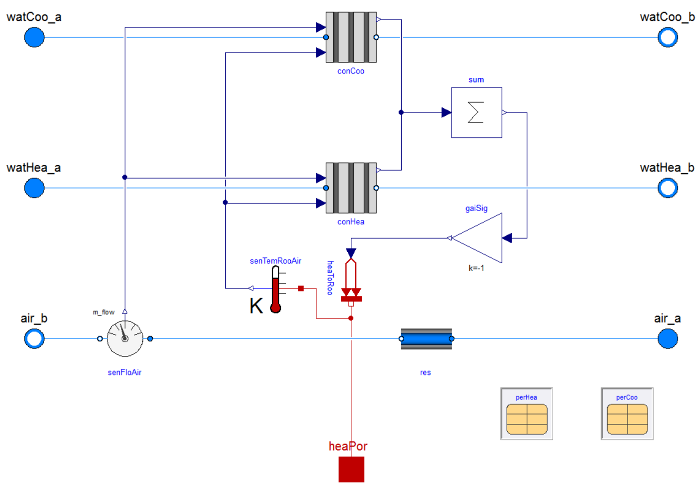

Figure A-1 Graphic layout of the enhanced active beam model.

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