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Ventilative cooling in Living Lab

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MASTER THESIS

for

Student Cathrine Kirkøen

Spring 2015

Ventilative cooling in Living Lab
Ventilasjonskjøling av Living Lab

Background and objective

In modern residential buildings like passive houses and Zero Energy Buildings there will be a requirement for removing excess heat for longer periods than in ordinary buildings. These buildings usually have a mechanical balanced ventilation system, designed to secure a satisfactory air quality.

To control the indoor air temperature without the use of mechanical cooling, increased use of the cooling effect of outdoor air (ventilative cooling) may be necessary. The most energy efficient solution would be to use natural ventilation in periods when heat recovery is not needed.

To study different ZEB strategies a detached residential building is under construction at the NTNU campus. The building is called Living Lab because it will be occupied during measurements. It is a part of Centre for Environment-friendly Energy Research (FME) ZEB.

To study the cooling effect and thermal comfort related to ventilative cooling the simulation tool IDA ICE should be used. The Living Lab should be used as a case.

The objective of the work is to study how to apply ventilative cooling and to evaluate the effect on the indoor environment and use of energy in residential building.

The work will be a continuation of the specialization project.

The following tasks are to be considered:

1. Supplement the literature survey conducted in the project work
2. Establish simulation models in IDA ICE
3. Verify different parts of the model, like for instance airflow through open windows and effect of wind
4. Perform simulation for different user conditions, solutions and control strategies

5. Draw general conclusion on the applicability of ventilative cooling on super insulated residential buildings with regard to thermal comfort and energy use.

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

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The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 14. January 2015

Olav Bolland
Department Head

Hans Martin Mathisen
Academic Supervisor

Research Advisor:
Maria Just Alonso

Preface

This thesis is written at the Institute for Energy and Process Engineering at the Norwegian University of Science and Technology (NTNU) in Trondheim, spring 2015. It is a master thesis performed as a part of the engineering study Energy and Environmental science. Professor Hans Martin Mathisen has been the main supervisor, representing NTNU. Maria Justo Alonso has worked as co-supervisor, representing SINTEF Building and Infrastructure.

I wish to express gratitude towards Hans Martin and Maria for excellent guidance throughout the whole year. A special thanks goes also to Bartosz Burzawa for valuable help conducting CFD-simulations. I am grateful for all the help and information I have received from Steinar Grynning, Luca Finocchiaro and Francesco Goia. I also wish to thank my family for support and motivation. Appreciation to Eivind Bere for keeping me happy and motivated is also in its rightful place.

Cathrine Kirkøen

Trondheim, June 7, 2015

Sammendrag på norsk

I denne oppgaven studeres ventilasjonskjøling av godt isolerte boliger. Det fokuseres på termisk komfort og energibruk. Studiet tar utgangspunkt i Living Lab, en frittliggende bolig på ca. 100m^2 utformet med et såkalt ”mixed-mode” ventilasjonssystem. Boligen bygges på campus NTNU i Trondheim i Norge. Simuleringsverktøyet IDA ICE 4.6 har blitt benyttet for å studere ventilasjonskjøling i Living Lab. Resultatene fra simuleringen har blitt sammenliknet med funn fra tidligere studier for å kunne trekke generelle konklusjoner om anvendbarheten til ventilasjonskjøling i lavenergiboliger.

Resultatene fra simuleringene tilsier at det vil være en betydelig risiko for overoppheting i Living Lab dersom ingen aktive eller passive kjøletiltak benyttes. Det fremkommer likevel at ventilatsjonskjøling kan forhindre overoppheting i Living Lab uten å medføre betydelig økning i bygningens energiforbruk. På den andre siden peker resultatene mot at det ikke er mulig å fullstendig eliminere risikoen for overkjøling som følge av ventilasjonskjøling. Likevel viser det seg at antall timer med overkjøling kan holdes på et akseptabelt nivå. Nattkjøling ser ikke ut til å ha noen positiv effekt på det termiske miljøet i bygget. Solstråling, utetemperatur og tilstedeværelse av beboere er de faktorene som har størst innvirkning på kjølebehovet i Living Lab. Simuleringene indikerer at åpningsarealene til vinduene i boligen generelt ikke er begrensende for kjølingen. Studiet kom fram til at den beste måten å utnytte ventilasjonskjøling i Living Lab er å implementere et såklat ”concurrent mixed-mode” system der kontrollsystemet for vinduene kun er aktivt på dagtid. Systemet bør utformes slik at sørvinduet og de høytliggende vinduene åpnes maksimalt når innetemperaturen overskrider 24^{o}C og lukkes når innetemperaturen synker under 22^{o}C. Simuleringene viser at et slikt system reduserer antall registrerte timer med overoppheting uten bruk av ventilasjonskjøling med 99%. Samtidig holdes antall timer med overkjøling på et moderat nivå, 48 timer/år ble registrert. Utnyttelse av et slikt ventilasjonskjølesystem vil resultere i en økning i energiforbruk på 52 kWh/år og 4 kWh/år sammenliknet med kun bruk av henholdsvis moderat og forsterket mekanisk ventilasjon.

Denne oppgaven og tidligere forskning viser at overoppheting i lavenergiboliger ofte er et problem. Dette må derfor tas hensyn til ved utforming av slike bygg. Overoppheting i lavenergiboliger kan forhindres ved å benytte ventilasjonskjøling. Ventilasjonskjøling har vist seg å kunne ha en betydelig positiv effekt på det termiske miljøet uten å ha betydelig negativ effekt på energiforbruket. I noen tilfeller vil det til og med kunne redusere det totale energiforbruket. Overkjøling kan derimot være et problem ved benyttelse av ventilasjonskjøling. Systemet må utformes med nøyaktighet og forsiktighet for at ventilasjonskjølingen skal ha ønsket effekt. Utformingen bør være individuell for ulike bygg og ulike klima. Et mer komplekst naturlig ventilasjonssystem krever med nøyaktig utforming. I tillegg må det akseptable temperaturområdet for mekanisk ventilerte bygg ofte justeres for bygg tiltenkt å bruke ventilasjonskjøling. Til tross for at nattkjøling ikke er anvendbart i enkelte lettvektsboliger, har det vist seg å gi ønsket effekt i bygg med mer termisk masse. For at ventilasjonskjøling skal sikre et tilstrekkelig godt termisk innemiljø er det ofte nødvendig med automatiske kontrollsystemer for vinduer. Brukerne bør likevel ha mulighet til å oversstyre det automatiske systemet.

Abstract

This thesis is a study of ventilative cooling in super insulated residential buildings with focus on thermal comfort and energy use. The case of the study is Living Lab, an approximately $100m^2$ detached residential building designed with a mixed-mode ventilation system. The building is currently under construction at NTNU campus in Trondheim, Norway. The simulation software IDA ICE 4.6 has been used to study ventilative cooling in Living Lab. The results from the simulations have been compared to findings from previous studies. General conclusions on the applicability of ventilative cooling in low-energy dwellings have been drawn.

The results from the simulations imply that there will be a severe risk of overheating in Living Lab if no active or passive cooling techniques are applied. Moreover, the results show that ventilative cooling can prevent overheating without significantly increasing the energy demand. Due to the uncertainties related to increased air velocities, it is not possible to eliminate the risk of overcooling caused by ventilative cooling completely. However, the simulations show that overcooling can be held at an acceptable level. The factors most influencing the need for ventilative cooling in Living Lab are in the following order: solar radiation, outdoor temperature and occupancy. The simulations indicate that the openable window area is not a limiting factor for cooling. The study found that the best way to apply ventilative cooling in Living Lab is to implement a concurrent mixed-mode system where the window control system is only active during the day. This system should be designed to open the south and skylight windows to maximum opening when indoor air temperatures exceed 24^oC and close them when indoor air temperatures drops below 22^oC. Simulations reveal that this system would reduce the number of overheated hours recorded when not utilizing ventilative cooling with 99%. The number of overcooled hours was kept at a moderate level, 48 hours/year. This ventilative cooling system would increase energy demand with 52 kWh/year and 4 kWh/year compared to use of only hygienic mechanical ventilation and only enhanced mechanical ventilation, respectively.

This assignment and previous research show that overheating in low-energy dwellings is often an issue. It should therefore be addressed during the design process. Overheating in low-energy dwellings can be prevented with ventilative cooling. Ventilative cooling can have a significant positive effect on the thermal environment without having a significant negative effect on the use of energy. In some cases, energy consumption can even be reduced when applying ventilative cooling. Overcooling can be an issue when utilizing ventilative cooling. A careful design process is needed for ventilative cooling to have the desired effect. The process should be individual for each building and climate. A more complex natural ventilation system requires a more accurate and careful design process. Also, the acceptable indoor temperatures for mechanically ventilated buildings often have to be adjusted for buildings intended to utilize ventilative cooling. Even though nighttime ventilative cooling is not applicable in certain lightweight dwellings, it has proven to be effective in buildings with more thermal mass. An automatic window control system is often necessary for ventilative cooling to achieve the desired thermal environment. The occupants should, however, be able to overrule the automatic system.

Contents

Nomenclature

ρ	Air density [kg/m^3]
ρ_i	Indoor air density [kg/m^3]
ρ_o	Density air outside [kg/m^3]
ε	Contraction coefficient
ζ_i	Loss coefficient through inlet window
ζ_o	Loss coefficient through outlet window
A_i	Area window used for supplying air [m^2]
A_o	Area window used for extracting air [m^2]
C_b	Loss coefficient
g	Acceleration due to gravity [m/s^2]
H	Height difference between inlet window and outlet window [m]
h	Width of the gap [m]
i	Impulse coefficient
k	Wind direction factor
p_l	Pressure loss [Pa]
$p_{d,buoy}$	Driving pressure from buoyancy [Pa]
$p_{d,wind}$	Driving pressure from wind [Pa]
p_{sat}	Saturation pressure water vapor [Pa]
p_{tot}	Atmospheric pressure [Pa]
R_{air}	Specific gas constant air [J/kgK]
R_{water}	Specific gas constant water [J/kgK]
T	Air temperature [K]
U_m	Maximum air velocity at distance x from inlet [m/s]
U_o	Air velocity at inlet [m/s]
v_a	Wind velocity [m/s]
v_i	Air velocity through inlet window [m/s]
v_o	Air velocity through outlet window [m/s]
x	Distance from inlet to chosen point [m]
X	Relative humidity [%]
x_p	Distance from inlet to polar point, virtual source [m]

1 Introduction

1.1 Motivation

According to EU-Directive (2010), buildings account for 40% of the total energy consumption in Europe. Reducing energy consumption in the buildings sector is therefore important to reduce our our green house gas emissions.

The increased focus on reducing energy use in buildings have led to the development of passive houses and Zero Emission Buildings. In such buildings it will be required to remove excess heat for longer periods of time than in ordinary buildings. These buildings usually have a mechanical balanced ventilation system designed to secure satisfactory indoor air quality. To control the indoor air temperature without the use of mechanical cooling, increased use of the cooling effect of outdoor air (ventilative cooling) may be necessary. The most energy efficient solution would be to use natural ventilation in periods when heat recovery is not needed.

1.2 About Living Lab

This master thesis is a part of a bigger research project, Living Lab, under the Centre for Environment-firendly Energy Research (FME) ZEB. (ZEB, 2014a). Living Lab is an approximately $100m^2$ single family house realized with state-of-the-art technologies for energy conservation measurements and renewable energy source exploitation. Different solutions and building equipment are planned to be installed, so that several options can be tested within the same building. It is built to demonstrate how CO_2-neutral constructions can be realized in the Nordic climate and also to conduct research on how occupants interact with the technologies in low-energy dwellings. (Finocchiaro et al., 2014).

Zero Emission Buildings have zero emissions of greenhouse gases related to their production, operation and demolition. (ZEB, 2014b). Hence, all $\mathrm{CO_2}$-emissions from operation of the building and for production, transport and demolition of building materials and components during the life cycle, must be compensated for by production or transformation of renewable energy sources at the building site. (Jelle and Gustavsen, 2014).

A building-integrated photovoltaic system is installed in Living Lab, on the two slopes of the roof. The total installed power is 12.5kW. The energy converted by the system is expected to cover the energy need of the building and to balance energy embedded in the materials and components used to realize Living Lab. (Finocchiaro et al., 2014).

The dwelling is designed with low U-values to minimize heat loss during winter. Heating, ventilation and domestic hot water are planned to be satisfied by a water-to-water heat pump, which is coupled with a ground heat-exchanger. The output of the heat pump is connected to a two-storage heat tank with two auxiliary electric coils that can be activated if necessary. Four building-integrated solar thermal panels are installed on the south facade. They are directly connected to the centralized water-based heat storage. (Finocchiaro et al., 2014).

Figure 1: Living Lab south and east facades. Printed with permission (Finocchiaro et al., 2014)

Living Lab is equipped with floor heating in occupied areas and a low-temperature radiator in the living room. There are no mechanical cooling options in Living Lab. (Mathisen, 2015). The ventilation is designed as a mixed-mode hybrid system with mechanical balanced ventilation. The dwelling has openable windows on all facades. It is to utilize both stack and cross-flow ventilation through mechanical opening of windows. The control system for the windows is yet to be constructed. The building should mainly utilize direct ventilative cooling during the daytime. It is uncertain whether night cooling of the building structure will be applied. (Finocchiaro, 2015).

1.3 Scope

This master thesis is a continuation of the specialization project conducted fall semester 2014. The objective has been to study how to apply ventilative cooling and to evaluate the effect on the indoor environment and use of energy in residential buildings. It was decided with the main supervisor, to focus on the thermal comfort aspect of the indoor environment.

Theory on thermal comfort has been reviewed to define the desired thermal environment in Living Lab. Also, theory regarding ventilative cooling and mixed-mode ventilation has been studied to acquire base knowledge on the subject.

It was determined along with the main supervisor that the literature study conducted in the specialization project was sufficient to cover the subsequent work. It has been included in this report and used to draw general conclusions on the applicability of ventilative cooling.

To support the choice of simulation software and validate the final results, parts of the IDA ICE model have been verified. The most crucial aspects when evaluating ventilative cooling are how the simulation software models airflow through open windows and the progress of changing indoor air temperature. It was decided, along with the main supervisor, that the validation should focus on these aspects and be based on the work conducted in the specialization project.

To study how to apply ventilative cooling in Living Lab it was decided to focus on the window control system first. In that context, two control systems have been developed. The first system utilized stack ventilation only. It used indoor air temperature sensors and maximum opening/no opening switches. The other system utilized both stack and cross-flow ventilation. It combined outdoor and indoor air temperature sensors with PI regulators. It was decided to first determine how to apply window control in terms of set-point temperatures. Then, in terms of when the window control system should be active. Later, the window control systems were combined with mechanical ventilation in mixed-mode systems for evaluation of how to apply a complete ventilative cooling system.

In order to study the effect of ventilative cooling on thermal comfort and use of energy, whole year simulations using the best solutions for window control have been performed. Simulations have been conducted using window control only in a natural ventilation system, in combination with mechanical ventilation in a concurrent mixed-mode system and in a change-over zoned mixed-mode system. The results from these simulations have been compared to whole year simulations using only mechanical ventilation, both to use of hygienic mechanical ventilation and enhanced mechanical ventilation.

The main findings from the study of ventilative cooling in Living Lab have been put in context with the literature survey. General conclusions on the applicability of ventilative cooling on low-energy dwellings have been drawn.

2 Background

This thesis will determine how to apply ventilative cooling in Living Lab. It will also evaluate the effect of ventilative cooling on thermal comfort. In that context, a definition of the desired thermal environment in Living Lab is needed. This section will elaborate on thermal comfort aspects and review the appropriate requirements and standards. To achieve the desired thermal environment, Living Lab is to utilize ventilative cooling through a mixed-mode ventilation system. The system combines natural ventilation through automatically controlled windows and mechanical balanced ventilation. The current section will therefore also present theory regarding natural ventilation, mechanical balanced ventilation, mixed-mode ventilation and the concept of ventilative cooling. In addition to the study of ventilative cooling in Living Lab, this thesis will draw conclusions on the applicability of ventilative cooling in super insulated buildings in general. In order to do so, a literature survey has been performed. This section provides an overview of the literature survey.

2.1 Thermal comfort

"Byggteknisk forskrift" (TEK10) provide requirements for thermal environments in Norwegian buildings. Related guidelines are found in Standard NS-EN 15251. These guidelines are based on concepts of body heat balance, activity and clothing level and methods for predicting occupant satisfaction.

2.1.1 The concept of thermal comfort

Thermal comfort is defined as a condition of mind which expresses satisfaction with the thermal environment. Dissatisfaction may be caused by warm or cool discomfort of the body as a whole or by an unwanted cooling or heating of one particular part of the body. Because of individual differences, it is impossible to specify a thermal environment that will satisfy all occupants. There will always be a percentage of dissatisfaction. But, it is possible to specify environments predicted to be acceptable by a certain percentage of the occupants. (Fanger, 1970).

Heat balance of the body

Existing methods for evaluating the general thermal state of the body in comfort considerations are based on an analysis of the heat balance of the human body, see Equation (1). (Fanger, 1970).

$$S = M - W - C - R - E_{sk} - C_{res} - E_{res} - K[W/m^2] \tag{1}$$

S = Heat storage in body
M = Metabolic heat production
W = External work
C = Heat loss by convection
R = Heat loss by radiation
E_{sk} = Evaporative heat loss from skin
C_{res} = Convective heat loss from respiration
E_{res} = Evaporative heat loss from respiration
K = Heat loss by conduction

These parameters have to be in balance so that the combined influence will result in a thermal storage of the body equal to zero. A negative thermal storage indicates that the environment is too cold. A positive thermal storage indicates too warm. Several factors influence the heat balance; activity level, thermal resistance of clothing, evaporative resistance of clothing, air temperature, mean radiant temperature, air speed and partial vapor pressure. (Fanger, 1970).

Even though heat balance is achieved, a person may still find the thermal environment uncomfortable if local influences on the body are experienced. Such local influences can be caused by asymmetric radiation, draught, vertical air temperature differences or contact with hot or cold surfaces.

Activity and clothing level

All assessments of thermal environments require an estimate of metabolic heat production of the occupants and the insulation level in their clothing. Metabolic rate depends on the activity level of a person and is measured in the unit met. One met is the activity level of a relaxed seated person. It is equivalent to 58 W/m^2. The area refers to body surface. The thermal resistance of clothing is measured in the unit clo. One clo is equivalent to 0,154 m^{2}K/W. Current standards provide typical values of met associated with typical activities and clo values for different clothing ensembles or typical seasonal clothing. (Nilsson, 2003).

Predicting occupant satisfactory

Fangers predicted mean vote (PMV) can be used to evaluate whether a given thermal environment complies with the comfort criteria specified and to establish requirements for different levels of acceptability. The PMV index predicts the mean value of the thermal votes of a large group of people exposed to the same environment. To quantify the degree of comfort, the PMV index gives a value on a 7-point thermal sensation scale: +3 hot, +2 warm, +1 slightly warm, 0 neutral, -1 slightly cool, -2 cool, -3 cold. The PMV is calculated based on clothing, activity, air temperature, mean radiant temperature, air speed and humidity. (Fanger, 1970). The equation for calculating PMV can be found in the Appendix Section A.1.1.

To predict the number of people likely to feel uncomfortably warm or cold, the predicted percentage dissatisfied (PPD) index can be used. The PPD index predicts the percentage of a large group of people voting hot (+3), warm (+2), cool (-2) or cold (-3) on the 7-point thermal sensation scale. (Fanger, 1970). The equation for calculating PPD can be found in the Appendix Section A.1.2.

The most relevant factor for analyzing local discomfort when studying ventilative cooling is draught. It is also one of the most critical factors for local discomfort in general. Draught is a common cause for occupant complaints in ventilated spaces. People performing low activity are sensitive to air velocities. Fluctuations of the air velocity have a significant influence on a persons sensation of draught. (Awbi, 2008). An equation for estimating the percentage of people feeling draught (DR) can be found in the Appendix Section A.1.3.

2.1.2 Requirements from TEK 10

TEK10 §13-4 states that the thermal environment in areas for permanent residence shall be arranged according to health and sufficient comfort considerations of its intended use. It is recommended that the operative temperature is kept within 19-26^{o}C when light work is expected. However, on days with high outdoor temperatures it is difficult to keep the indoor temperature from rising above 26^{o}C. It is therefore recommended that indoor temperatures above the upper limit is accepted on warm summer periods for 50 hours in a normal year. It is also mentioned that somewhat higher indoor temperatures can be accepted in shorter periods for dwellings without equipment for cooling installed. This is because the occupants have a larger personal impact on the thermal environment and possibilities for adjusting to higher indoor temperatures in dwellings. It is also recommended that temperature differences between the lower and upper part of the body above 3-4^{o}C are avoided. Also, daily or periodic variations larger than 4^{o}C should not occur. The recommendations from TEK 10 states that the ventilation system should be designed so that airflow and supply air temperature fulfill the need for cooling without causing draught or noise. (TEK10, 2010).

2.1.3 Recommendations from standard NS-EN 15251

Standard NS-EN 15251 (2007) recommends operative temperature design values for different types of buildings. When obtaining recommendations from this standard, the level of desired user satisfaction for the specific building has to be determined. The standard defines three categories representing different levels of expectations. Category number one represents high level of expectations. It should be used when designing buildings with very sensitive and vulnerable occupants with special needs. This level of expectation is not needed in the current building. Category number two, represents a normal level of expectation. It should be used in new or rehabilitated buildings. (NS-EN-15251, 2007). This category is suitable for Living Lab. Category number two is associated with PMV in the range of -0,5 to +0.5, a PPD lower than 10%, and DR lower than 20%. (NS-EN-15251, 2007).

Standard NS-EN 15251 (2007) gives different recommended values for indoor temperature based on whether the building is equipped with a mechanical cooling system or not. All ventilation driven by fans are considered to be mechanical cooling. Hence, the thermal conditions in Living Lab can be evaluated according to these recommendations. However, during summer, natural ventilation through windows will be the main way of controlling the indoor temperature. NS-EN 15251 (2007) states that temperature requirements for dwellings without mechanical cooling are applicable for dwellings with fans installed, as long as the mechanical ventilation utilizes unconditioned air. Also, the opening and closing of windows have to be the main way of controlling the thermal environment.

In dwellings with mechanical cooling it is recommended that bedrooms, kitchen and living rooms have a minimum operative temperature of 20^oC in the winter and a maximum operative temperature of 26^oC in summer. The design values are given with the assumption of 1.2 met. It is assumed a clo level of 1.0 in the winter and 0.5 in summer.

In dwellings without mechanical cooling the recommended operative temperatures are given as a function of the continuous middle value of the outdoor temperature (T_{rm}). For category number two, these recommendations are presented in Equation (2) and (3). (NS-EN-15251, 2007).

$$T_{max} = 0,33 \times T_{rm} + 18,8 + 3[K] \tag{2}$$

$$T_{min} = 0,33 \times T_{rm} + 18,8 - 3[K] \tag{3}$$

NS-EN-15251 (2007) acknowledges that increased air velocity can be used to offset the warmth sensation caused by increased temperature. The maximum temperatures given in Equation (2) can therefore be increased when the air velocity is adequately high. The temperature correction as a function of air velocities is given in Figure 2.

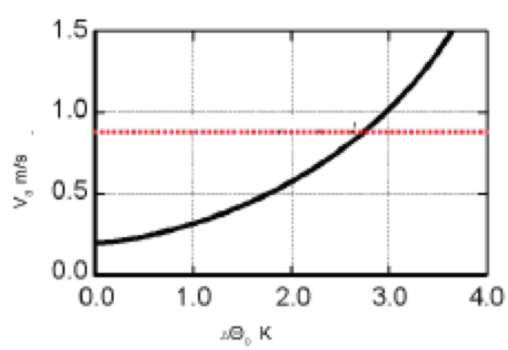


Figure 2: Air velocity to compensate for increased temperature (NS-EN-15251, 2007)

2.2 Ventilative cooling

2.2.1 Natural ventilation

Natural ventilation utilizes only the natural forces wind and thermal buoyancy to supply and extract air. The effectiveness of natural ventilation is determined by the prevailing outdoor conditions; wind speed, temperature, humidity and surrounding topography. It is also determined by the building itself; orientation, number of windows or openings, their size and location. Natural ventilation can provide higher indoor environmental quality, higher degree of end user satisfaction and lower energy use and environmental impact compared to mechanical ventilation. However, a major disadvantage is the uncertainty in performance. It can also result in an increased risk of draught problems and unacceptable thermal comfort conditions during summer. The effectiveness of natural ventilation depends greatly on the design process. Ventilation systems using only natural forces have to be designed together with the building. The building itself and its components are elements that can reduce or increase air movement as well as influence the air content. (Heiselberg, 2008).

Wind
When wind pressure acts on a building it generates a positive pressure on the windward side and negative pressure on the opposing side and in the wake region of the side facades. This causes wind to enter the building on the positive pressure side, and escape through the sides with negative pressure. (Liddament, 1996).

Buoyancy
Buoyancy is the upward force experienced by a body of fluid at a higher temperature than the fluid which surrounds it. Differences in density between the indoor air and the surrounding outdoor air create an imbalance in the pressure gradients of the internal and external air masses. Hence, a vertical pressure difference occur. (Liddament, 1996). To acquire a cooling effect, the outdoor air temperature have to be lower than the indoor air temperature. The indoor air then obtain a buoyancy force equal to the weight of the suppressed outdoor air. The air will enter the building through the openings at the lower part of the building and exit through openings at higher levels. A large temperature difference creates a bigger driving force. (Stensaas, 2001).

A natural ventilation system will often rely on both wind and thermal buoyancy as driving forces. However, one of them will be predominant, and both the building and ventilation system should be designed for optimal utilization of this driving force. The dominating natural driving force has consequences for the shape and layout of the building, for the selection of ventilation elements and for the air paths into, out of and through the building. The natural ventilation principles can be divided into three types; single-sided, cross-flow and stack ventilation. (Heiselberg, 2008). The ventilation principles are illustrated in Figure 3.

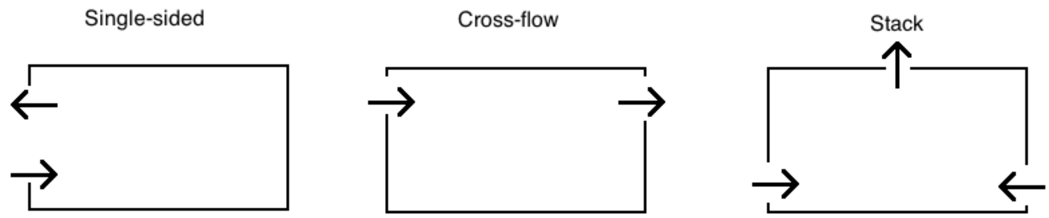


Figure 3: Natural ventilation principles

Single-sided

This ventilation principle is based on having one or more openings on only one side of the room. The main driving force is thermal buoyancy in winter and wind turbulence in summer. Compared to cross-flow and stack ventilation, lower ventilation rates are generated. The ventilation air does not penetrate far into the space. (Heiselberg, 2008).

Cross-flow

This ventilation principle is based on having ventilation openings on two or more sides of the room. The main driving force is wind-induced pressure differentials between the openings. High ventilation airflow rates can be achieved. However, because of large and rapid variations in wind flows, it is difficult to control. Greater room depths can be ventilated using this principle, as the air is crossing the room. (Heiselberg, 2008).

Stack

When ventilation openings are placed at both low and high levels, stack ventilation can be utilized. The main driving force is thermal buoyancy. High and steady ventilation flow rates can be achieved at moderate temperature differences. Larger room depths can be ventilated if the ventilation air is crossing the room. (Heiselberg, 2008).

Mechanically controlled windows

A mechanically controlled window system operates windows based on set preferences for indoor temperature and/or CO_2-level. The control system regulates the window openings based on these pre-set preferences, outdoor temperature, rain and/or wind speed measurements. The system consists of sensors, communication units, window monitoring equipment and motors for opening or closing windows. A mechanical control system usually allows for the occupants to manually operate the windows. It then returns to the automatic mode after a specified period of time. (Windowmaster, 2015).

A mechanically controlled window system has the advantage of securing the functionality of the natural ventilation system independently of the occupants presence and behavior. This can result in better thermal comfort and increased utilization of natural ventilative cooling. Hence, reducing electricity use for fans. However, using mechanically controlled windows in a natural or mixed-mode ventilation system requires a more complex system.

More design time and increased installation and operational costs are likely. It is important that the control system functions properly at all times. The occupants should therefore be trained in use of the system. In addition, increased maintenance might be required. A mechanical system is also more likely to come in conflict with the occupants individual preferences than a manually operated system. (Thomsen et al., 2005).

2.2.2 Mechanical balanced ventilation

In a mechanical ventilation system, electrical fans are used to create the driving force for the air through the building. This type of ventilation is independent of the relation between the indoor and outdoor conditions. Mechanical ventilation can be designed as a supply system, exhaust system or a balanced system. When using mechanical balanced ventilation, fans are used to both supply and extract air from the enclosed space. Air is typically supplied to the most occupied zones, and extracted from the most polluted zones. (Liddament, 1996).

The use of both supply and extract fans in a mechanical balanced system allows proper balancing of pressure and better control of the airflow pattern. (Awbi, 2008). Another advantage is the ability to target the supply air to the zones where fresh air is needed the most and extract air from the most polluted zones. In addition, the absence of high suction pressures reduces the risk of backdraughting and entry of radon or soil gas. Also, such a system allows for filtration of the incoming air, pre-heating and air-to-air heat recovery. In a balanced system, it is possible to customize the ventilation to the users needs to a greater extent than when using only supply or extract ventilation. However, since there are two systems present, the installation and operational costs are usually high. Also, long term maintenance is necessary. If a mechanical balanced system is to operate correctly, the system must be installed in air-tight enclosures. This reduces the safety margins if the system fails to operate properly or if high polluting sources are introduced into the building. (Liddament, 1986).

2.2.3 Mixed-mode ventilation

Mixed-mode ventilation is a type of hybrid system. In hybrid ventilation, mechanical and natural driving forces are combined in a two-mode system. The goal is to maximize comfort and avoid significant energy use and operating costs. (Liddament, 1986). A mixed-mode system uses a combination of natural ventilation from openable windows and fans. Hence, the airflows provided by the mechanical and natural system use different pathways through the building. Natural ventilation is used when it is feasible or desirable and the mechanical system when it is necessary and when heat recovery is needed. (UC:Berkeley, 2014).

A well designed and operated mixed-mode system can reduce the use of mechanical cooling and ventilation. Hence, reducing the electricity use and operating costs. It offers the occupants a higher degree of personal control over the thermal conditions, ventilation conditions and connection to the outdoors. This could lead to increased occupant satisfaction.

These systems are flexible. They can make the mechanical system periodically redundant, which can result in longer system lifetime and reduced lifecycle costs. However, these systems also have the potential to add cost and complexity to the building. There is less familiarity with these types of systems and more design time is needed. In addition, there is the potential of wasted energy if the mechanical and natural ventilation occur in conflict with one another. Also, natural ventilation may be undesirable in some situations due to air-borne pollutants, allergens or outdoor noise. (UC:Berkeley, 2014).

UC:Berkeley (2014) classifies mixed-mode ventilation in terms of whether natural and mechanical ventilation exists in the same space or operate at the same time. A mixed-mode system can either be designed as a concurrent, change-over or zoned system.

Concurrent
In a concurrent mixed-mode system, the mechanical ventilation and the openable windows operate in the same space and at the same time. Windows are the main ventilation operator, while the mechanical system is used as supplement or background ventilation. (UC:Berkeley, 2014).

Change-over
In a change-over mixed-mode system, the building alternates between natural and mechanical ventilation. The operating mode can be determined based on outdoor temperature, occupancy, window opening or other operating commands. (UC:Berkeley, 2014).

Zoned
In a zoned system, the building is divided into different zones. Each zone can be assigned with different strategies. This solution allows use of mechanical and natural ventilation in different parts of the building at the same time. (UC:Berkeley, 2014).

2.2.4 The concept of ventilative cooling

There are two main ways of removing heat surplus from a building; with direct cooling or ventilative cooling. (Nilsson, 2003). While direct cooling uses radiant technology or fan-coil units directly inside the room, ventilative cooling uses ventilation air to cool indoor spaces. (Venticool, 2012).

Ventilative cooling can be an attractive and energy efficient solution to avoid overheating in buildings. Ventilation is already present in most buildings through mechanical and/or natural ventilation systems. It can both remove excess heat gains as well as increase air velocities and shift the thermal comfort range. (Venticool, 2012). Also, cooling by ventilation can satisfy both the requirements for indoor air quality and temperature simultaneously. (Nilsson, 2003). However, if large airflow rates are needed, this method could increase the risk of noise and draught. (Dreau and Heiselberg, 2014).

There are two different methods for ventilative cooling; thermal storage ventilative cooling and direct ventilative cooling.

Thermal storage ventilative cooling
When using thermal storage ventilative cooling, the cool ventilation air is supplied to the building primarily during non-cooling periods to reduce the temperature of the building. (Fustel et al., 1992). The objective is to use the thermal mass of the building as an intermediate storage medium which will cool the building when the temperatures rise. (Santanamouris et al., 1998).

Direct ventilative cooling
With direct ventilative cooling the ventilation air is supplied at the time when cooling is needed. This technique has three objectives: cooling the indoor air, cooling the structure of the building and a direct cooling effect over the human body through convection and evaporation. (Santanamouris et al., 1998).

2.3 Literature survey on ventilative cooling

The literature survey focused on the motives behind implementation of ventilate cooling, the advantages of ventilative cooling, previous evaluations of the performance of ventilative cooling and possibilities of improving ventilative cooling.

2.3.1 End-user satisfaction in well insulated dwellings

Previous research have documented that passive houses can be a good solution for achieving indoor comfort and low energy consumption. (Knudstrup et al., 2009). However, according to both quantitative and qualitative assessments, the indoor temperature can in some cases rise above comfort limits during summer. (Oropeza-Perez and Østergaard, 2014). In this context, several studies have been conducted on end-user satisfaction in passive houses and low-energy dwellings.

End-user satisfaction in passive house dwellings in five European countries

Feist et al. (2005) summarized the results of the EU project Cost Efficient Passive Houses as European Standards (CEPHEUS). Within this project, 221 housing units complying with the Passive House standard were built in five European countries. All CEPHEUS projects were equipped with balanced mechanical ventilation with heat recovery. Half of them also had ground heat exchangers. The mean indoor temperatures in summer for four of the houses were presented in the paper, varying from about 21^{o}C to 27^{o}C. Social science surveys conducted amongst occupants were also presented. They revealed that thermal comfort was reported to be good or very good. 88% of the participants in the survey were very pleased with the indoor climate in summer. The paper concluded that the buildings provided comfortable indoor environments. (Feist et al., 2005).

It should be noted that summer temperatures for the remaining buildings were not presented in the paper. Also, little was mentioned about how the 12% of participants that were not very pleased with the indoor climate perceived the thermal environment.

End-user satisfaction in low-energy dwellings in Germany, Austria, Switzerland and the Netherlands

Mlecnik et al. (2012) analyzed post-occupancy evaluations on nearly zero energy dwellings in Germany, Austria, Switzerland and the Netherlands. The research results from Germany, Austria and Switzerland revealed that these nearly zero-energy houses were appreciated by the residents. However, the occupants often felt more comfortable during the winter than in summer. In the Netherlands a questionnaire revealed that 97% of the users were satisfied with their house. On questions regarding thermal comfort in the summer 7% indicated dissatisfaction in the living room and 16% in the bedroom. 34% experienced high indoor temperatures sometimes in the living room and 49% found the bedroom too hot sometimes during summer. (Mlecnik et al., 2012).

End-user satisfaction in passive house dwellings in Gumslov, Sweden
Samuelsson and Luddeckens (2009) conducted a similar survey on three different passive houses in Frillesås in Gumslov, Sweden. The survey contained questions about experienced temperature, temperature variations and perceived indoor climate. This research revealed that the residents were generally not satisfied with the indoor climate. In one of the houses in particular, more than 50% of the residents reported that it was too hot in the summer. They also complained that they could not adjust the temperature. (Samuelsson and Luddeckens, 2009).

End-user satisfaction in a low-energy housing complex in Stjørdal, Norway
Kleiven (2007) conducted a user-evaluation of the Husby Amfi building in Stjørdal, Norway. The building was a low-energy housing complex with 56 flats. The evaluation contained questions on comfort both in winter and summer. The results from the study revealed that the occupants were mostly satisfied with the building. Thermal comfort was very high, both in winter and in summer. However, most of the residents reported that the building got too hot on the warmest summer days. (Kleiven, 2007).

The studies presented here show that the end-user satisfaction in low-energy dwellings varies between the different projects. The reason for this could be explained by climate variations, different building design and also individual expectations and preferences. However, some of the research revealed that overheating in low-energy dwellings during summer was an issue. It should therefore be addressed when designing low-energy dwellings.

2.3.2 Advantages of ventilative cooling

Previous studies have investigated the range of thermal comfort when utilizing natural ventilation and the effects of increased air velocity.

Thermal evaluation of naturally ventilated buildings and buildings with HVAC systems
de Dear and Brager (2002) summarized earlier research on the adaptive comfort standard (ACS). The starting point was the project ASHRAE RP-884. The RP-884 database contained approximately 21,000 sets of raw data from 160 different office buildings located on four different continents. The data included thermal questionnaire responses, clothing and metabolic estimates, indoor climate measurements, calculated thermal indices and outdoor meteorological observations.

The office buildings in the database were separated into naturally ventilated buildings and buildings with HVAC systems. The naturally ventilated buildings had no mechanical air-conditioning, but openable windows directly controlled by the occupants. Occupants of the HVAC buildings had little or no control over their thermal environment.

The paper presented regression graphs of indoor comfort temperature against the mean outdoor air temperature. The results revealed that the naturally ventilated buildings had a steeper gradient of observed responses compared to HVAC buildings. Hence, indicating that occupants of HVAC buildings become more adapted to the narrow, constant conditions typically provided by mechanical conditioning. Occupants of naturally ventilated buildings preferred a wider range of conditions that more closely reflected the outdoor climate.

The results also revealed that the PMV index was very successful at predicting comfort temperatures in HVAC buildings. However, in the naturally ventilated buildings there was a bigger difference between PMV and the actually responses. The paper suggested that the indoor comfort temperatures in the naturally ventilated buildings were strongly influenced by shifting thermal expectations. This was due to higher levels of perceived control and a greater diversity of thermal experience. (de Dear and Brager, 2002).

This study was conducted on office buildings and not in dwellings. However, the research shows that occupants of naturally ventilated buildings have a broader range of temperatures they perceive as comfortable. Utilizing ventilative cooling from openable windows in dwellings could therefore result in the occupants being comfortable at higher operative indoor temperatures.

Investigating the effects of increased air velocities and personal control on thermal comfort

Cattarin et al. (2012) conducted a climatic chamber study to examine the achievable thermal comfort of traditional bladed less desk fans. 32 Scandinavians performing office activities and wearing light clothes were exposed to increased air movement generated by a personal desk fan. The subjects were exposed to three fixed environment conditions with operative temperatures equal to 26^{o}C, 28^{o}C and 30^{o}C. Relative humidity was in the range of 40-50%. After an adaptation time, the subjects were invited to adjust the air movement for achieving their preferred thermal comfort. The individual preferred air velocities were recorded.

The results revealed a tendency towards higher air speeds at increasing air temperatures. The study therefore concluded that higher air velocity under personal control make the indoor environment acceptable at higher air temperatures. (Cattarin et al., 2012).

This study was conducted on desk fans and not on general room ventilation. However, the results show that increased air velocities can compensate for high indoor temperatures. Utilizing ventilative cooling with high air velocities could therefore result in the occupants being comfortable at higher operative indoor temperatures.

2.3.3 Performance of ventilative cooling in low-energy dwellings

Various strategies of ventilative cooling in low energy dwellings have previously been evaluated in terms of indoor air quality, energy saving potential and thermal comfort.

Indoor air quality evaluation and energy saving potential of natural ventilation in an ecological house in Tapanila, Finland

Simonson (2005) conducted a study to investigate the energy consumption and ventilation performance of a naturally ventilated ecological house. The current building was a two-story, single-family dwelling located in Tapanila district of Helsinki, Finland. It was constructed with high use of natural and ecological materials and it was well-insulated. It had no plastic vapor retarder which permitted diffusion mass transfer through the porous building envelope. The building utilized natural ventilation. To investigate the ventilation performance, the buildings CO_2-level was measured. The simulation program WinEtana was used to analyze energy consumption. Simulations were conducted using a model of the building as it was, and with an alternative system solution utilizing mechanical ventilation with 50% heat recovery.

Measurements revealed that the ventilation rate was seldom below 4L/s per person in the bedrooms and generally above 0.5 ach outside the bedrooms. The paper therefore concluded that the ventilation rate was adequate and that the indoor air quality was good. Since the occupants often utilized natural ventilation through open windows, the measured indoor concentration of CO_2 was similar in summer and winter. The results showed that the natural ventilation system increased space heating by 22% due to lack of heat recovery. It decreased electrical energy by 14% because there were no ventilation fans. The primary energy consumption was nearly the same for both systems. (Simonson, 2005).

It should be noted that higher heat recovery rates than the 50% used in this study are available. Also, this study is not directly comparable to regular low-energy dwellings because the building was constructed with special materials and did not have a plastic vapor retarder. However, it demonstrates that use of natural ventilation could lead to increased space heating and decreased electricity-use compared to use of mechanical ventilation.

Thermal evaluation and end-user satisfaction in low-energy dwellings with passive cooling strategies in Belgium, Netherlands, Canada and Denmark

Thomsen et al. (2005) published a paper presenting the results obtained from measurements and interviews from occupants in 12 advanced solar low-energy houses utilizing ventilative cooling. Four of these projects provided results concerning thermal comfort. These buildings were located in Belgium, Netherlands, Canada and Denmark.

For the Belgian house, the results from the measurements showed that comfortable indoor conditions were provided even during periods of extremely warm weather. This was due to the application of solar control and nighttime ventilation. The mean indoor temperature remained below 25^{o}C even though the outdoor temperature reached 30^{o}C. The CO_2 concentration was never above the Belgian recommendations of 1500ppm. Results from the interviews revealed that the end-users in this house displayed general satisfaction with the indoor climate.

The results from the measurements in the Netherlands revealed that temperatures in rooms facing south hardly fluctuated. Not even during days with intensive solar radiation. This was due to the properties of the building mass, solar shading and the functionality of the passive cooling system. In summer, the indoor temperature in the bedroom was around 25^{o}C even though the outdoor temperature reached 35^{o}C. The interviews revealed that the end-users found the indoor climate quite good, but the control system was found to be too complex.

Results from the Canadian house revealed that the increased ventilation rate depended on the occupants being home to open windows. The study concluded that an automatic window opening system would have led to better performance.

Measurements of the house in Denmark displayed far higher indoor temperatures than predicted. The main reason was that it was difficult to create cross-flow ventilation. Furthermore, the increased ventilation rate was dependent on windows being manually opened. Also, solar shading was lacking in front of a large south-oriented window. Results from the interviews revealed that the end-users found the overheating problems very serious. They complained that overheating occurred as soon as the sun was shining. The occupants even moved out because of the problem with high indoor temperatures. (Thomsen et al., 2005).

The evaluation of the thermal performance of these buildings relied on several cooling strategies. The performance of the ventilative cooling alone was not determined. However, this paper shows that preventing overheating is possible, with the right implementation of ventilative cooling. It also shows the importance of automating the natural ventilation strategies.

Thermal evaluation of nighttime ventilative cooling in a passive house dwelling in Limbus, Slovenia

Mlakar and Strancar (2011) investigated overheating in a single family passive house in Limbus, Slovenia. Overheating was characterized by comparing simulated and measured internal temperatures during summer months. The building was analyzed with and without strict shading and night-ventilation. The results showed that strict shading during the day and excessive ventilation through manually opened windows during the night could keep the internal temperatures within the comfort level. Not applying these two strategies, would lead to extreme overheating. (Mlakar and Strancar, 2011).

When strategies to prevent overheating are manually operated, temperatures as comfortable as obtained in this study might not occur if the occupants are not well acquainted with the system. Mlakar and Strancar emphasize themselves the importance of teaching the residents how to use the system. However, this paper shows that applying the right passive strategies can eliminate the problem of overheating.

Energy saving potential and thermal evaluation of natural ventilation in a passive house in Denmark

Oropeza-Perez and Østergaard (2014) conducted a case study to investigate the performance of natural ventilation as a passive cooling method. Indoor temperatures and energy saving potential were analyzed for a $112.2m^2$ passive house in Vejle, southeast Jutland, in Denmark. Simulations were conducted with the thermal-airflow program EnergyPlus for the months of June, July and August. The simulations were validated with measured data.

The results showed that out of the three months studied, only 85 hours had uncomfortable temperatures when using natural ventilation in combination with mechanical ventilation. This was a reduction of 90.4% compared to the measurements of the indoor temperature when not using natural ventilation.

Energy savings up to 42 kWh were obtained during the cooling season, avoiding 839 hours of electric fan use. Compared to average electricity demand for Danish dwellings, this resulted in an energy saving of 37.5%. (Oropeza-Perez and Østergaard, 2014).

It should be noted that the percentage of energy reduction was based on the average electricity demand in Danish dwellings. Hence, the energy reduction of 37.5% was not the direct reduction in the specific building. However, the study shows that utilizing natural ventilation in combination with mechanical ventilation can significantly reduce both the number of uncomfortable days and the energy use.

Evaluating thermal comfort of ventilative cooling in an Active House dwelling in Denmark

Fjoldberg et al. (2011) investigated thermal comfort in the active house Home For Life in Denmark. The study focused on the role of solar shading and natural ventilation. The dwelling was a $1\frac{1}{2}$ story house with a floor area of $190m^2$. It followed the Active House principle, which means a balanced priority of energy use, indoor environment and connection to the external environment. The goal was for the dwelling to have very low use of energy and an excellent indoor environment. There was a particular focus on good daylight conditions and fresh air from natural ventilation. The house was ventilated by a hybrid system. Natural ventilation was used during summer and mechanical ventilation with heat recovery during winter. Hybrid ventilation was used in spring and fall. In both natural and mechanical mode, the ventilation rate was demand-controlled with CO_2 and humidity indicators. External automatic solar shading was applied on all south directed windows and overhangs were used where appropriate. The occupants could overrule the automatic controlled ventilation and solar shading at any time. Measurements were performed and the results were compared to simulations.

The results revealed that during summer, windows were almost permanently open between 09.00 and 22.00. There were also many episodes with open windows between 22.00 and 09.00, which was assumed to be caused by automatic window opening for night cooling. Windows were generally closed when outdoor temperatures were below 10^oC. When outdoor temperatures were above 12^{o}C, windows were frequently open when the indoor temperature exceeded 22-23^{o}C. The results also showed that there was very limited overheating during summer. Only a few episodes with temperatures above the maximum adaptive temperature given in category 1 of NS-EN-15251 occurred.

The study concluded that a clear correlation between window openings and acceptable thermal comfort was found, indicating that ventilative cooling from open windows was particularly important to maintain good thermal conditions. The paper also concluded that the dwelling achieved good thermal environment in real use. (Fjoldberg et al., 2011).

This study was conducted in an Active House, which differs slightly from Zero Energy Buildings. However, the strategies used to prevent overheating are similar. This study is therefore relevant for the current assignment and it shows that ventilative cooling from open windows can contribute to good thermal comfort.

Evaluating thermal comfort and energy saving potential of ventilative cooling in dwellings in Athens, Rome, Berlin and Copenhagen

Pellegrini et al. (2012) conducted a study on daytime comfort ventilation and night cooling in domestic buildings. Ten different ventilation and cooling strategies were simulated for a $1\frac{1}{2}$ story, single family house of $175m^2$ in Athens, Rome, Berlin and Copenhagen. Simulations were conducted with the IDA ICE based software EIC Visualizer. Thermal comfort and indoor air quality in the summer was evaluated.

The results showed that significantly increasing air velocities during daytime and frequent use of night cooling achieved very good thermal comfort in Athens. For Rome, Berlin and Copenhagen this combination caused overcooling. In Rome and Berlin limited increase in air velocity during the day and use of night cooling provided very good thermal conditions. In Copenhagen the best performance was obtained with the use of night cooling only.

In Athens and Rome, utilizing passive cooling techniques lead to a consistent reduction of energy consumption. 83% and 65% was reported for Athens and Rome, respectively. Implementation of ventilative cooling in Berlin and Copenhagen reduced the energy demand by 5.6% and 1.3%, respectively. In Copenhagen that meant reducing the cooling load to zero.

Natural ventilation provided better indoor air quality than mechanical ventilation in all cases. The study therefore concluded that natural ventilation had the best overall performance. (Pellegrini et al., 2012).

2.3.4 Proposed solutions to improve ventilative cooling

Previous studies have evaluated proposed solutions for improving ventilative cooling.

Thermal evaluation of a PCM air heat exchanger in a passive house dwelling in Sweden

Persson and Westermark (2012) investigated the potential of providing space cooling to a Swedish Passive House through the use of a phase change material (PCM) air heat exchanger. The evaluation was performed using Matlab code and simulations in IDA ICE from June to August. The building model was based on a 4-room apartment in a passive house building in Lambohov in Linkoping, Sweden. The apartment had a CAV ventilation system. The PCM air exchanger was placed in an insulated box on the outside of the wall where the supply air entered the building. During the night, the PCM in the storage cooled down and solidified. If the outdoor temperatures rose over the transition temperature the following day, the PCM would melt and cool the air flowing through the storage. The supply air entering the building could be connected to or disconnected from the PCM depending on the need for cooling.

The simulations demonstrated a substantial removal of excessive indoor temperatures when implementing PCM night cool storage. However, the overheating could not be completely eliminated on the warmest days. When using 50-400kg of PCM in the cool storage, the reduction varied between 22% and 36% of the total 2500 degree hours over 26^{o}C in the reference case. (Persson and Westermark, 2012).

This study shows improvement on the indoor temperatures during summer when implementing PCM night cool storage. However, the current paper did not evaluate the effect on fan power. Since the PCM was placed at the supply inlet, the use of fan power is likely to have increased.

Evaluating cooling provided by a ground culvert in Norway

Zinzi and Citterio (2010) investigated the cooling effect from a ground culvert connected to the air intake tower of a primary school in Norway. The culvert was a 20m long ground-coupled duct made of concrete. It had a diameter of 1.6m. A fan was installed in the duct to provide additional pressure. Air temperatures, surface temperatures and airflows were monitored for two years.

The results showed that the buried duct had a significant cooling effect. Conservative calculations showed that the duct provided 4 kW of cooling with an outdoor temperature of 18^{o}C at $0.9m^3/s$. (Zinzi and Citterio, 2010).

This study shows that applying a ground culvert can provide significant cooling. However, the costs and greenhouse gas emissions required to build a concrete ground culvert was not assessed. Neither was the potential problem of moisture development and fouling.

2.3.5 Summary of main findings from literature

This literature survey revealed that overheating has been an issue in some of the previous studied low-energy dwellings, however not in all. Overheating should therefore be studied in Living Lab to see if it is likely to be an issue for this particular low-energy dwelling.

Previous research have revealed that preventing overheating in low-energy dwellings is possible with the right implementation of passive solutions. Ventilative cooling from open windows has proven to be able to provide good thermal comfort. The study of Living Lab should examine whether ventilative cooling is able to achieve sufficient thermal comfort for this building also.

Previous studies have also revealed that increased air velocities can compensate for high indoor temperatures. The study of the effect of ventilative cooling on thermal comfort in Living Lab should therefore examine the combined effect of indoor air temperatures and increased air velocities.

One study revealed that increased air velocities during daytime and frequent use of night cooling achieved good thermal comfort in some cases. It did, however, result in overcooling in other cases. The study of thermal comfort in Living Lab should therefore address the issue of overcooling. Also, it should determine whether applying ventilative cooling day and night is necessary to prevent overheating, or if it will result in overcooling. If it does result in overcooling, the study should determine if the best option would be to utilize daytime or nighttime ventilative cooling.

One of the studies reviewed in this literature survey revealed that manually operated ventilative cooling provided sufficient thermal comfort. However, other studies have concluded that automating passive cooling strategies was crucial for achieving the desired effect. The study of how to apply ventilative cooling in Living Lab should therefore evaluate whether an automatic window control system is necessary.

One of the previous studies concluded that natural ventilation in combination with mechanical ventilation significantly reduced the energy use of the building. Another study revealed that use of natural ventilation lead to decreased electricity use but increased space heating compared to use of mechanical ventilation. A third study showed that utilizing passive cooling techniques lead to a consistent reduction in energy consumption. The study of Living Lab should therefore determine the effect off ventilative cooling on energy use. It should also distinguish between energy use for heating and electricity to enable a comparison to previous findings.

Lastly, the literature survey revealed improvement on the indoor temperature during summer when implementing PCM night cool storage. It also revealed that applying a ground culvert could provide significant cooling. If the current study of Living Lab finds ventilative cooling insufficient, these actions could be recommended.

3 Preparing simulations

Considering Living Lab is still under construction, the study of ventilative cooling had to be conducted using simulations only. This section includes argumentation for the chosen simulation tool. It also verifies relevant parts of the model, to support the choice and to validate the final results. Before testing different solutions for ventilative cooling, a base model of Living Lab was constructed and a set of criteria for evaluation of the different system solutions determined. The following section describes how this base model was built and presents argumentation for the chosen evaluation criteria.

3.1 Simulation software

3.1.1 Choice of simulation software

The choice of simulation software was conducted during the project work. IDA ICE, Simien, Contam w, EnergyPlus, TRNSYS and ESP-r were assessed. The thesis concluded that where Simien was considered to be a too simple simulation tool, both EnergyPlus, TRNSYS and ESPR-r had a user interface too complex for this assignment. Both Contam and IDA ICE could have been used for the current task. However, IDA ICE having a thermal part, and being able to simulate energy flows, made it a better choice. IDA ICE is also more frequently used in engineering work in real life. IDA ICE having a range of possibilities and still being designed for engineers who are not simulation experts, makes it a suitable program. (Kirkøen, 2014). For the complete evaluation of the different simulations tools, the reader is directed to the project thesis.

3.1.2 Verifying parts of the IDA ICE model

When evaluating ventilative cooling, the most important aspects are how the airflow through open windows and the progress of changing indoor air temperature are modelled. These aspects of the IDA ICE model were therefore verified. The verification was based on work performed during the specialization project. In this project a simple model of a building, with only two openable windows, was used to study natural ventilation through openings. Well-established mathematical formulas were used to create a calculation procedure for the airflow through these windows and the changing indoor air temperature. Calculations were conducted for various outdoor summer conditions. The same simplified model was then built in IDA ICE and simulations were conducted using the same set of outdoor conditions. Similar results regarding indoor air temperatures were obtained, hence verifying that part of the model. However, results concerning airflow through open windows were not obtained from IDA ICE. Additional simulations were therefore necessary in order to complete the verification. A short description of the analysis conducted in the previous work can be found in the Appendix Section A.2. For a full description of the study, the reader is directed to the project thesis. (Kirkøen, 2014).

Verifying the progress of changing indoor air temperature

Calculated indoor air temperature and simulated results are presented in Figure 4 and 5 for two different summer days. Some differences between calculated and simulated results can be observed. The biggest difference was found for the starting temperature on the warmest summer day. The explanation was that IDA ICE uses an iterative process to determine the starting temperature, while in the calculations it was set to 21^oC. Other differences were found at low outdoor air temperatures. This was explained with the calculations starting at a temperature of 21^oC and assuming that the temperatures would not drop below this level. While in IDA ICE, heating was not implemented. The conclusion was, nevertheless, that the results were conceding, hence verifying the part of the IDA ICE model treating temperature variations. (Kirkøen, 2014).

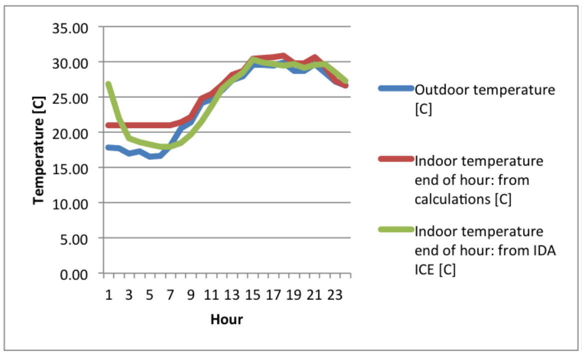


Figure 4: Comparing temperatures: warmest summer day (Kirkøen, 2014)

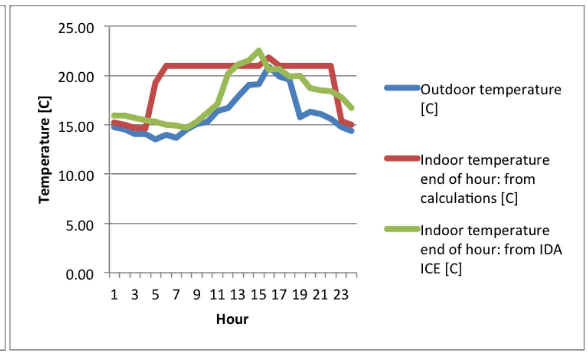


Figure 5: Comparing temperatures: warm summer day (Kirkøen, 2014)

Verifying airflows through open windows

Figure 6 and 7 present the comparisons between calculated airflows and airflows obtained from IDA ICE.

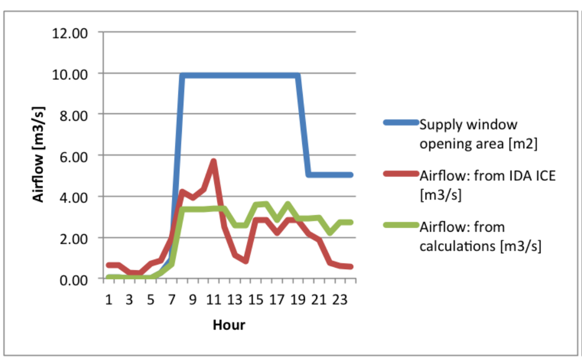


Figure 6: Comparing airflows: warmest summer day

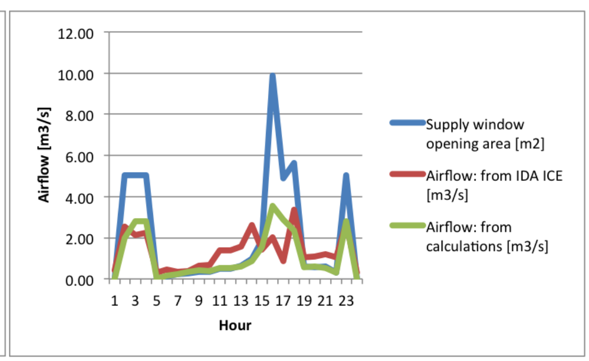


Figure 7: Comparing airflows: warm summer day

It can be seen from the illustrations that the airflow rates obtained were very high. This was due to large available window opening areas, high wind velocity and the use of a low set-point temperature (21^oC) in the building despite high outdoor air temperatures (up to 30^oC). More reasonable airflow rates could have been obtained if the set-point temperature had been set to a more appropriate (higher) level.

The comparisons show some differences. These differences can be explained with IDA ICE using an iterative process to determine the starting conditions. In the calculations, the starting point conditions was set to assumed values. IDA ICE also accounted for airflows due to infiltration although this was omitted from the calculations. Nevertheless, the results show that the airflow pattern was similar and values were in the same magnitude. Hence, verifying the part of the IDA ICE model treating airflows through open windows.

3.2 Modelling Living Lab in IDA ICE

3.2.1 Architectural design

The process of constructing the building body in IDA ICE was based on the architectural drawings of the building. The floor plan and a sectional drawing can be found in the Appendix Section A.3.1 and A.3.2. Necessary additional information was provided by the project architect, Luca Finocchiaro. The complete building model in IDA ICE and the resulting floor plan can be seen in Figure 8 and 9.

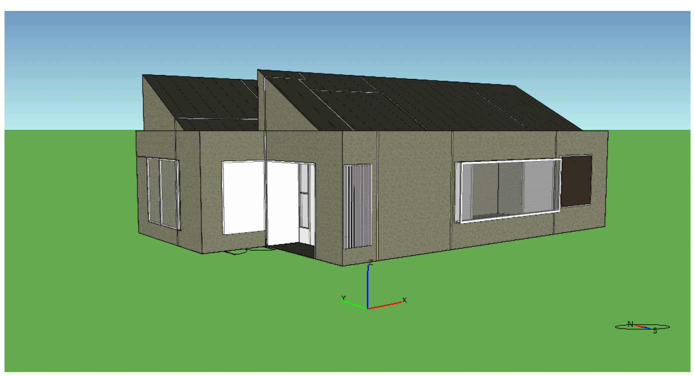


Figure 8: Exterior model illustration

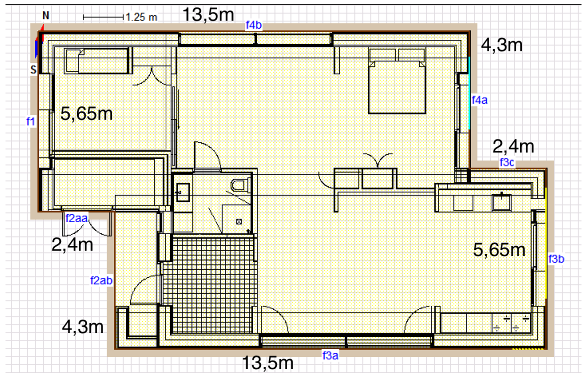


Figure 9: Ground Floor illustration

It can be seen from Figure 9 that the entrance area outside of the building is included in the building body. This was necessary in order to model the complex roof design. However, this part of the building was assigned an independent zone, so it did not affect the rest of the building.

3.2.2 Construction

The building elements were implemented according to information provided by the projects architect, Luca Finocchiaro. The construction of the different building parts are presented in Table 1 and 2. (Finocchiaro, 2015).

Table 1: Materials used for the roof and wall constructions. From top to bottom: interior to exterior.

Roof		External wall		Internal wall	
Material	Thickness [m]	Material	Thickness [m]	Material	Thickness [m]
Plywood	0.016	Plywood	0.012	Plywood	0.015
Insulation	0.05	Insulation	0.048	Rockwool	0.035
Vapour barrier	0.0004	Vapor barrier	0.0004	Plywood	0.015
Rockwool	0.2	Rockwool	0.15		
Rockwool	0.2	Paper	0.0004		
Water proof barrier	0.0004	Rockwool	0.2		
Air gap before PV	0.058	Wind barrier	0.0004		
		Airgap	0.06		
		Cladding	0.022		

Table 2: Materials used for the floor construction. From top to bottom: interior to exterior.

External floor		Internal floor (-bathroom)		Internal floor bathroom	
Material	Thickness [m]	Material	Thickness [m]	Material	Thickness [m]
Wood HDF	0.022	Mass pinewood	0.022m	Tiles	0.015
Rockwool	0.4			Concrete	0.03
Wind barrier	0.0004				
Airgap	0.048				
Cladding	0.022				

The model in IDA ICE assumed that the external wall around the entrance door was an internal wall due to the outdoor entrance part being modeled as a part of the building body. To avoid this, an additional type of internal wall was defined, having the same values as an external wall, and applied to this part of the building.

Typical values for heat conductivity, density and specific heat for the different materials used were collected from Engineering Toolbox (2015). These values can be found in the Appendix Section A.4.1. The values obtained for heat conductivity had to be slightly adjusted to acquire the correct U-values for the building.

The resulting U-values for the different building elements were then in compliance with the real building. They are presented in Table 3. Thermal bridges were set to 0.03 and infiltration to 0.5.

Table 3: U-values of construction elements

U-values construction elements	
Building element	U-value [W/m^{2}K]
Exterior wall	0.11
Roof	0.10
Floor	0.10

3.2.3 Zones

It was decided to apply one individual zone for each room to obtain the desired accuracy and to enable evaluation of thermal comfort in each room. The partitioning of zones in IDA ICE are presented in Figure 10 and 11. The resulting total area for each of the zones can be found in the Appendix Section A.4.2.

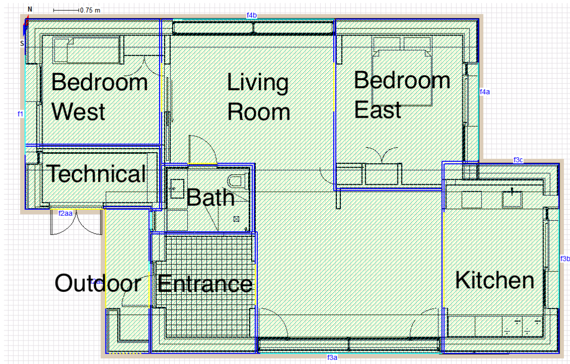


Figure 10: Zones level 1

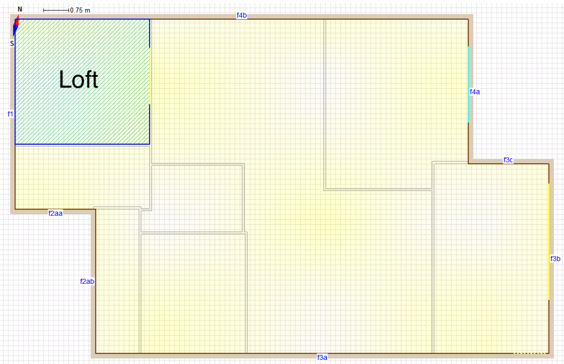


Figure 11: Sones level 2

Set-point temperatures and ventilation airflows were defined for each zone. The building is most likely to utilize only mechanical ventilation in winter. It would therefore not be sufficient to use the thermal criteria for a building without fans installed. The set-point temperature were therefore implemented according to values obtained from NS-EN-15251 (2007) for a building with mechanical ventilation and occupant satisfactory level two. Hence, the range of acceptable temperatures were 20-26^{o}C.

Ventilation in Living Lab is designed as a balanced mechanical system. Supply jets are located in the living room and in the bedrooms, exhaust in the bathroom and kitchen. It integrates a heat recovery unit with efficiency of 85% at the nominal value, and an additional electric coil capable of warming up the inlet air up to 40^oC. (Finocchiaro et al., 2014). The ventilation unit is a Flexit UNI 3 designed to operate with constant air volumes (CAV) on three levels. Level 1 is designed to cover the ventilation requirements when the building is not occupied. Level 2 will mainly be used and represents hygienic mechanical ventilation, supplying and extracting $144m^3/h$. Level 3 represents enhanced mechanical ventilation and will be used when additional airflow rates are needed. This level has the ability to supply and extract up to $260m^3/h$. (Mathisen, 2015).

The hygienic mechanical ventilation level was implemented in the base model of Living Lab. Supply and extract units and corresponding airflows were implemented according the ventilation design, which can be found in the Appendix Section A.3.3. The implemented values for each zone are presented in Table 4.

Table 4: Zone values for temperature and ventilation

	Zone set-points			
Zone	Min temp[^{o}C]	Max temp [^{o}C]	Supply air CAV [L/sm^2]	Return air CAV [L/sm^2]
Outdoor	-50	50	0	0
Entrance	20	26	0	0
Bathroom	20	26	0	5.43
Living room	20	26	0.36	0
Kitchen	20	26	0	0.52
Bedroom East	20	26	0.7	0
Bedroom West	20	26	0.49	0
Loft	20	26	0	0
Technical	20	26	0	0

In Living Lab there are two bedrooms and one loft. It is therefore reasonable to assume that three or four people will occupy the building. To include the worst case scenario with regard to internal heat gains, four occupants were used. Two different occupancy schedules were defined. It was assumed that on a typical weekday the building would be occupied for all hours except between 08.00 and 17.00. It was decided that the weekend pattern would represent a weekend or a holiday where the whole family stayed at home. This decision was made to distinguish between weekdays and weekends and to consider the worst case scenario in terms of internal heat gains.

The number of people present and where they reside throughout the day are presented in Figure 12 for weekdays, and Figure 13 for weekends.

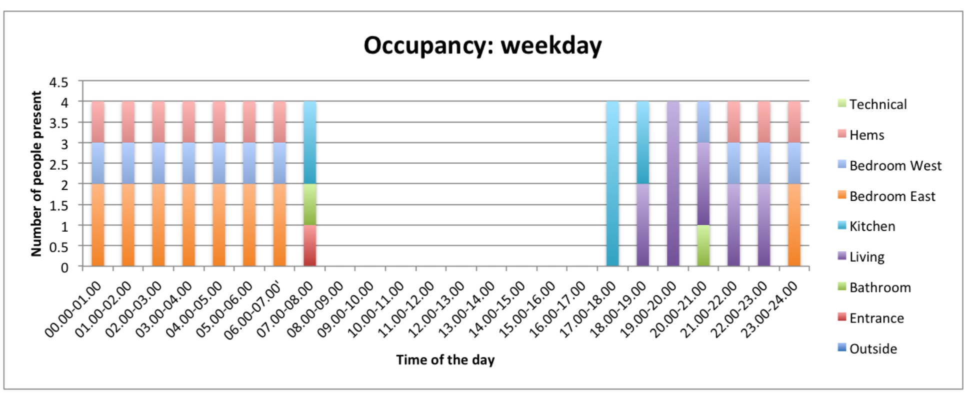


Figure 12: Occupancy schedule on weekdays

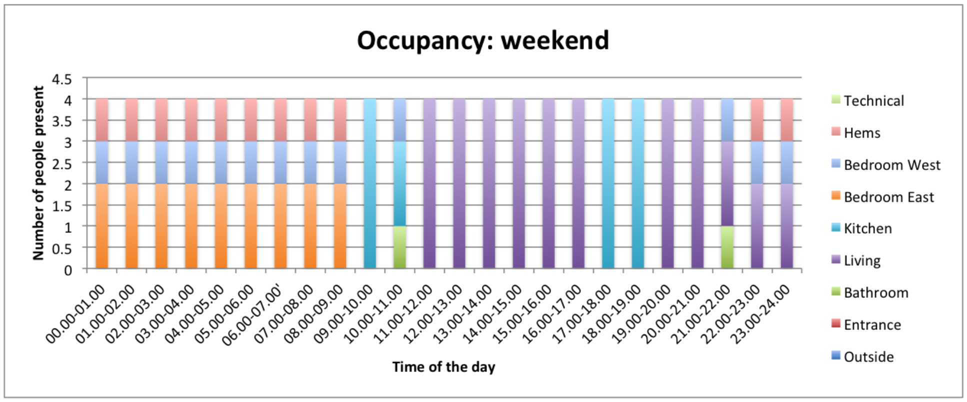


Figure 13: Occupancy schedule on weekends

Lighting and equipment schedules were assumed to follow the presence of the occupants. Hence, lighting and equipment were turned on when the building was occupied. The resulting schedules for lighting and equipment can be found in the Appendix Section A.4.3.

Typical values for met and clo obtained from Novakovic et al. (2007) were used. Real values for internal heat gain due to lighting and equipment in Living Lab were obtained from Finocchiaro (2014). The implemented values for the different zones are presented in Table 5.

Table 5: Zone values for internal heat gains

	Internal gains				
Zone	Max occupancy	Met	Clo	Heat: equipment [W/m^2]	Heat: light [W/m^2]
Outdoor	0	0	0	0	0
Entrance	1	1.5	1	19.85	21.51
Bathroom	1	1.2	0.5	9.94	0.11
Living room	4	1	0.5	91.13	0.99
Kitchen	4	1.2	0.5	34.78	0.38
Bedroom East	2	0.8	0.5	36.90	0.4
Bedroom West	1	0.8	0.5	26.80	0.29
Loft	1	0.8	0.5	26.80	0.29
Technical	0	1.5	0.5	13.30	0.14

3.2.4 Openings, internal and external doors

The process of constructing internal openings was based on the floor plan and a sectional drawing. These drawings can be found in the Appendix Section A.3.1 and A.3.2. Necessary additional information were obtained from observations and measurements at the building site. Figure 14 illustrates the interior of the building model in IDA ICE.

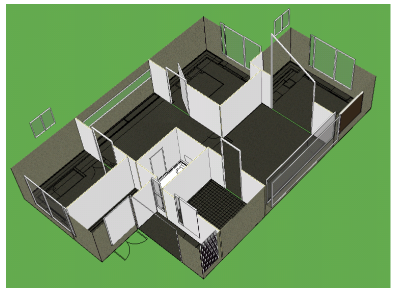


Figure 14: Internal model illustration

The process of constructing the internal and external doors was based on the schematic of the doors, which can be found in the Appendix Section A.3.4. Additional information about their placement was provided by the projects architect Luca Finocchiaro. The resulting U-values for the different doors can be found in the Appendix Section A.4.4.

It was assumed that the exterior doors and the internal doors to the bathroom and technical room would rarely be open. The opening schedule for these doors were set to never open. The bedroom doors were assumed to be half open during daytime (from 07.00 to 23.00).

3.2.5 Windows

Living Lab has openable windows or sliding doors on all facades. Sketches of how the three different types of windows and the sliding doors open are shown in Figure 15. On the north side there is an ablong window. It is has hinges at the top, and opens towards the interior to a maximum angle of 39^oC. (Goia, 2015). On the west and east side there are sliding glass doors. There are also two sets of rooftop skylight tripled glazed windows facing north. They open horizontally to a maximum angle of 30^oC. (Justo-Alonso, 2015).

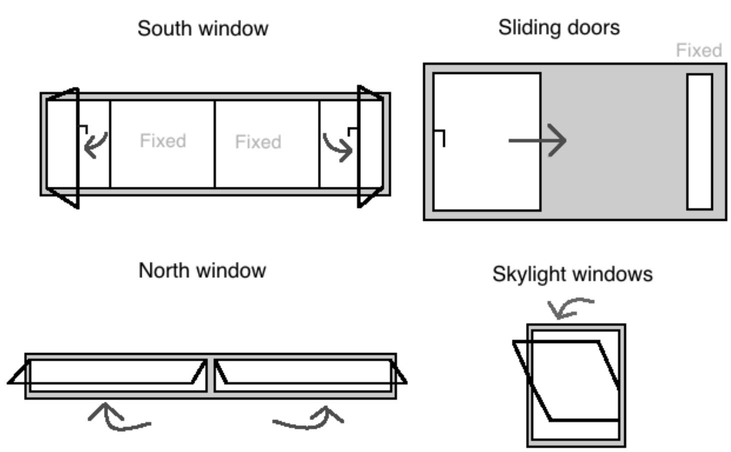


Figure 15: Illustration of the window types in Living Lab

On the south facade there is a large ventilated Scapa window. It has two sets of glass constructions, one single-layered, and the other trippled-layered. This allows for outdoor air to flow from the outside between the glazings and into the building. It is currently constructed with inlets at the bottom of the window. (Finocchiaro, 2015). The inner trippled-layered glazing is divided into four parts. The two outer parts can be opened to a maximum angle of $37^o C$, while the two middle parts are fixed. A venetian blind has been implemented between the outer and inner glazing. (Goia, 2015). The ablong window on the north facade, the large window on the south facade and the skylight windows all have the ability to be mechanically controlled. (Finocchiaro, 2015).

The implementation of windows was based on the window schematic, which can be found in the Appendix Section A.3.5. Information about their placement was provided by the projects architect. The windows as implemented in IDA ICE are presented in Figure 16.

Figure 16: Facade illustrations. Top left: north facade, top right: east facade, bottom left: west facade, bottom right: south facade

Information regarding U-values and g-values for the windows was provided by the projects architect and the window suppliers. Specific window values which could not be obtained from the window suppliers were set to assumed values in cooperation with window specialist Steinar Grynning from SINTEF. It was assumed that the sliding doors in the bedrooms and the skylight windows in the loft would be equipped with curtains which would be drawn at nighttime. Default curtains with g-value of 0.65 and T-value of 0.16 were therefore implemented in IDA ICE for these windows. The resulting inputs for the non-ventilated windows and sliding doors are presented in Table 6.

The sliding doors in Living Lab consists of three parts; one part openable glass door, one part fixed wood section and a small fixed glass window. The three parts were all implemented as separate windows in IDA ICE. The two parts made of glass were assigned values listed under sliding door glass in Table 6. The middle wood part was implemented as a window, but assigned values equal to a wall, see values listed under sliding door wood in Table 6. The complete sliding door have U-value 0.64, hence all three parts of the sliding door were assigned this value.

Table 6: Window data regular windows

	Window data			
	North	Skylight	Sliding door glass	Sliding door wood
Number of type	1	2x2	3	3
Window area [m^2]	3.12	0.66	2.88	3.48
Layers	3	2	3	3
U-value [W/m^2K]	0.87	1	0.64	0.64
g-value	0.52	0.7	0.37	1.00E-09
T, Solar transmittance	0.5	0.69	0.35	1.00E-10
Visible transmittance	0.6	0.81	0.57	1.00E-10
Emissivity	0.84	0.84	0.84	1.00E-10
Solar shading	No	Night (Loft)	Night	No
Shading schedule		23.00-07.00	23.00-07.00	

The large window on the south facade of Living Lab has a unique design. In IDA ICE, this window was defined as a ventilated window. The construction was based on a sectional drawing of the window, which can be found in the Appendix Section A.3.6. Where specific values could not be obtained from the window supplier, assumed values were determined in cooperation with window specialist Steinar Grynning from SINTEF.

It was decided that the solar shading for this window should be drawn whenever solar radiation exceeded 100W/m^2. The resulting ventilated construction, as implemented in IDA ICE, can be seen in Figure 17. Detailed information about the inputs for the window is presented in Table 7.

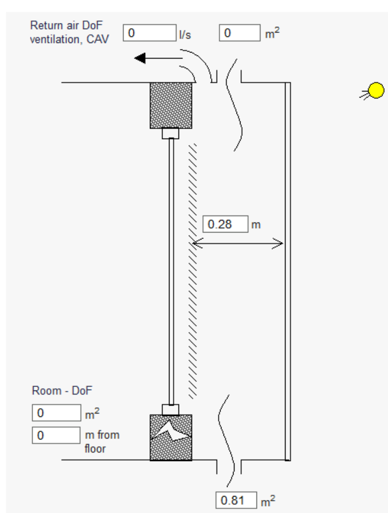


Figure 17: Construction south window

Table 7: Window data south window

	South window details
Inner glass structure	Triple low-e (argon): triple LoE270-6 glass with air and argon mix between layers
Airgap	20mm
Shading	Venetian blind: Slat metal A Spacing: 25mm Slat width: 110mm Slat angle: 80 degrees Shading hole to total area ratio: 0.05m^2/m^2
Shading control	Drawn when radiation exceeds 100W/m^2
Outer glass structure	Single clear
Total window area [m2]	10.91
Total U-value	0.65 W/m^{2}K

3.2.6 Heating and ventilation

Plant

The plant was implemented in IDA ICE with default versions of the main components. The construction was based on the system design drawing, which can be found in the Appendix Section A.3.7. The most important values were changed to be in accordance with the real plant. The resulting plant in IDA ICE can be seen in Figure 18.

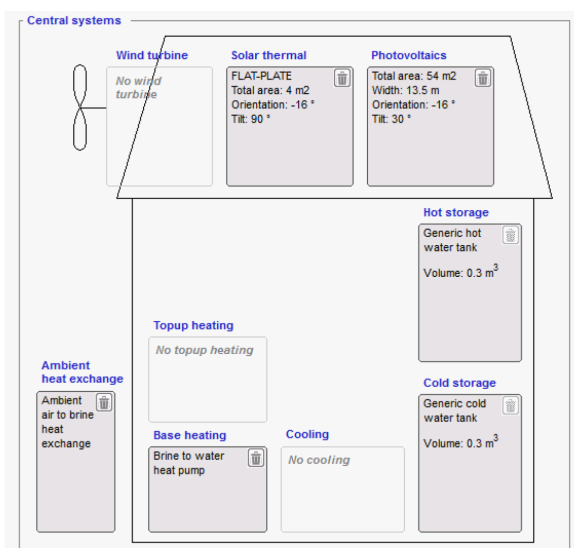


Figure 18: Plant structure

AHU

The process of constructing the Air Handling Unit (AHU) was based on the standard AHU in IDA ICE and the ventilation design drawing, which can be found in the Appendix Section A.3.3. The specific fan power (SFP) was set according to passive house values. (NS-3700, 2013). The supply air was set to a constant temperature of 18^{o}C. (Mathisen, 2015). Since Living Lab is not equipped with any form of cooling, the cooling coil was turned off. Hence, the mechanical ventilation would supply air at a minimum temperature of 18^{o}C. The supply air temperature would only exceed 18^{o}C if outdoor temperatures exceeded this temperature. An illustration of the system in IDA ICE is given in Figure 19. Key input values are listed in Table 8.

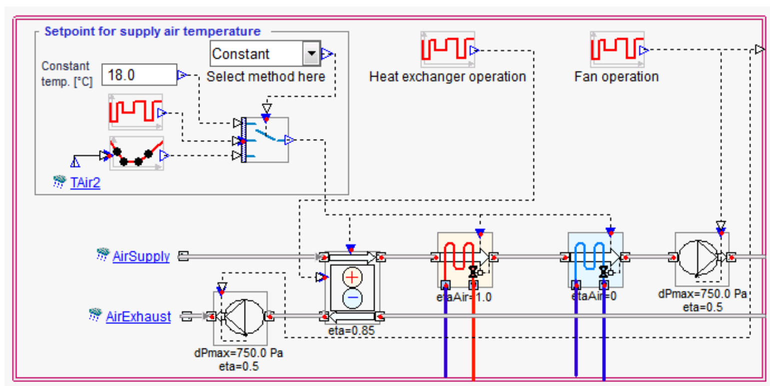


Figure 19: Central air handling unit

Table 8: Details central air handling unit

AHU details	
Supply air temperature	Constant at minimum 18 degrees
SFP	1.5 kW/(m^3/s)
Fan efficiency	0.5
Heat exchanger efficiency	0.85

The fan operation schedule seen in Figure 19 has the ability to change from always on to always off depending on whether mechanical ventilation should be utilized or not.

Heating

Living Lab is equipped with floor heating in occupied areas and a low-temperature radiator in the living room. Three different heating options are available; use of floor heating and radiator heat, use of floor heating only in the bathroom and heat from the radiator or heating by ventilation air. (Mathisen, 2015). Since the main focus of this study is cooling, only one of the heating options were implemented. It was decided to use the option combining floor heating in all occupied areas with the radiator in the living room. This option was assumed to be sufficient to cover the whole heating load. That way, if uncomfortably low temperatures occurred, it would most likely be a result of overcooling due to ventilative cooling. A default radiator and default floor heating were used. Values were changed according to the technical specifications, which can be found in the Appendix Section A.3.8. The floor heating and the radiator were set to be regulated by air temperature so they would not work against the ventilative cooling. Hence, heating would be supplied to prevent indoor temperatures below the minimum temperature of 20^{o}C. Details regarding the floor heating system and the radiator are given in Table 9.

Table 9: Heating systems

Floor heating		Radiator	
Location	Bathroom, entrance, living room, kitchen bedrooms	Location	Living room
Design power [W/m^2]	30	Type	Water radiator
Delta T design power [^{o}C]	5	Mass flow full power [kg/s]	0.12
Controller	Thermostat: air temp	Maximum power [W]	2543
		Supply T max power [^{o}C]	55
		Return T max power [^{o}C]	50
		Controller	PI control: air temp

3.2.7 Building site

In order to obtain the right orientation of the building, measurements using a compass were conducted on the actual building site. An orientation of 16^o was recorded. The resulting orientation of the building in IDA ICE can be seen in Figure 20.

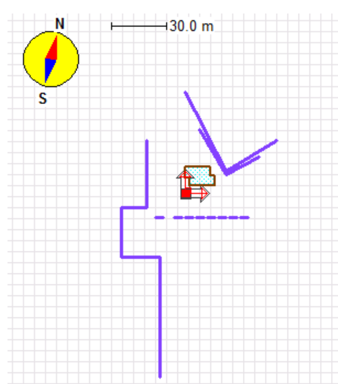


Figure 20: Orientation and site shading illustration

When modeling shade from nearby buildings and surrounding vegetation, observations on the building site and pictures from Google Map were used. The closest buildings and vegetation were implemented in IDA ICE. For the vegetation, assumed transparency values were assigned. The resulting building site in IDA ICE can be seen in Figure 22, the real building site in Figure 21.

Figure 21: Real building site (GoogleMaps, 2015).

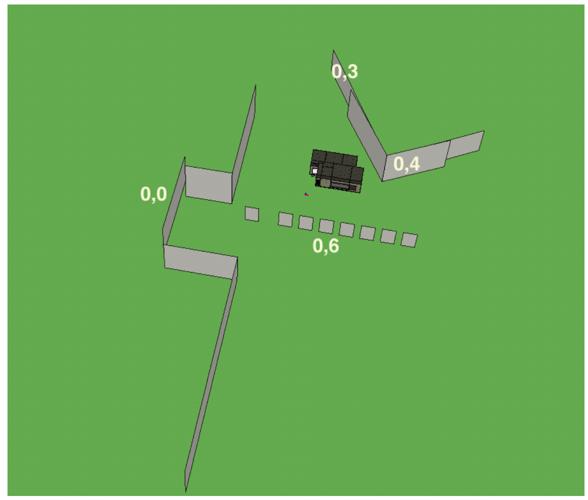


Figure 22: Building site in IDA ICE. The numbers represent transparency

The two trees closest to Living Lab seen in Figure 21 were not implemented in the IDA ICE model. These trees will be removed before the measurements period starts.

The location of the building was set to Trondheim, the wind profile to default urban.

3.3 Evaluation criteria

To create an impression of the thermal environment and use of energy for the different system solutions, a careful selection of what results to obtain had to be made.

3.3.1 Air temperatures

The desired range of temperatures for the building was 20-26^{o}C. The results should show whether these requirements were held. To limit the amount of results presented, results regarding indoor temperature would only be obtained for a few zones. It was decided to use the zone most representative for the thermal environment and also the zones most vulnerable to high indoor temperatures and low indoor temperatures.

Most representative zone

The zone assumed to best represent the whole building in terms of thermal conditions, was the living room. It contains the largest area of the building and it is where the occupants are mostly expected to be. It is vulnerable to overheating due to the large window on the south facade. It is also vulnerable to overcooling considering both windows dedicated to cross-flow ventilation are located in this room. To support this choice, a comparison of the indoor temperatures in all zones was studied. An example case of eight days with outdoor temperatures 10-15^{o}C, occupancy all hours except 08.00-17.00 and various combinations of solar radiation and wind was used. All windows were held closed and mechanical ventilation turned on. The resulting indoor air temperatures are presented in Figure 23. The results show that the temperatures do not vary much through the different zones. It also shows that the living room represents a temperature close to the maximum during warm periods and minimum during cold periods. Results regarding maximum and minimum temperature for this zone were included in the evaluation criteria. To give a better impression on the thermal conditions, it was decided to also include the number of hours with temperature below 20^{o}C and hours with temperature above 24^{o}C, 26^{o}C and 28^{o}C.

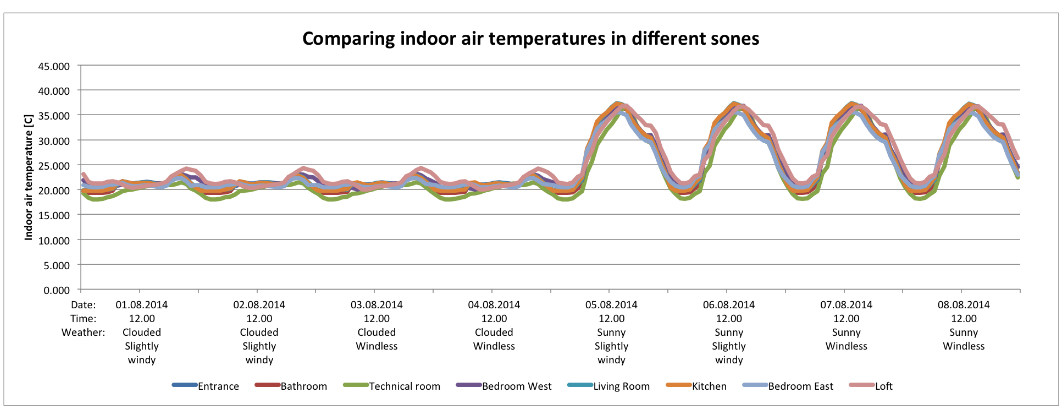


Figure 23: Example of indoor air temperature for all rooms

Most vulnerable zones
To determine which zones were most vulnerable to overheating and overcooling, two simulations using the same eight day example case were conducted. For the first simulation all windows were held closed to determine which zones were most vulnerable to overheating. In the second simulation all mechanically controlled windows were opened to half of the maximum opening to determine which zones were most vulnerable to overcooling due to window control. The results are presented in Table 10.

Table 10: Average temperatures for each zone used to determine the most vulnerable areas

	Average temperatures from 8-day simulations of example case	
	All windows closed	Controlled windows half open
Entrance	24.76^{o}C	13.55^{o}C
Bathroom	24.46^{o}C	15.50^{o}C
Technical room	23.42^{o}C	16.02^{o}C
Bedroom West	**24.93^{o}C**	15.30^{o}C
Living Room	24.87^{o}C	13.27^{o}C
Kitchen	24.87^{o}C	**13.06^{o}C**
Bedroom East	24.54^{o}C	15.29^{o}C
Loft	**25.56^{o}C**	14.90^{o}C

The loft and west bedroom obtained the highest average temperature when all windows were held closed. The loft is vulnerable to overheating due to its location. It is the only zone located at the second floor. The bedrooms are vulnerable to overheating because they are the only rooms where the occupants are expected to reside without having mechanically controlled windows. The bedroom located in the west part of the building is more exposed to solar radiation, and is therefore more vulnerable to overheating. It was decided that the maximum temperature and the number of hours with temperature above 26^{o}C in these zones should be included in the evaluation criteria.

When all mechanically controlled windows were set to half opening, the minimum average temperature was found in the kitchen. Skylight windows are located in the kitchen, and the large south window is positioned right next to the room. It is therefore exposed to rapidly airflow exchanges when natural ventilation is utilized. It was decided to include the minimum temperature and the number of hours with temperature below 20^{o}C for this zone.

3.3.2 Air velocities

The air velocities have a particular impact on the thermal environment in buildings that utilize ventilative cooling. IDA ICE does not provide results regarding air velocities in zones. In order to include air velocities in the evaluation, additional estimates were therefore necessary. The method included a calculation procedure for estimating the inlet air velocity through the windows. Also, CFD-analysis and additional calculations were used to estimate the impact of various inlet air velocities on the air velocities in the actual occupied areas.

It was decided to only include air velocities in one zone, to limit the amount of results. Air velocities caused by the south window were chosen. This window is located in the most representative zone and it is the main window for inlet air. It is also the window most likely to cause draught, due to its location and height.

Inlet air velocities
Stensaas (2001) has published a method for estimating the air velocity through a window by combining pressure balance with the continuity equation. This method was chosen because it is simple enough to only require inputs that IDA ICE could provide and still takes both buoyancy and wind forces into account.

The method starts with estimations of the driving pressure. The total driving pressure for a building is found by the sum of the wind pressure, Equation (4), and buoyancy pressure, Equation (5). Combining the driving pressure and the pressure loss, Equation (6), results in a pressure balance. When this pressure balance is combined with the continuity formula, Equation (7), an expression for the supply air velocity is found, see Equation (8). (Stensaas, 2001). Description of all parameters used are given in the nomenclature.

$$p_{d,wind} = k \times \frac{1}{2} \times \rho_o \times v_a^2 [Pa] \tag{4}$$

$$p_{d,bouy} = g \times (\rho_o - \rho_i) \times H [Pa] \tag{5}$$

$$p_l = \zeta_i \times \rho_o \times \frac{v_i^2}{2} + \zeta_o \times \rho_i \times \frac{v_o^2}{2} [Pa] \tag{6}$$

$$\rho_o \times v_i \times A_i = \rho_i \times v_o \times A_o \tag{7}$$

$$v_i = \sqrt{\frac{(g \times (\rho_o - \rho_i) \times H) + (k \times \frac{1}{2} \times \rho_o \times v_a^2)}{(\zeta_I \times \rho_o \times \frac{1}{2}) + (\zeta_o \times \rho_i \times \frac{1}{2} \times \rho_o^2 \times A_i^2 \times \frac{1}{\rho_i^2} \times \frac{1}{A_o^2})}} [m^3/s] \tag{8}$$

Outdoor temperature, indoor temperature, window opening area, wind direction and wind velocity were obtained from IDA ICE and used to calculate the inlet air velocity. The air densities were calculated using a method by Orlando et al. (2004), presented in Equation (9). The formula used when determining the saturation pressure is rendered in the Appendix Section A.5.1. The wind direction factor was set to 0.9 when directed south and 0.45 when directed east or west. All calculations were conducted with a time-step of one hour.

$$\rho = \frac{1}{T} \times \{\frac{p_{tot} - X \times p_{sat}}{R_{air}} + \frac{X \times p_{sat}}{R_{water}}\}[kg/m^3] \quad (9)$$

Air velocities in the occupied areas

It is not likely for occupants to reside in the immediate vicinity of the windows. Hence, a method for estimating the air velocity in the actual occupied areas as a function of inlet air velocities had to be developed. Several test simulations were conducted to define the magnitude of the window opening and expected inlet air velocity when applying ventilative cooling. The results revealed that the window openings were in general either large (close to maximum opening) or only ajar. Inlet air velocities from 0.5m/s to 1.0m/s were frequently recorded when the windows were close to, or at maximum opening. For ajar openings the inlet air velocity was often in the range of 1.0-2.0m/s. It was decided that the ajar openings could be analyzed with simple calculations. The large openings had to be analyzed with CFD-simulations.

Results from two CFD-analysis were studied. The simulations were conducted in cooperation with Bartosz Burzawa using the k-epsilon turbulence model in the software ANSYS Fluent. The living room was modeled and both the south window openings and the north window were implemented. The north window was held open and the south openings were set to the maximum opening of 37^o. The mesh was made more dense around the south window to obtain more accurate results in this area. Steady state simulations were conducted first to decrease the time period required to get the desired results. Then unsteady state conditions were simulated until the average velocity at the horizontal plane 1,5m above ground was stable or deviations repetitive. Inlet air velocities through the south window of 0.5m/s and 1.0m/s were tested. Illustrations are provided in Figure 24 and 25 for inlet air velocities of 0.5m/s and in Figure 26 and 27 for 1.0m/s.

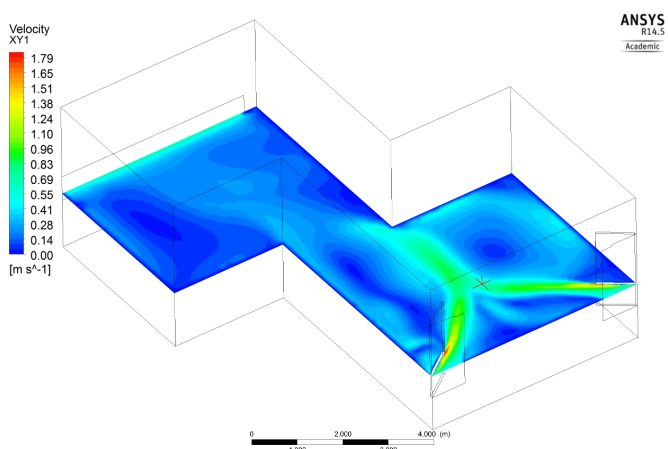


Figure 24: Illustration of the horizontal air velocity variation at height 1.5m. Inlet velocity 0.5m/s.

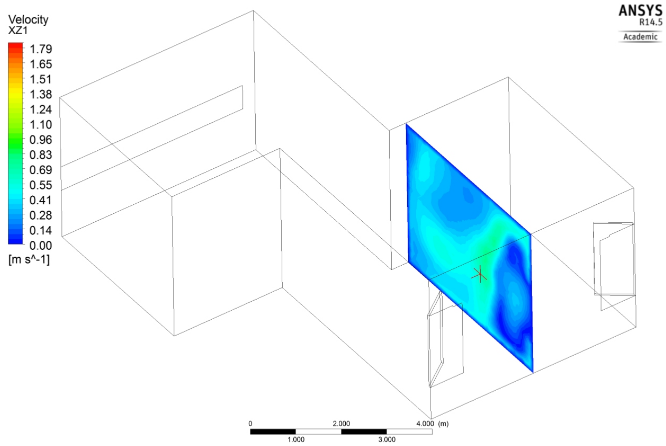


Figure 25: Illustration of the vertical air velocity variation between windows. Inlet velocity 0.5m/s.

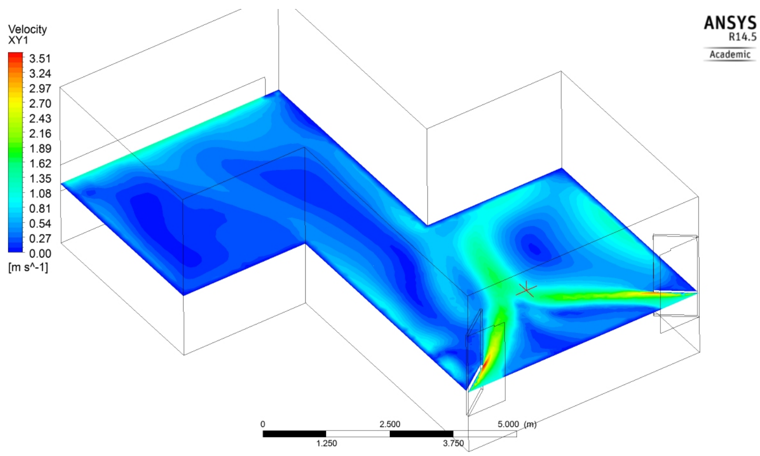


Figure 26: Illustration of the horizontal air velocity variation at height 1.5m. Inlet velocity 1.0m/s.

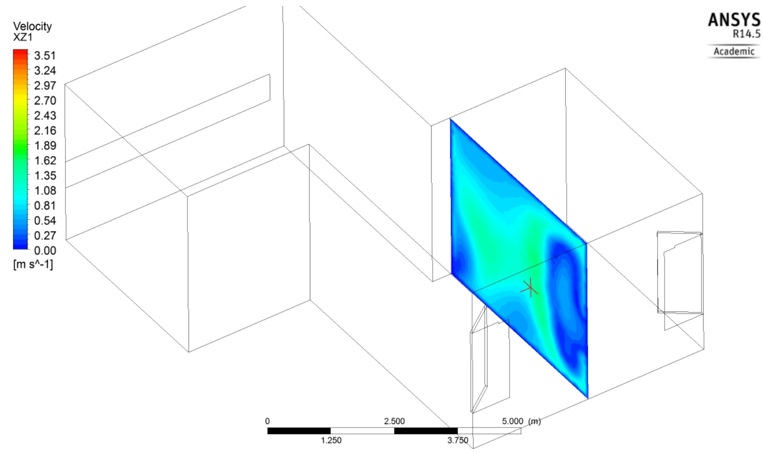


Figure 27: Illustration of the vertical air velocity variation between windows. Inlet velocity 1.0m/s.

The average air velocity 2m from the window and 1.5m above the floor was recorded. This area is particularly exposed to draught considering it being close to where the airflows from the two openings are likely to meet. The results are given in Table 11.

Table 11: Results from CFD-simulations for south window opening 37^o when applying various inlet velocities.

Results from CFD-simulation	
Velocity through window [m/s]	Velocity 2m from window, height 1.5m [m/s]
0.500	0.534
1.000	1.089

Figures 24, 25, 26 and 27 show that air velocities smaller than the inlet air velocities occurred for some areas, however not for all. When looking at Table 11, it is clear that the area close to the windows obtained air velocities similar to the inlet air velocities. Hence, air velocities in the actual occupied areas can be expected to be in the same magnitude as the inlet air velocity for large openings. It was therefore decided to view the inlet air velocities through large openings as if they were air velocities in the actual room. The average and maximum air velocity as well as amount of hours with inlet air velocities over 0.5m/s and 1.0m/s for large openings were included in the evaluation criteria.

For ajar openings, the situation was expected to be different. Ajar openings would create a more distinct jet and the velocity was assumed to reduce more rapidly as it penetrated the room. An assumption that this situation would resemble that of planar free jets was therefore made. Equation (10) obtained from Skåret (1976) was used to analyze this situation. Parameter h would here represent the width of the gap. Descriptions for the rest of the parameters can be found in the nomenclature section.

$$\frac{U_m}{U_o} = \frac{\rho_o}{\rho_i} \times \frac{1.25}{\sqrt{C_b}} \times \sqrt{\frac{h}{x + x_p}} \times \sqrt{\frac{i}{\varepsilon}} \tag{10}$$

The loss coefficient was assumed to be C_b=0.22, the contraction coefficient and the impulse coefficient ε=i=0.9. (Skåret, 1976). x_p was assumed to be very small compared to x and therefore neglected. An assumption that $\rho_o \approx \rho_r$ was made. (Mathisen, 2015). Solving for the width of the gap resulted in Equation (11).

$$h = x \times (\frac{U_m}{U_o} \times \frac{\sqrt{0.22}}{1.25})^2 \tag{11}$$

It was assumed that no one would reside closer than 0.5m from the jet. Hence, an ajar opening was narrow enough so that the inlet velocity would only have half of the impact on the air velocity in the room if the inlet air velocity was halved within 0.5m. The equation was solved using these values. The result was a maximum opening of 0.0176m. Each opening section of the south window is 0.9m wide. Hence, an opening of approximately 2% or less would at least half the inlet air velocity within 0.5m. For window openings smaller than 2% it was therefore decided to view the air velocities in the room as half of the inlet air velocity. The average and maximum air velocity as well as amount of hours with inlet air velocities over 1.0m/s and 2.0m/s for openings smaller than smaller than 2% were included in the evaluation.

3.3.3 Combining the effect of air temperature and air velocities

Indoor air temperatures and air velocities should not be studied separately when evaluating thermal comfort. It was therefore decided to include an evaluation criteria that combined the effect of both factors. Temperature corrections due to increased air velocities described in Section 2.1.3 were used. These corrections allow air temperatures above the maximum level when air velocities are adequately high. However, it was assumed that the same temperature corrections could be applied to increase the minimum level as well. For example, the standard states that the indoor air temperatures could exceed the initial maximum value by approximately 3K when air velocities are 1.0m/s. It is therefore reasonable to assume that indoor air temperatures should not be below 3K higher than the minimum temperature for the same velocity. Using the expected air velocity in the room as a function of inlet air velocities, temperature corrections as a function of inlet air velocity were found. The results are presented in Table 12.

Table 12: Indoor temperature corrections for various inlet air velocities.

Large opening 2-25%		Small opening <2%	
Inlet velocity [m/s]	Temperature correction [K]	Inlet velocity [m/s]	Temperature correction [K]
0.5	+1.8	1.0	+1.8
0.8	+2.5	2.0	+3.0
1.0	+3.0		
2.0	+4.0		

Applying the temperature corrections from Table 12 to the initial range of acceptable temperatures of 20-26^{o}C resulted in new acceptable temperature ranges given as a function of inlet air velocities, see Table 13.

Table 13: Acceptable indoor air temperatures for various inlet air velocities.

Large opening 2-25%		Small opening <2%	
Inlet velocity range[m/s]	Accepted temperatures [^{o}C]	Inlet velocity range[m/s]	Accepted temperatures [^{o}C]
<0.25	20.0-26.0	<0.5	20.0-26.0
0-25-0.65	21.8-27.8	0.5-1.5	21.8-27.8
0.65-0.9	22.5-28.5	>1.5	23.0-29.0
0.9-1.5	23.0-29.0		
1.5+	24.0-30.0		

The building was considered overheated when the indoor temperatures in the living room exceeded the maximum temperature corresponding to the current inlet air velocity with over 0.2K. It was considered overcooled when the indoor temperatures were 0.2K or more below the minimum temperature corresponding to the current inlet air velocity. The amount of hours with overheating and overcooling were included in the evaluation. Hours with overcooling when utilizing window openings were also included. This was done to distinguish overcooling due to inadequate heating from overcooling due to overuse of ventilative cooling.

3.3.4 Energy

When evaluating the effect of the different system solutions on use of energy, it was decided to focus on energy demand. This would enable a more general conclusion on the impact on energy use. Not all low-energy buildings are equipped with the same energy system as Living Lab. It was decided to present energy demand as the sum of zone heating, AHU heating, fan energy, energy for lighting and equipment. IDA ICE does not obtain results regarding energy use for the mechanical window control system. Energy for domestic hot water and pumps were omitted. Heat recovery in the AHU was included in the results, but kept separate from the energy demand.

4 Simulations and results

The study of how to apply ventilative cooling was divided in two. First the natural ventilation part of ventilative cooling was studied. In that context, two window control systems were defined. The first system utilized stack ventilation only. This system used indoor air temperature sensors and maximum opening/no opening switches in a on/off system. The other system utilized both stack and cross-flow ventilation. This system combined outdoor and indoor air temperature sensors with PI regulators. For each of the two systems, three possible solutions for when the window control system could be active were defined. The first represented daytime cooling only, the second nighttime cooling only and the third allowing ventilative cooling at all times (day and night). The best suited set-points for window opening were determined for each combination of control system and active window period. Then, the best set-point solutions were compared and the most suited active window period for each control system determined. The evaluation was based on thermal comfort only. The factors most influencing the need for window opening were also analyzed.

Secondly, complete ventilative cooling solutions were analyzed. The best window control systems were used. Simulations were conducted using window control only in a natural ventilation system and window control in combination with mechanical ventilation in concurrent and change-over mixed-mode systems. The solutions were evaluated on thermal comfort and energy use. Also, expected indoor air quality and system complexity were taken into account. An overview of the simulations performed to determine how to apply ventilative cooling is presented in Table 14.

Table 14: Overview of simulations performed to determine how to apply ventilative cooling

How to apply ventilative cooling		
Study	Solutions to be tested	Outcome
How to apply window control: Set-point temperature for opening	Various set-point temperatures for the window control systems	Set-points On/off control, Daytime exclusively Set-points On/off control, Nighttime exclusively Set-points On/off control, All hours Set-points PI-Control, Daytime exclusively Set-points PI-Control, Nighttime exclusively Set-points PI-Control, All hours
How to apply window control: Active window period	On/off control, Daytime exclusively On/off control, All hours PI control Daytime exclusively PI control, All hours	Best solution On/off control Best solution PI control Factors most influencing the need for window opening
How to apply ventilative cooling	Natural ventilation On/off control Concurrent On/off control Change-over On/off control Natural ventilation PI-control Concurrent PI-control Change-over PI-control	Best ventilative cooling solution

To decide the effect of ventilative cooling on thermal comfort and energy use, the complete ventilative cooling solutions were compared to two reference cases. Both reference cases used only mechanical ventilation and no window operation. The first represented use of hygienic ventilation (supply and extract 144m^3/h), the other represented enhanced use of mechanical ventilation (supply and extract up to 260m^3/h). An overview of the simulations performed to study the effect of ventilative cooling is presented in Table 14.

Table 15: Overview of simulations performed to determine the effect of ventilative cooling

Evaluate the effect of ventilative cooling		
Solutions to be tested	Reference cases	Outcome
Natural ventilation On/off control Concurrent On/off control Change-over On/off control Natural ventilation PI-control Concurrent PI-control Change-over PI-control	Hygienic mechanical ventilation (144m^3/h) Enhanced mechanical ventilation (260m^3/h)	Effect on the thermal environment Effect on energy use

The following section provides more detailed descriptions of the simulations conducted. It also presents and analyzes the results. The results will be further discussed in Section 5.

4.1 How to apply window control

4.1.1 Scenarios used for the simulations

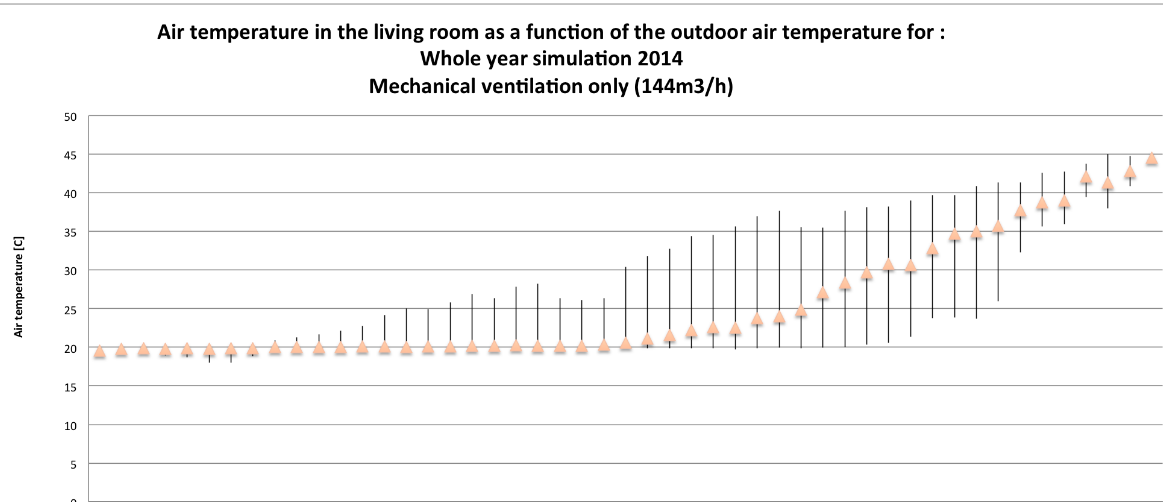
To enable testing of many different window set-point temperatures and active window periods, scenarios with short time periods had to be constructed. Several scenarios were designed to cover a specter of the most relevant occupancy schedules and outdoor conditions.

It was assumed that an eight-day period would be sufficient to demonstrate how the various window control systems worked for a specific set of conditions. Separate climate files would be used for these periods. Hence, the choice of simulation period would only influence the way IDA ICE positions the sun in the sky. The 8-day period used for the simulations was from May 1st to May 8th. This period could represent eight days of spring or eight days of fall in terms of solar positioning. Ventilative cooling is assumed to be frequently needed at this time. It is also assumed to be more challenging to utilize ventilative cooling in spring or fall than in summer, due to increased risk of draught and overcooling.

To enable an evaluation of how occupancy influence the need for window control, the schedules representing weekday and weekends were studied separately.

Artificial climate files were constructed and used for the eight-day simulations. This decision was made to enable an evaluation on how the outdoor conditions influence the need for window control. To determine what outdoor temperatures to include in the artificial climate files, a whole year simulation without ventilative cooling was performed. This simulation was conducted using climate data from 2014 in Værnes, Trondheim. All windows were held shut and mechanical ventilation turned on. The occupancy was set to change from the weekday pattern Monday-Friday to weekend pattern Saturday-Sunday. The results were organized to illustrate which indoor temperatures could be expected at various outdoor temperatures. The results are presented in Figure 28.

Figure 28 reveals that there is a significant risk of overheating in Living Lab. High indoor air temperatures occurred even when outdoor temperatures were below zero. However, looking at the average and maximum indoor temperatures, it was evident that the risk of uncomfortably high indoor air temperatures mostly occurred for outdoor temperatures above 10^{o}C. The range of outdoor temperatures most likely to result in the need for window control was therefore from 10-30^{o}C. However, when the outdoor temperature exceed 20^{o}C, the risk of draught and overcooling will decrease. Hence, windows could be fully open without causing any significant difficulties. Also, temperatures are rarely exceeding 20^{o}C in Trondheim. The range of outdoor temperatures most interesting when studying window control for this building was therefore 10-20^{o}C. To break down the study, it was decided to look at outdoor temperatures from 10-15^{o}C and 15-20^{o}C separately. Hence, two artificial climate files were constructed, one representing outdoor air temperatures in the range of 10-15^{o}C and the other 15-20^{o}C.



	-18	-17	-16	-15	-14	-13	-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5
Maximum temperature	20	20	20	20	20	20	21	20	21	21	22	22	23	24	25	25	26	27	26	28	28	26	26	26
Minimum temperature	20	20	20	19	19	18	18	19	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20
Average temperature	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20

	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Maximum temperature	30	32	33	34	35	36	37	38	36	35	38	38	38	39	40	40	41	41	41	43	43	44	45	45	45
Minimum temperature	20	20	20	20	20	20	20	20	20	20	20	20	21	21	24	24	24	26	32	36	36	39	38	41	44
Average temperature	21	21	22	22	23	23	24	24	25	27	28	30	31	31	33	35	35	36	38	39	39	42	41	43	45

Figure 28: Results from a whole year simulation using only mechanical ventilation. The indoor air temperatures are sorted and presented as a function of the outdoor air temperature.

The thermal environment can also be influenced by the presence of solar radiation and wind. It was therefore determined that each of the two the climatic files should include cloudy days, sunny days, days with almost no wind and more windy days. In order to enable an evaluation of what outdoor conditions mostly influence the need and effect of window control, each couple of days in the eight-day period had a unique combination of solar radiation and wind.

The climatic files contained hourly values for air temperature, relative humidity, wind velocity, wind direction, direct and diffuse solar radiation. The temperatures from the set ranges were allocated to the different hours as they often are distributed through the day. Values for relative humidity and wind direction were obtained for a typical day provided by YR (2014). Values for wind velocity were obtained from the same source. For the windless days, air velocities for the date in Trondheim 2014 with the smallest mean and maximum velocity were used. For the windy days, air velocities for the date with mean velocity equal to the yearly mean velocity in Trondheim 2014 were used. Values for solar radiation were obtained from Stensaas (1980). For sunny days, the maximum values for solar radiation on July 15th were used. For the clouded days the same values were used but divided by four. For diffuse radiation, values equivalent to one fourth of the direct radiation were applied. The specific values used for all outdoor parameters are rendered in the Appendix Section A.7.

When combining the occupancy schedules with the outdoor temperature ranges, the result were four different scenarios. An overview of these scenarios is provided in Table 16. The weather applied to each of the eight days is presented in Table 17.

Table 16: Description of the four scenarios used for the eight-day simulations

Scenarios for the eight-day simulations		
Scenario nr	Temperature profile	User pattern
1	10-15 ^{o}C outdoor	Occupied all hours except from 08.00-17.00
2	15-20 ^{o}C outdoor	Occupied all hours
3	10-15 ^{o}C outdoor	Occupied all hours except from 08.00-17.00
4	15-20 ^{o}C outdoor	Occupied all hours

Table 17: Description of the 8-day weather file used for each scenario

Outdoor conditions	
Day nr	Weather profile
1	Windy and clouded
2	Windy and clouded
3	Windless and clouded
4	Windless and clouded
5	Windy and sunny
6	Windy and sunny
7	Windless and sunny
8	Windless and sunny

To assure that the four scenarios represent the desired part of the whole year simulations,

simulations were conducted for each of the four scenarios. As for the whole year simulation, all windows were held shut and mechanical ventilation turned on. The maximum, minimum and average indoor temperature for each scenarios were compared to the results from the whole year simulation. The results were coinciding, see Table 18 and 19.

Table 18: Indoor temperatures from the whole year simulation.

Whole year simulation			
Outdoor temperature	Max indoor temperature	Min indoor temperature	Average indoor temperature
10-15^{o}C	37.7^{o}C	19.7^{o}C	24.2^{o}C
15-20^{o}C	39.7^{o}C	20.0^{o}C	29.9^{o}C

Table 19: Indoor temperatures from the 8-day simulations.

8-Day year simulations			
Occupancy/Outdoor temperature	Max indoor temperature	Min indoor temperature	Average indoor temperature
10-15^{o}C/All day except 08.00-17.00	37.4^{o}C	19.9^{o}C	24.8^{o}C
10-15^{o}C/All day	38.9^{o}C	19.9^{o}C	26.0^{o}C
15-20^{o}C/All day except 08.00-17.00	40.6^{o}C	19.9^{o}C	27.0^{o}C
15-20^{o}C/All day	41.2^{o}C	19.9^{o}C	28.0^{o}C

4.1.2 Window set-point temperature for opening

To narrow the study of how to apply window control in Living Lab, it was decided that the control systems would only utilize temperature sensors. Two different control systems were defined. The first was an on/off control system and the second a PI-regulation system. That way, an evaluation on whether more complex temperature control was needed to acquire the desired effect, or if a simple control system would be sufficient, could be made. Natural ventilation was studied without the use of mechanical ventilation to distinct the effect of window control alone. Hence, the mechanical fans in the central AHU were turned off. Before designing the window control systems, the maximum window openings as percentage area of the total window area had to be determined. The calculation of these percentages can be found in the Appendix Section A.6. The results are presented in Table 20.

Table 20: Maximum window opening for the mechanically controlled windows

Window type	Maximum opening [^{o}]	Maximum opening [%]
South window	37	25
Skylight windows	30	50
North window	39	63

On/off window control

The on/off control system utilized indoor air temperature sensors and maximum opening/no opening switches. The system was designed to open the windows to maximum allowed opening when indoor air temperatures exceeded a certain set-point, and keep the windows open until another set-point temperature was reached. That way, this system could also illustrate how one could manually open windows. The set-points to be determined for this window control system was which indoor temperature should signal window opening, and which should signal window closing. A description of the on/off window control design and the set-points to be determined are presented in Table 21.

Table 21: Description of the on/off window control system. Set-points to be determined: T1-T6. 0=windows closed, 1=windows open.

On/off design description			
Type	Hours applied	Windows utilized	Design control signal
WC Day	07.00-23.00	South + Skylight	Tzone < T1 -> 0 Tzone > T1 -> 1 until T2 is reached
WC Night	23.00-07.00	South + Skylight Kitchen only	Tzone < T3 -> 0 Tzone > T3 -> 1 until T4 is reached
WC All hours	All hours	South + Skylight (-23-07 Loft)	Tzone < T5 -> 0 Tzone > T5 -> 1 until T6 is reached

It was decided that the on/off window control system should be activated by a timer, to reduce the complexity. Daytime and nighttime were divided between 07.00-23.00 and 23.00-07.00. Since controlled by a timer, the system would operate also when no one was home. It was therefore decided not to utilize the north window, due to safety reasons. The north window opens directly in to the living room and it has a large opening area. Hence, only stack ventilation through the south window and the skylight windows would be utilized. Not to jeopardize the comfort in the loft, windows in this room would not be mechanically controlled at night.

An example of the construction of a on/off window control system in IDA ICE is given in Figure 69. The example illustrates the window control system for the south window when set to open at 24^{o}C and close at 20^{o}C. A detailed description of how the control system works is provided in the Appendix Section A.8.1. Figure 30 illustrates the operation of the control system.

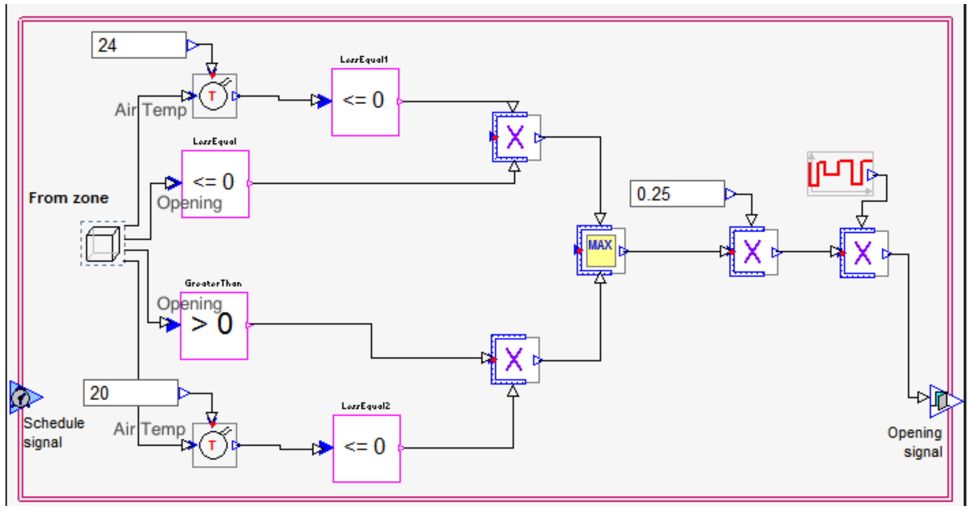


Figure 29: Example of construction of a on/off window control system in IDA ICE. Here: south window open at 24^{o}C, close at 20^{o}C

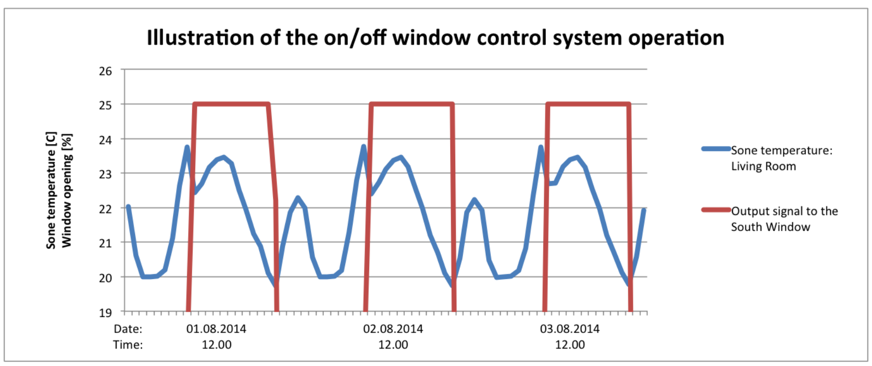


Figure 30: Illustration of the operation of a on/off window control system in IDA ICE. Here: south window open at 24^{o}C, close at 20^{o}C

Exclusively daytime operation

When testing various set-points for the on/off window control system, the starting point was the initial range of acceptable temperatures. To keep the indoor temperatures below 26^{o}C opening temperatures of 25^{o}C, 24^{o}C and 23^{o}C were tested. The minimum indoor temperature in the initial range was 20^{o}C. It was therefore decided to test window closing at this temperature. The results are presented in Table 22.

The results show that the temperatures were almost kept within the initial acceptable range. However, increased air velocities and indoor air temperatures at the lower part of the initial acceptable temperature range were occurring simultaneously causing overcooling of the building. It was therefore decided to increase the closing set-point temperature. Recorded maximum air velocities were 0.86m/s which corresponds to an approximate temperature correction of 2K. The indoor air temperature should therefore not be below 22^{o}C during cooling periods. Hence, for the second set-point testing, a new closing temperature of 22^{o}C was tested.

Table 22 also shows that opening temperature 23^{o}C significantly increased the risk of overcooling compared to opening temperatures of 25^{o}C and 24^{o}C. It was therefore decided that this opening temperature was the weakest solution. For the second set-point testing of the daytime, on/off window control system opening at 25^{o}C and 24^{o}C and closing at 22^{o}C was therefore tested. The results are presented in Table 23.

Table 22: Results for the first set-point temperature testing for the on/off daytime window control system. Time is given as percentage of the simulation period (768 hours).

	On/off Day 25-20^{o}C		On/off Day 24-20^{o}C		On/off Day 23-20^{o}C	
	Result	Time [%]	Result	Time [%]	Result	Time[%]
Overheating						
Max temp Living [^{o}C]	25.45		25.13		24.65	
H Tliving >24^{o}C	96	12.5	33	4.3	8	1.0
H Tliving >26^{o}C	0	0.0	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0	0	0.0
Max temp Loft [^{o}C]	25.81		25.76		25.54	
H Tloft >26^{o}C	0	0.0	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	25.53		25.13		24.76	
H TbedroomW >26^{o}C	0	0.0	0	0.0	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***	***0***	***0.0***
Overcooling						
Min temp Living [^{o}C]	19.88		19.90		19.82	
H Tliving <20^{o}C	66	8.6	49	6.4	65	8.5
Min temp Kitchen [^{o}C]	19.53		19.53		19.53	
H Tkitchen <20^{o}C	168	21.9	171	22.3	183	23.8
H overcooled	***74***	***9.6***	***80***	***10.4***	***103***	***13.4***
Air velocities						
H south window open	308	40.1	300	39.1	330	43.0
Average velocity [m/s]	0.38		0.39		0.39	
Average velocity 2-5% [m/s]	0.38		0.39		0.39	
Max velocity 2-25% [m/s]	0.86		0.86		0.86	
H >0.5m/s 2-25%	49	6.4	50	6.5	58	7.6
H >1.0m/s 5-25%	0	0.0	0	0.0	0	0.0
Average velocity <2% [m/s]	n.a.		n.a.		n.a.	
Max velocity <2% [m/s]	0.00		0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0	0	0.0
H > 2.0m/s <2%	0	0.0	0	0.0	0	0.0

Table 23: Results for the second set-point temperature testing for the one/off daytime window control system. Time is given as percentage of the simulation period (768 hours).

	On/off Day 25-22^{o}C		On/off Day 24-22^{o}C	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	25.40		25.06	
H Tliving >24^{o}C	248	32.3	83	10.8
H Tliving >26^{o}C	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0
Max temp Loft [^{o}C]	25.86		25.73	
H Tloft >26^{o}C	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	26.21		25.67	
H TbedroomW >26^{o}C	9	1.2	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***
Overcooling				
Min temp Living [^{o}C]	19.96		19.97	
H Tliving <20^{o}C	56	7.3	49	6.4
Min temp Kitchen [^{o}C]	19.78		19.80	
H Tkitchen <20^{o}C	105	13.7	110	14.3
H overcooled	***2***	***0.3***	***1***	***0.1***
Air velocities				
H south window open	256	33.3	261	34.0
Average velocity [m/s]	0.38		0.39	
Average velocity 2-5% [m/s]	0.38		0.39	
Max velocity 2-25% [m/s]	0.80		0.84	
H >0.5m/s 2-25%	37	4.8	40	5.2
H >1.0m/s 5-25%	0	0.0	0	0.0
Average velocity <2% [m/s]	n.a.		n.a.	
Max velocity <2% [m/s]	0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0
H > 2.0m/s <2%	0	0.0	0	0.0

Table 23 shows that both solutions prevented overheating and caused almost no overcooling. The biggest difference between the two set-point solutions was found on cloudy days when the outdoor temperatures were between 10^{o}C and 15^{o}C. On these days, the indoor temperatures barely exceeded 24^{o}C. The windows would open for the 24-22^{o}C set-point solution but not for the 25-22^{o}C solution. The amount of hours with temperatures over 24^{o}C was therefore reduced for the 24-22^{o}C set-point solution.

The results also show that set-point solution 25-22^{o}C caused indoor air temperatures below 20^{o}C in the living room more frequently than set-point solution 24-22^{o}C. This was not expected. When taking a closer look at the results, the difference was found at night time. Set-point solution 25-22^{o}C had more hours with temperatures barely below 20^{o}C. The difference between the two solutions during these hours were less than 0.1K, hence insignificantly small.

The risk of overcooling was therefore approximately the same for the two solutions. Due to more frequently use of window openings at set-point 24-22^{o}C, the amount of hours in the upper range of the acceptable temperature range were reduced. Hence, creating a more stabile thermal environment with smaller temperature variations. The best solution was therefore 24-22^{o}C. This solution recorded both air velocity of 0.66m/s when indoor air temperature was 22.29^{o}C and zero increase in air velocity when indoor air temperature was 25^{o}C, supporting the choice that the set-point should neither be increased nor decreased.

To illustrate the operation of the best set-point solution and the resulting thermal environment, the scenario with the biggest risk of overcooling is presented in Figure 31 and 32. The scenario with the biggest risk of overheating is illustrated in Figure 33 and 34.

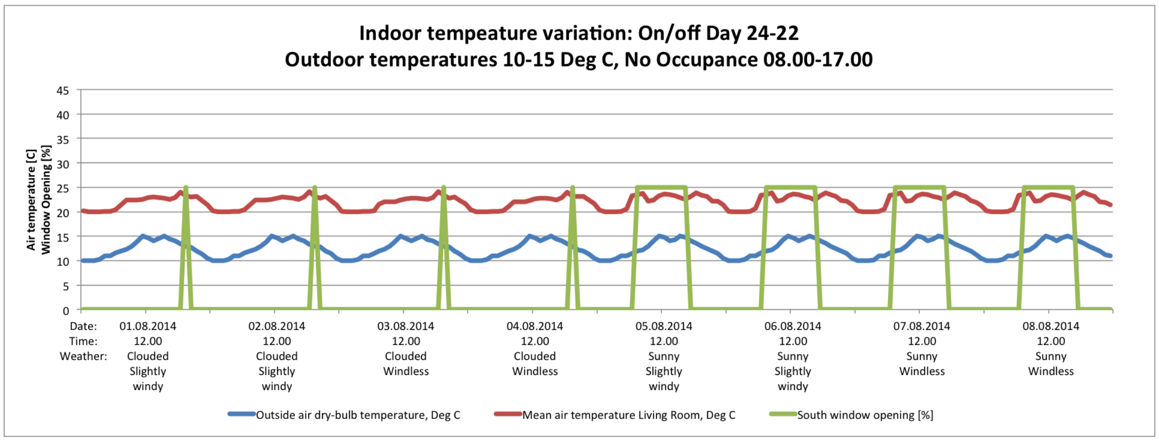


Figure 31: Window opening and indoor air temperature for the best on/off daytime window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

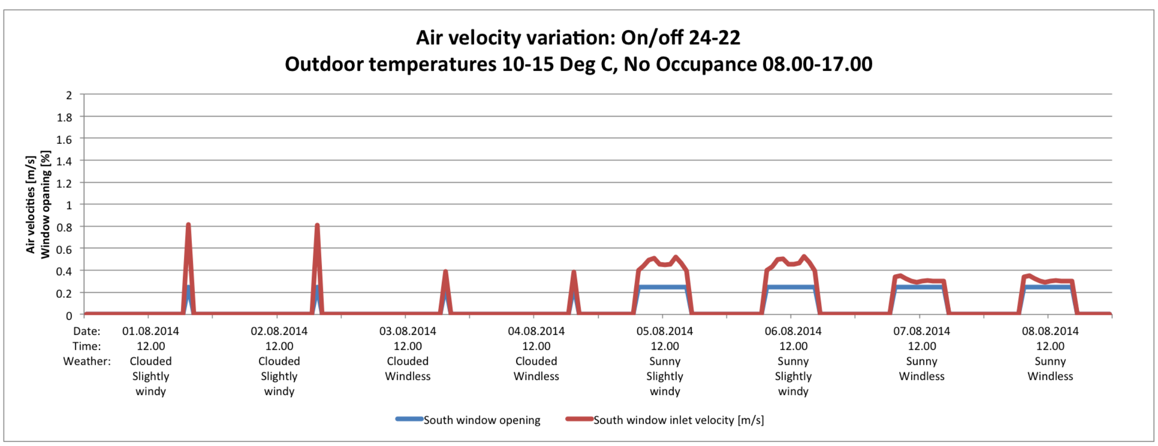


Figure 32: Window opening and indoor air velocities for the best on/off daytime window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

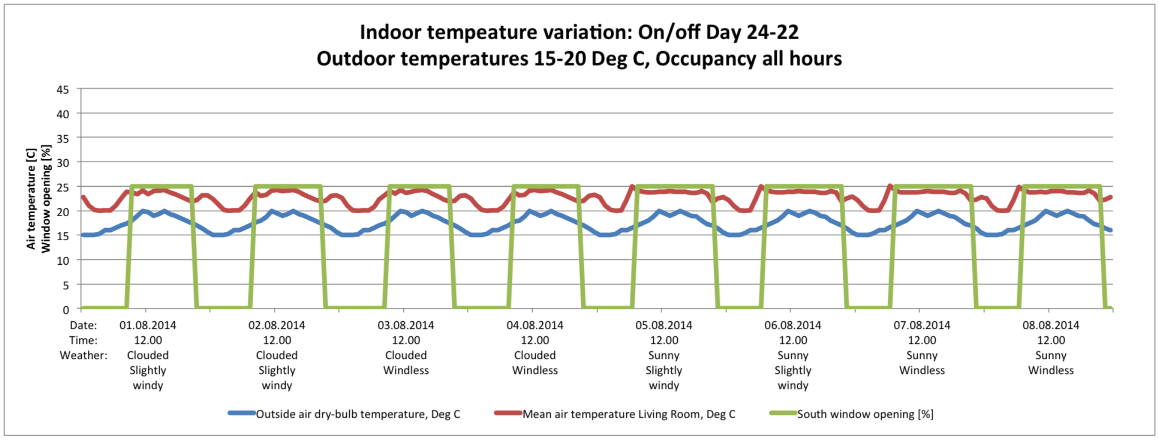


Figure 33: Window opening and indoor air temperature for the best on/off daytime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

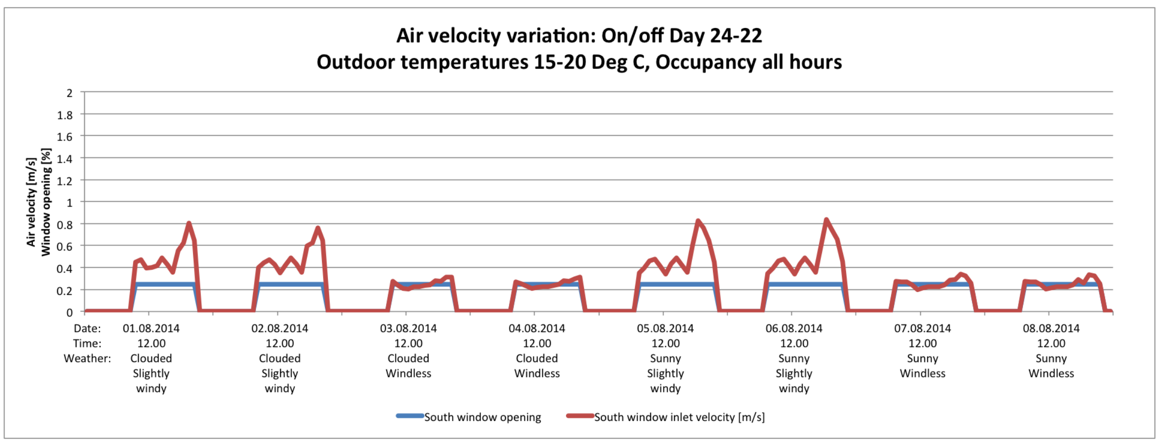


Figure 34: Window opening and indoor air velocities for the best on/off daytime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

Exclusively nighttime operation

The nighttime window control system was not expected to provide sufficient results when applied separately because it did not provide any form of ventilation during daytime. However, it was simulated to see if thermal storage ventilative cooling had an effect on the thermal environment in Living Lab.

It was decided to first test a solution were the windows would open at 22^{o}C and close at 20^{o}C. This was done to see if frequent cooling could be obtained without reducing the temperatures below the initial minimum level. When applying nighttime ventilative cooling, the only operating windows were located in the kitchen and living room. No one was expected to be there during the night. Hence, reducing the indoor temperature to 19^{o}C for nighttime ventilative cooling could be accepted. It was therefore decided to also test a set-point solution that would open windows at 20^{o}C and close them at 19^{o}C. The results from both testings are presented in Table 24.

Table 24 shows that window openings would only be used for very few hours when not allowing the indoor air temperatures to be reduced below the initial acceptable level. It also shows that reducing the indoor air temperature slightly at night had no positive effect on the daytime temperatures.

Table 24: Results for the set-point temperature testing for the on/off nighttime window control system. Time is given as percentage of the simulation period (768 hours).

	On/off Night 22-20^{o}C		On/off Night 20-19^{o}C	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	48.38		48.17	
H Tliving >24^{o}C	435	56.6	417	54.3
H Tliving >26^{o}C	380	49.5	377	49.1
H Tliving >28^{o}C	316	41.1	316	41.1
Max temp Loft [^{o}C]	47.69		47.65	
H Tloft >26^{o}C	399	52.0	393	51.2
Max temp BedroomW [^{o}C]	48.75		48.58	
H TbedroomW >26^{o}C	394	51.3	393	51.2
H overheated	***377***	***49.1***	***373***	***48.6***
Overcooling				
Min temp Living [^{o}C]	19.56		18.73	
H Tliving <20^{o}C	80	10.4	79	10.3
Min temp Kitchen [^{o}C]	19.51		18.59	
H Tkitchen <20^{o}C	152	19.8	170	22.1
H overcooled	***18***	***2.3***	***78***	***10.2***
Air velocities				
H south window open	40	5.2	87	11.3
Average velocity [m/s]	0.34		0.31	
Average velocity 2-5% [m/s]	0.34		0.31	
Max velocity 2-25% [m/s]	0.47		0.49	
H >0.5m/s 2-25%	0	0.0	0	0.0
H >1.0m/s 5-25%	0	0.0	0	0.0
Average velocity <2% [m/s]	n.a.		n.a.	
Max velocity <2% [m/s]	0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0
H > 2.0m/s <2	0	0.0	0	0.0

The results from set-point solution 20-19^{o}C revealed that lowering the set-point temperatures did not reduce the daytime temperatures significantly. It did however cause an increase in overcooled hours during the night. Indoor air temperatures below 19^{o}C were occurring simultaneously as air velocities up to 0.5m/s which should not be accepted, even during nighttime. It was therefore already evident that thermal storage ventilative cooling would not have a positive effect on the thermal environment in Living Lab. Illustrations of the operation and the resulting thermal environment for the 22-20^{o}C set-point solution can be found in the Appendix Section A.9.1.

All hours opening

When testing the all hour on/off system, nighttime window control had already been proven not to have a positive effect on the thermal environment. It was therefore decided that this system should only examine if applying the same window control system used for daytime both day and night would further improve or worsen the thermal conditions. Opening at 25^{o}C, 24^{o}C and 23^{o}C and closing at 20^{o}C was therefore simulated for the first set-point testing. The results are presented in Table 25.

Table 25: Results for the first set-point temperature testing for the on/off all hours window control system. Time is given as percentage of the simulation period (768 hours).

	On/off All hours 25-20^{o}C		On/off All hours 24-20^{o}C		On/off All hours 23-20^{o}C	
	Result	Time [%]	Result	Time [%]	Result	Time[%]
Overheating						
Max temp Living [^{o}C]	25.28		24.41		23.46	
H Tliving >24^{o}C	94	12.2	29	3.8	0	0.0
H Tliving >26^{o}C	0	0.0	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0	0	0.0
Max temp Loft [^{o}C]	25.72		25.83		25.36	
H Tloft >26^{o}C	0	0.0	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	25.47		25.43		24.45	
H TbedroomW >26^{o}C	0	0.0	0	0.0	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***	***0***	***0.0***
Overcooling						
Min temp Living [^{o}C]	19.68		19.69		19.68	
H Tliving <20^{o}C	74	9.6	65	8.5	69	9.0
Min temp Kitchen [^{o}C]	19.54		19.53		19.53	
H Tkitchen <20^{o}C	186	24.2	190	24.7	195	25.4
H overcooled	***76***	***9.9***	***74***	***9.6***	***106***	***13.8***
Air velocities						
H south window open	309	40.2	305	39.7	341	44.4
Average velocity [m/s]	0.38		0.39		0.38	
Average velocity 2-5% [m/s]	0.38		0.39		0.38	
Max velocity 2-25% [m/s]	0.86		0.86		0.85	
H >0.5m/s 2-25%	49	6.4	53	6.9	57	7.4
H >1.0m/s 5-25%	0	0.0	0	0.0	0	0.0
Average velocity <2% [m/s]	n.a.		n.a.		n.a.	
Max velocity <2% [m/s]	0.00		0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0	0	0.0
H > 2.0m/s <2%	0	0.0	0	0.0	0	0.0

Table 25 shows similar results to those obtained for daytime window control system. The current set-point solutions resulted in no overheating, but significant amounts of overcooling. Indicating that the closing set-point temperature should be increased and the lowest opening set-point temperature eliminated. The second set-point testing would therefore simulate opening temperatures of 25^{o}C and 24^{o}C and closing temperature of 22^{o}C. The results are presented in Table 26.

Table 26: Results for the second set-point temperature testing for the on/off all hours window control solution. Time is given as percentage of the simulation period (768 hours).

	On/off All hours 25-22^{o}C		On/off All hours 24-22^{o}C	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	25.28		24.36	
H Tliving >24^{o}C	248	32.3	66	8.6
H Tliving >26^{o}C	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0
Max temp Loft [^{o}C]	26.00		25.83	
H Tloft >26^{o}C	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	26.22		25.64	
H TbedroomW >26^{o}C	7	0.9	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***
Overcooling				
Min temp Living [^{o}C]	19.96		19.97	
H Tliving <20^{o}C	54	7.0	50	6.5
Min temp Kitchen [^{o}C]	19.79		19.80	
H Tkitchen <20^{o}C	108	14.1	111	14.5
H overcooled	***3***	***0.4***	***0***	***0.0***
Air velocities				
H south window open	262	34.1	265	34.5
Average velocity [m/s]	0.38		0.39	
Average velocity 2-5% [m/s]	0.38		0.39	
Max velocity 2-25% [m/s]	0.81		0.85	
H >0.5m/s 2-25%	38	4.9	41	5.3
H >1.0m/s 5-25%	0	0.0	0	0.0
Average velocity <2% [m/s]	n.a.		n.a.	
Max velocity <2% [m/s]	0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0
H > 2.0m/s <2%	0	0.0	0	0.0

The results from the second set-point testing also resembled the results from daytime window control system. There were no overheated hours for either of the test-point solutions. Overcooling was almost eliminated. Due to more frequent window operation for the 24-22^{o}C solution, smaller temperature variations were obtained. This solution was therefore deemed the most fitted solution also for the all hour window control system.

To illustrate the operation of the best set-point solution and the resulting thermal environment, the results for the scenario with the biggest risk of overcooling are presented in Figure 35 and 36. The scenario with the biggest risk of overheating is illustrated in Figure 37 and 38.

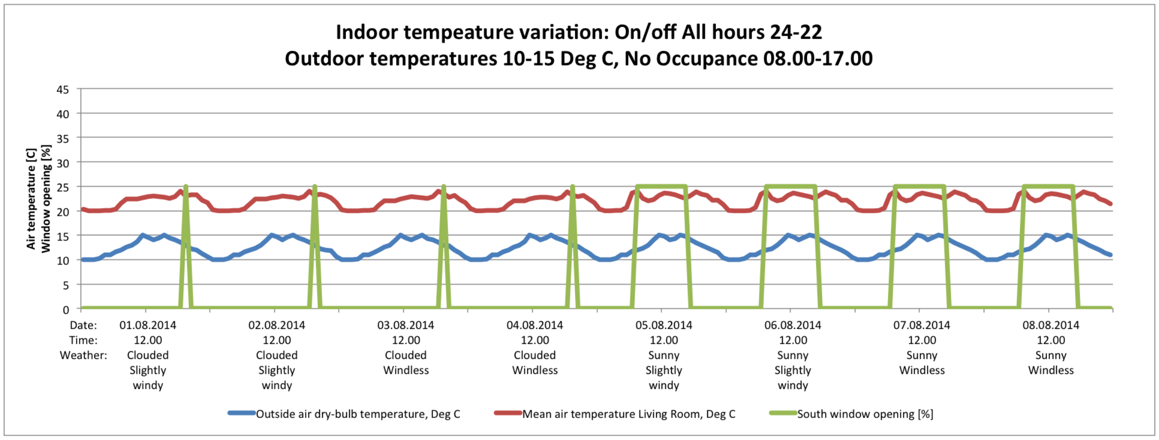


Figure 35: Window opening and indoor air temperature for the best on/off all hours window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

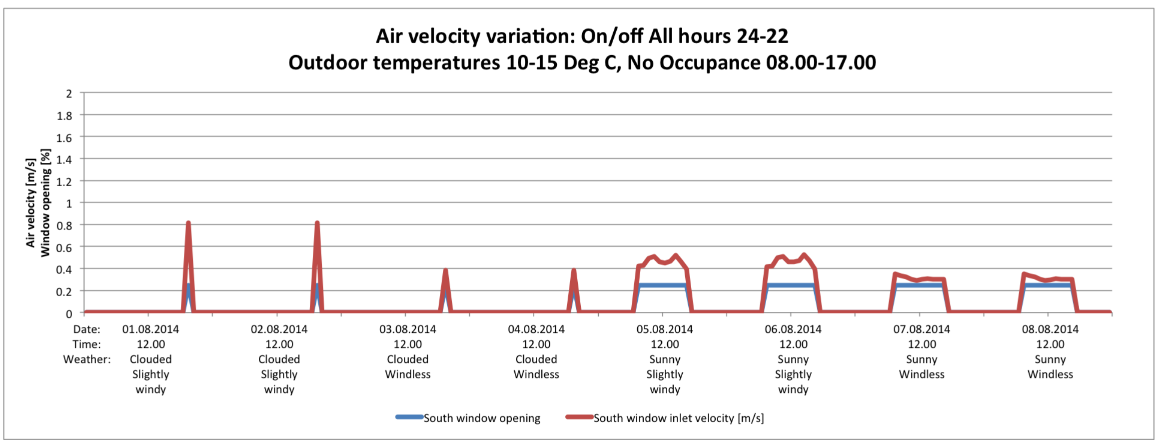


Figure 36: Window opening and indoor air velocities for the best on/off all hours window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

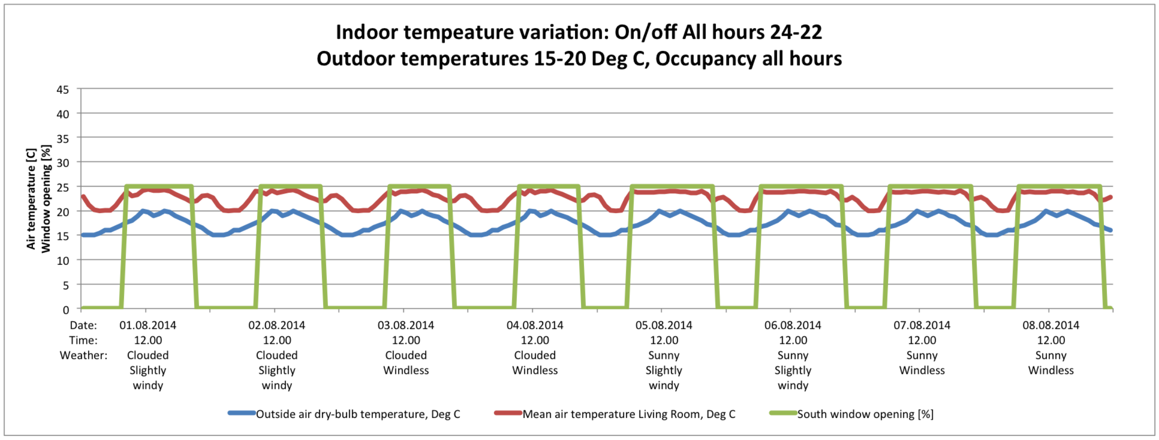


Figure 37: Window opening and indoor air temperature for the best on/off all hours window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

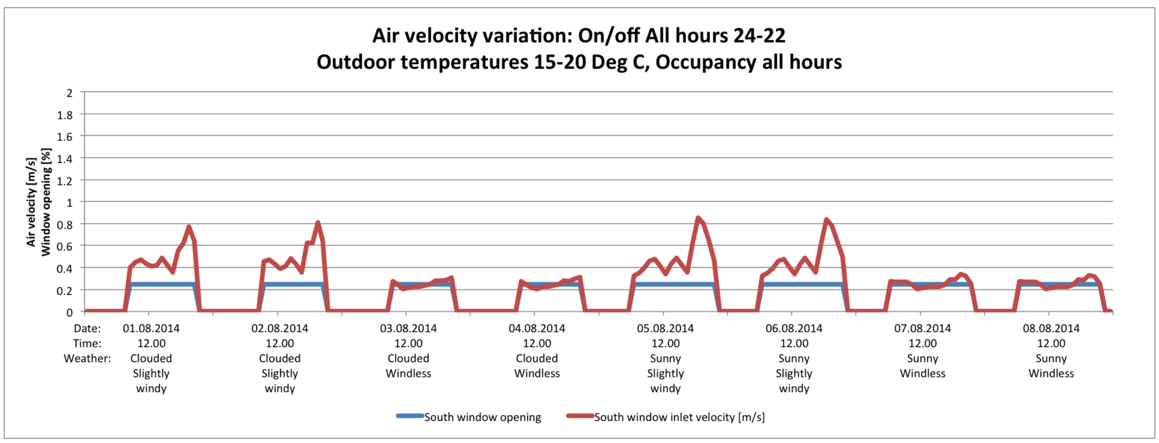


Figure 38: Window opening and indoor air velocities for the best on/off all hours window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

PI window control

The more complex temperature control system utilized both indoor and outdoor air temperature sensors and PI regulators. The PI regulators were programmed to keep the indoor temperature below certain temperatures at different outdoor air temperature levels. In other words, the outdoor temperatures would be determining what maximum indoor temperatures the PI regulators should attempt to maintain. Three different levels were used. These three outdoor temperature levels were predefined, to limit the possible set-points solutions. The system was designed to keep indoor temperatures below one level when the outdoor temperatures were under 15^{o}C, another when the outdoor temperatures were between 15 and 20^{o}C, and a third when the outdoor temperature exceeded 20^{o}C. The set-points to be determined for this window control system were which maximum indoor temperatures to keep at these three levels. A description of the PI window control design and the set-points to be determined are presented in Table 27.

Table 27: Description of the PI window control system. Set-point temperatures to be determined: T7-T15.

PI design description			
Type	Hours applied	Windows utilized	Control signal design
WC Day	Weekday:07.00-23.00 Weekend:09.00-24.00	South + Skylight + North (-08-17 weekdays)	Tout < 15^oC -> Hold T7 15^oC < Tout < 20^oC -> Hold T8 20^oC < Tout -> Hold T9
WC Night	Weekday:23.00-07.00 Weekend:24.00-09.00	South + Skylight Kitchen only	Tout < 15^oC-> Hold T10 15^oC < Tout < 20^oC -> Hold T11 20^oC < Tout -> Hold T12
WC All hours	All Hours	South + Skylight(-23-07 weekday -24-09 weekend Loft) + North(-08-17 weekdays)	Tout < 15^oC -> Hold T13 15^oC < Tout < 20^oC -> Hold T14 20^oC < Tout -> Hold T15

To make this system more advanced than the previous system it was decided that it should be activated by presence. The schedules for these systems were designed based on the occupancy schedules. The daytime window control system would turn on when the occupants got up in the morning, and turn off when they went to bed. The nighttime control system would do the opposite, while the all hour system would be active day and night. Since controlled by presence, the north window could be utilized when the building was occupied. Hence, cross-flow ventilation would be utilized through the south and north window in addition to stack ventilation through the skylight windows. For security reasons, the north window control system would be turned off when no one was home. Not to jeopardize the comfort in the loft, the skylight windows in this room would not be mechanically controlled at night.

An example of the construction of a PI window control system in IDA ICE is given in Figure 70. The example illustrates the control system for the south window when set to keep indoor air temperature below 21^oC when outdoor temperatures were under 15^oC, 23^oC when outdoor temperatures were between 15 and 20^oC and 25^oC when outdoor temperatures exceeded 20^oC. A detailed description of how the control system works is provided in the Appendix Section A.8.2. Figure 40 illustrates the operation of the control system.

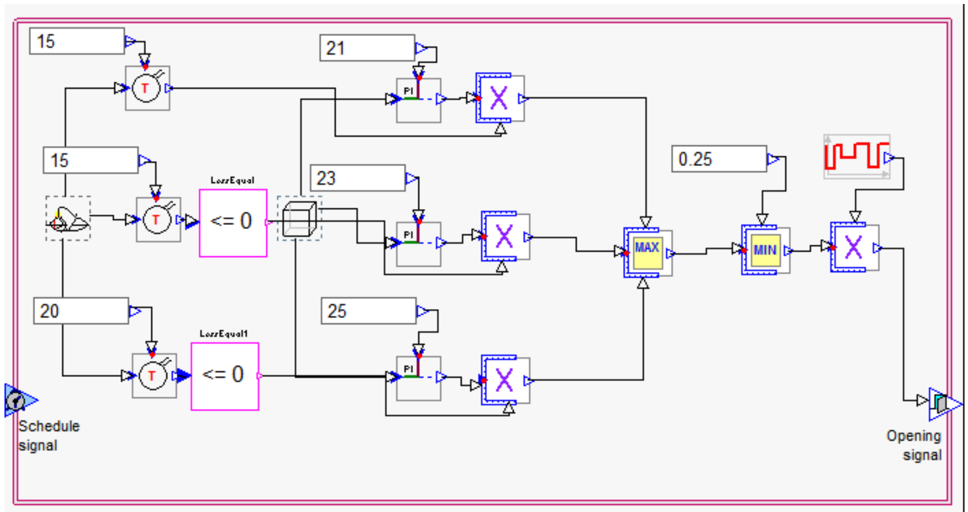


Figure 39: Example of construction of a PI window control system in IDA ICE. Here: south window keep 21^{o}C when below 15^{o}C outside, keep 23^{o}C when 15-20^{o}C outside, keep 25^{o}C when above 20^{o}C outside

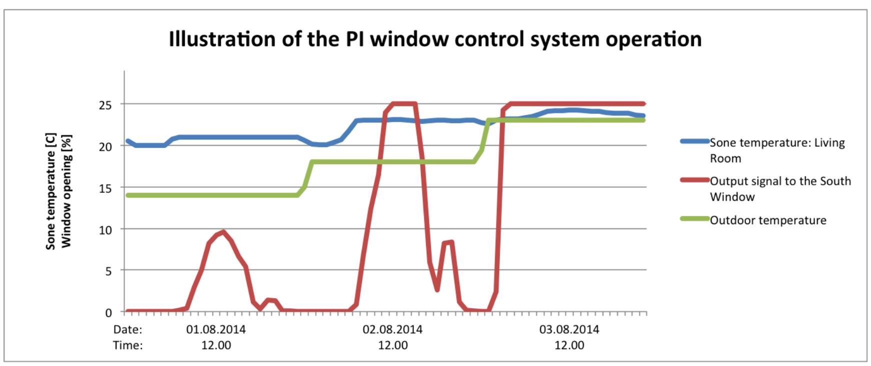


Figure 40: Illustration of the operation of a PI window control system in IDA ICE. Here: south window keep 21^{o}C when below 15^{o}C outside, keep 23^{o}C when 15-20^{o}C outside, keep 25^{o}C when above 20^{o}C outside

Exclusively daytime operation

It was assumed that indoor air temperatures in the lower range of the acceptable temperatures could be maintained when outdoor temperatures were below 15^{o}C. It was therefore decided to test set-points 22^{o}C, 21^{o}C and 20^{o}C for outdoor air temperatures under 15^{o}C. The results from the on/off window control simulations revealed that indoor temperatures could be kept under 24^{o}C when outdoor temperatures were below 20^{o}C. It was therefore decided to test the same set-point for the PI system. To see if a PI window control system could maintain lower indoor air temperatures, set-points 23^{o}C and 22^{o}C were also tested for outdoor temperatures between 15^{o}C and 20^{o}C. It was decided that the system should attempt to keep the indoor air temperatures below the maximum level of 26^{o}C when outdoor temperatures exceeded 20^{o}C. The lowest set-point and the highest set-point temperatures were combined for each outdoor temperature level. This was done to enable a more clear evaluation of the effect of the different system set-points. The results are presented in Table 28.

Table 28: Results for the first set-point temperature testing for the PI daytime window control solution. Time is given as percentage of the simulation period (768 hours).

	PI Day 22-24-26^{o}C		PI Day 21-23-26^{o}C		PI Day 20-22-26^{o}C	
	Result	Time [%]	Result	Time [%]	Result	Time[%]
Overheating						
Max temp Living [^{o}C]	34.77		34.65		34.59	
H Tliving >24^{o}C	118	15.4	27	3.5	34	4.4
H Tliving >26^{o}C	23	3.0	20	2.6	20	2.6
H Tliving >28^{o}C	13	1.7	11	1.4	11	1.4
Max temp Loft [^{o}C]	32.34		32.23		32.20	
H Tloft >26^{o}C	24	3.1	23	3.0	18	2.3
Max temp BedroomW [^{o}C]	33.63		33.51		33.47	
H TbedroomW >26^{o}C	28	3.6	24	3.1	24	3.1
H overheated	***19***	***2.5***	***17***	***2.2***	***16***	***2.1***
Overcooling						
Min temp Living [^{o}C]	19.91		19.75		18.73	
H Tliving <20^{o}C	43	5.6	57	7.4	173	22.5
Min temp Kitchen [^{o}C]	19.76		19.73		18.83	
H Tkitchen <20^{o}C	116	15.1	133	17.3	312	40.6
H overcooled	***58***	***7.6***	***211***	***27.5***	***308***	***40.1***
Air velocities						
H south window open	467	60.8	491	63.9	526	68.5
Average velocity [m/s]	0.84		0.79		0.74	
Average velocity 2-5% [m/s]	0.60		0.59		0.61	
Max velocity 2-25% [m/s]	1.34		1.84		1.85	
H >0.5m/s 2-25%	87	11.3	104	13.5	143	18.6
H >1.0m/s 5-25%	19	2.5	22	2.9	35	4.6
Average velocity <2% [m/s]	0.98		0.96		0.92	
Max velocity <2% [m/s]	2.01		1.99		2.00	
H > 1.0m/s <2%	102	13.3	89	11.6	68	8.9
H > 2.0m/s <2%	3	0.4	0	0.0	0	0.0

Table 28 shows that there were significant amounts of uncomfortable hours due to both overheating and overcooling. When taking a closer look at the results, it was found that all overheated hours had occurred between 08.00 and 10.00 on sunny weekends. The daytime PI window control system was activated when the occupants were present and awake. Hence, it did not begin to operate until 09.00 in the morning on weekends. Lack of window opening before this time resulted in overheating. Overheating occurred also after the system was activated. It took two hours of ventilative cooling to reduce the indoor air temperatures to an acceptable level. Overcooling was occurring due to simultaneous moderate indoor air temperatures and high air velocities.

Both the daytime active window control period and the set-point temperatures had to be adjusted to improve this control system. The daytime on/off window control simulations had revealed that active hours conceding with expected solar radiation (from 07.00 to 23.00) resulted in good thermal comfort. The second set-point testing of the daytime PI control system used the same active window control period to examine if it would result in better thermal conditions. To reduce the amount of overcooling in the building, the set-point temperatures were increased. Table 28 revealed that the maximum air velocity for the solution with the least amount of overcooling (22-24-26^{o}C) was 1.34m/s for large openings and 2.01m/s for small openings. Both air velocities corresponded to a temperature correction of approximately 3K. It was therefore decided that indoor air temperatures should not be below 23^{o}C when utilizing window openings for this system. The set-points tested for the second daytime PI window control simulations were to keep indoor temperatures below 23^{o}C for outdoor temperatures under 15^{o}C and below 23 or 24^{o}C for outdoor temperatures 15-20^{o}C. The results are presented in Table 29.

Table 29: Second set-point temperature testing for the PI daytime window control system. Time is given as percentage of the simulation period (768 hours).

	PI Day 23-24-26^{o}C		PI Day 23-23-26^{o}C	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	25.04		24.56	
H Tliving >24^{o}C	128	16.7	12	1.6
H Tliving >26^{o}C	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0
Max temp Loft [^{o}C]	25.53		25.50	
H Tloft >26^{o}C	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	26.41		25.48	
H TbedroomW >26^{o}C	14	1.8	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***
Overcooling				
Min temp Living [^{o}C]	19.90		19.89	
H Tliving <20^{o}C	33	4.3	41	5.3
Min temp Kitchen [^{o}C]	19.73		19.73	
H Tkitchen <20^{o}C	112	14.6	126	16.4
H overcooled	***5***	***0.7***	***8***	***1.0***
Air velocities				
H south window open	448	58.3	462	60.2
Average velocity [m/s]	0.75		0.73	
Average velocity 2-5% [m/s]	0.48		0.51	
Max velocity 2-25% [m/s]	1.19		1.84	
H >0.5m/s 2-25%	74	9.6	92	12.0
H >1.0m/s 5-25%	12	1.6	15	2.0
Average velocity <2% [m/s]	1.05		1.02	
Max velocity <2% [m/s]	2.04		2.02	
H > 1.0m/s <2%	74	9.6	67	8.7
H > 2.0m/s <2%	4	0.5	3	0.4

Table 29 shows that adjusting the active control period to concede with solar radiation eliminated overheating. Set-point solution 23-24-26^{o}C had the lowest amount of overcooled hours and was therefore considered the best solution. This solution caused indoor air temperatures almost exceeding 26^{o}C when windows were not operating. Indicating that the set-point temperatures should not be increased further. Considering 23-23-26^{o}C set-point solution caused more overcooling, the set-point temperatures should not be decreased either. The overcooled hours occurred when indoor temperatures were approximately 22.3^{o}C and air velocities in the occupied areas between 0.67m/s and 1.2m/s. This results in expected felt temperatures 19.3-19.8^{o}C which could be accepted for 0.7% of the time.

To illustrate the operation of the best set-point solution and the resulting thermal environment, the scenario with the biggest risk of overcooling is presented in Table 41 and 42. The scenario with the biggest risk of overheating is illustrated in Figure 43 and 44.

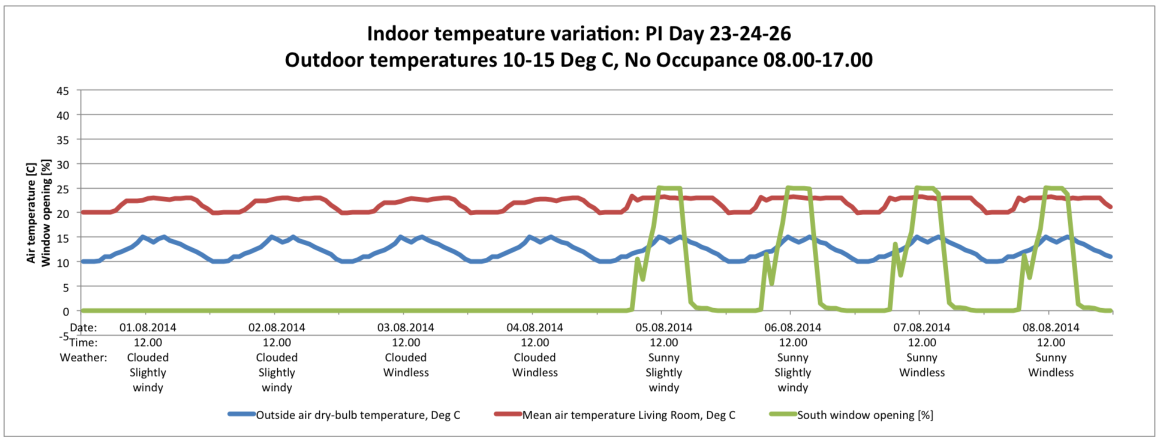


Figure 41: Window opening and indoor air temperature for the best PI daytime window set-point solution. Outdoor temperatures 10-15^{o}C, occupancy all hours except 08.00-17.00.

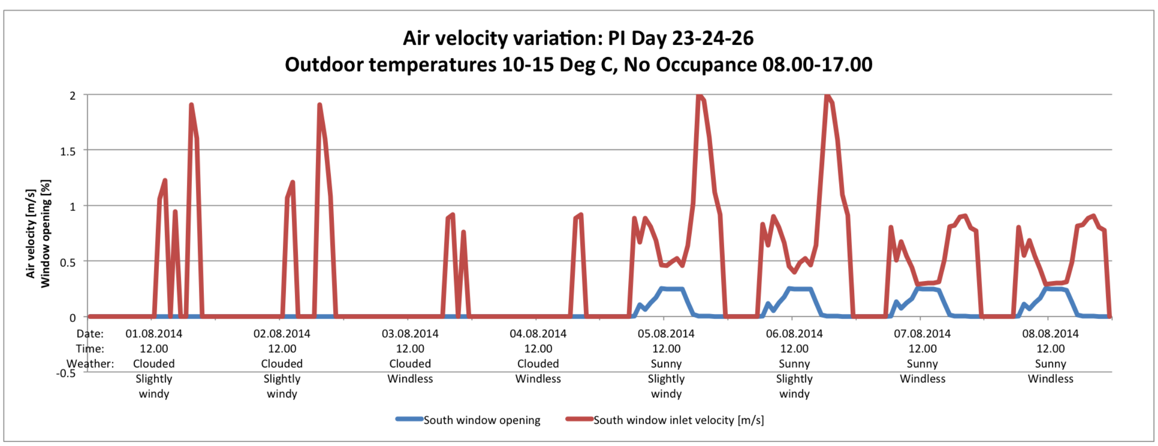


Figure 42: Window opening and indoor air velocities for the best PI daytime window set-point solution. Outdoor temperatures 10-15^{o}C, occupancy all hours except 08.00-17.00.

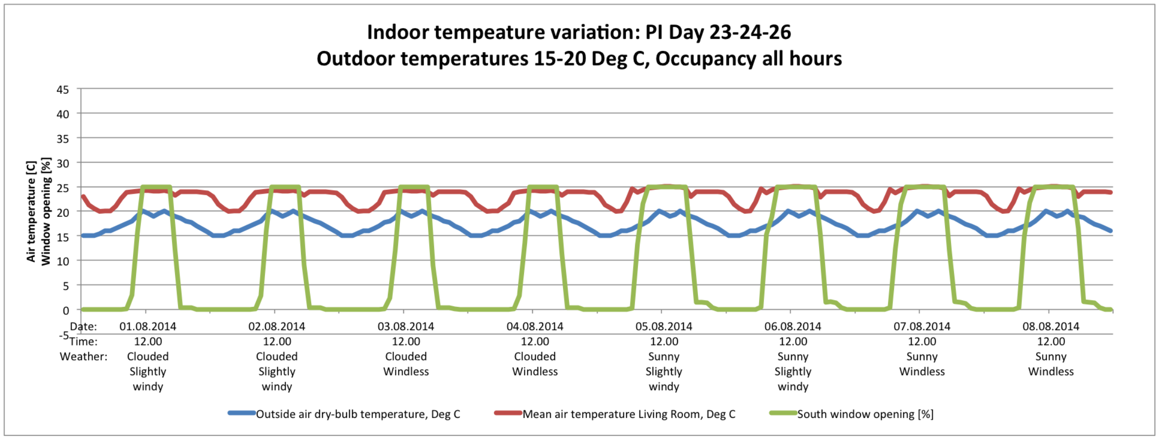


Figure 43: Window opening and indoor air temperature for the best PI daytime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

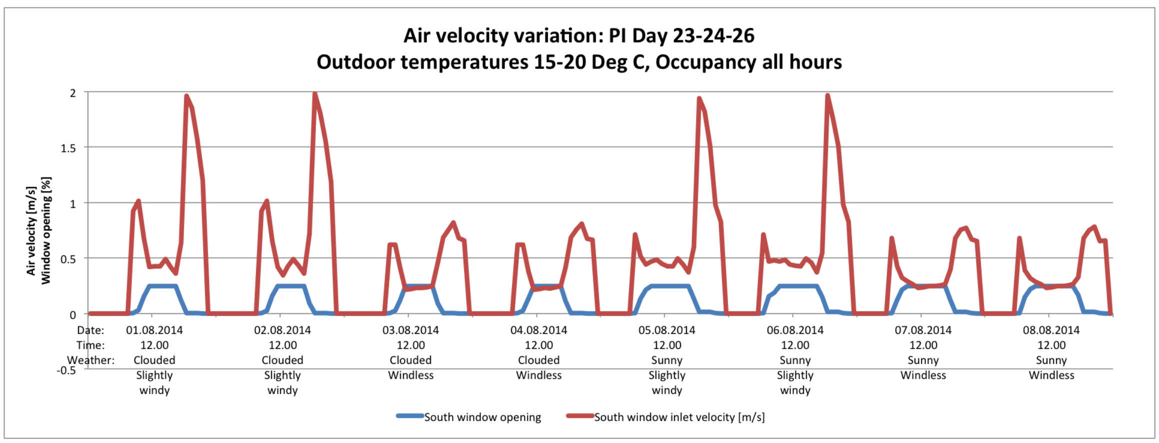


Figure 44: Window opening and indoor air velocities for the best PI daytime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

Exclusively nighttime operation

Although the on/off control system indicated that thermal storage ventilative cooling did not have a positive effect on the thermal environment, two set-point testings of the nighttime PI control system were performed. This was done to prove that this system obtained similar results. The first simulation used set-points to keep the temperatures below 20^{o}C during night, the second 19^{o}C. The results agreed with the findings from the nighttime on/off window control system. The PI solution resulted in more frequent use of window opening. However, reducing the indoor air temperature slightly at night had no positive effect on the daytime temperatures. The results are therefore not presented. They can however be found in the Appendix Section A.9.2

All hours opening

As for the on/off control system, it was decided that the all hour system would simulate the same window control system used for the daytime solution both day and night. The daytime control system had to adjust the active window period to concede with the hours where ventilative cooling was mostly needed. It was therefore not expected that an all hour system would significantly improve thermal comfort. Nevertheless, the simulations were conducted to see if an all hour system would result in equal or worse thermal conditions. The same set-point temperatures used for daytime control were tested. The results from the first testing are presented in Table 30.

Table 30: Results for the first set-point temperature testing for the PI all hours window control system. Time is given as percentage of the simulation period (768 hours).

	PI All hours 22-24-26^{o}C		PI All hours 21-23-26^{o}C		PI All hours 20-22-26^{o}C	
	Result	Time [%]	Result	Time [%]	Result	Time[%]
Overheating						
Max temp Living [^{o}C]	24.87		23.89		22.90	
H Tliving >24^{o}C	86	11.2	0	0.0	0	0.0
H Tliving >26^{o}C	0	0.0	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0	0	0.0
Max temp Loft [^{o}C]	26.66		26.11		25.58	
H Tloft >26^{o}C	6	0.8	4	0.5	0	0.0
Max temp BedroomW [^{o}C]	26.10		25.16		24.26	
H TbedroomW >26^{o}C	4	0.5	0	0.0	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***	***0***	***0.0***
Overcooling						
Min temp Living [^{o}C]	19.90		19.76		18.74	
H Tliving <20^{o}C	48	6.3	55	7.2	167	21.7
Min temp Kitchen [^{o}C]	19.75		19.73		19.40	
H Tkitchen <20^{o}C	116	15.1	134	17.4	333	43.4
H overcooled	***58***	***7.6***	***238***	***31.0***	***449***	***58.5***
Air velocities						
H south window open	481	62.6	517	67.3	686	89.3
Average velocity [m/s]	0.83		0.78		0.73	
Average velocity 2-5% [m/s]	0.58		0.57		0.59	
Max velocity 2-25% [m/s]	1.30		1.80		1.88	
H >0.5m/s 2-25%	78	10.2	108	14.1	166	21.6
H >1.0m/s 5-25%	16	2.1	16	2.1	30	3.9
Average velocity <2% [m/s]	0.97		0.94		0.87	
Max velocity <2% [m/s]	2.02		1.98		1.99	
H > 1.0m/s <2%	108	14.1	89	11.6	87	11.3
H > 2.0m/s <2%	5	0.7	0	0.0	0	0.0

The first set-point testing resulted in no overheating. This supports the statement that better thermal comfort is obtained when the system is not activated by occupancy. However, the results showed significant risk of overcooling. The set-point temperatures were therefore increased. The same set-points used for the second simulation of the daytime control system were used. The results are presented in Table 31.

Table 31: Results for the second set-point temperature testing for the PI all hours window control system. Time is given as percentage of the simulation period (768 hours).

	PI All hours 23-24-26^{o}C		PI All hours 23-23-26^{o}C	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	29.07		29.02	
H Tliving >24^{o}C	88	11.5	80	10.4
H Tliving >26^{o}C	48	6.3	51	6.6
H Tliving >28^{o}C	20	2.6	19	2.5
Max temp Loft [^{o}C]	25.51		25.21	
H Tloft >26^{o}C	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	29.47		29.46	
H TbedroomW >26^{o}C	52	6.8	48	6.3
H overheated	***20***	***2.6***	***19***	***2.5***
Overcooling				
Min temp Living [^{o}C]	19.92		19.92	
H Tliving <20^{o}C	33	4.3	41	5.3
Min temp Kitchen [^{o}C]	19.75		19.76	
H Tkitchen <20^{o}C	122	15.9	124	16.1
H overcooled	***5***	***0.7***	***11***	***1.4***
Air velocities				
H south window open	481	62.6	489	63.7
Average velocity [m/s]	0.75		0.75	
Average velocity 2-5% [m/s]	0.52		0.53	
Max velocity 2-25% [m/s]	1.94		1.91	
H >0.5m/s 2-25%	96	12.5	107	13.9
H >1.0m/s 5-25%	12	1.6	18	2.3
Average velocity <2% [m/s]	0.99		0.99	
Max velocity <2% [m/s]	2.03		2.03	
H > 1.0m/s <2%	70	9.1	67	8.7
H > 2.0m/s <2%	4	0.5	4	0.5

The higher set-point temperatures resulted in reduced overcooling compared to the lower temperature set-points. The amount of overheated hours had, however, significantly increased. Nevertheless, the combined number of uncomfortable hours were reduced. The increased set-point temperatures were therefore deemed a better solution than the lower temperature set-points. Using set-points 23-24-25^{o}C and 23-23-26^{o}C resulted in approximately the same risk of overheating. However, set-point solution 23-24-26^{o}C had a lower number of overcooled hours. This solution was therefore considered the best option. Using these set-points caused high maximum indoor temperatures and some overheating, supporting the choice that the set-point temperatures should not be further increased. Lowering the set-point temperatures would rapidly increase the risk of overcooling. Hence, the set-point temperatures should not be decreased either.

To illustrate the operation of the best set-point solution and the resulting thermal environment, the results for the scenario with the biggest risk of overcooling are presented in Figure 45 and 46. The scenario with the biggest risk of overheating is illustrated in Figure 47 and 48.

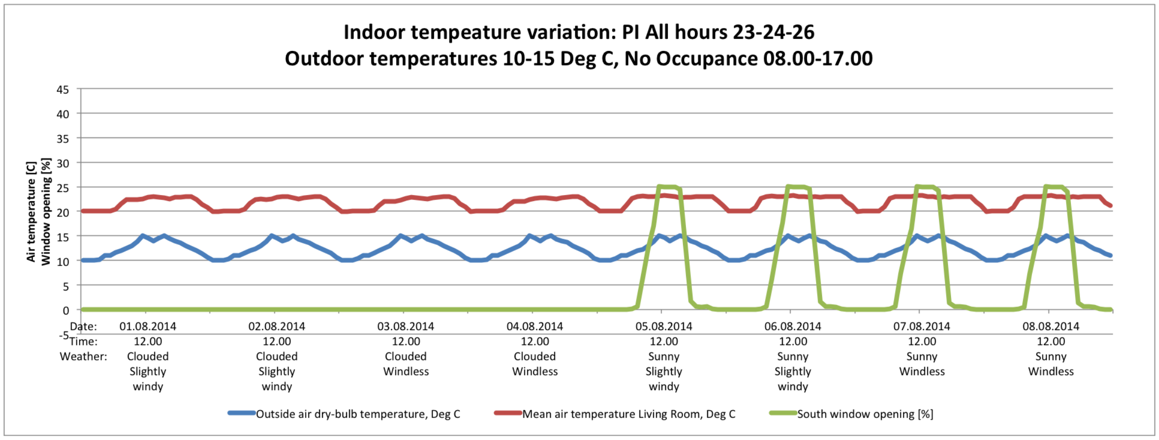


Figure 45: Window opening and indoor air temperature for the best PI all hours window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

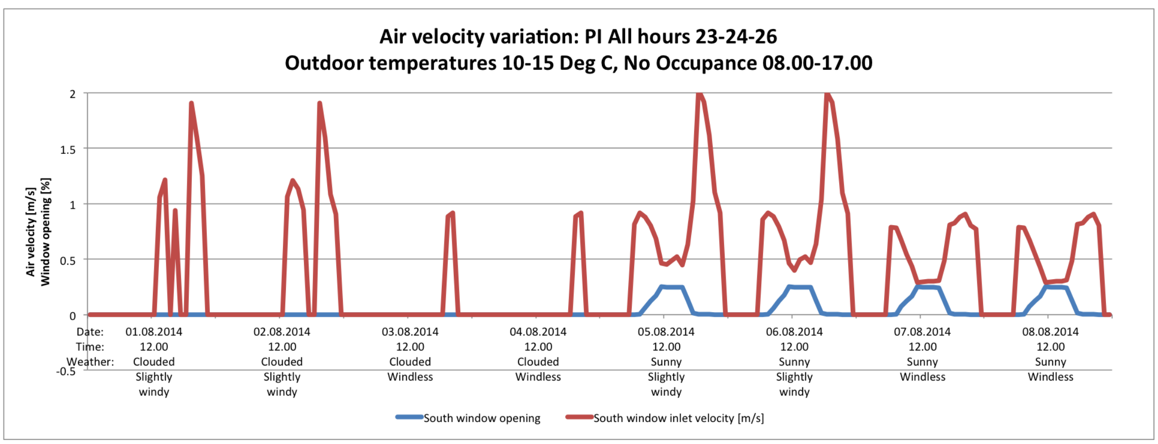


Figure 46: Window opening and indoor air velocities for the best PI all hours window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

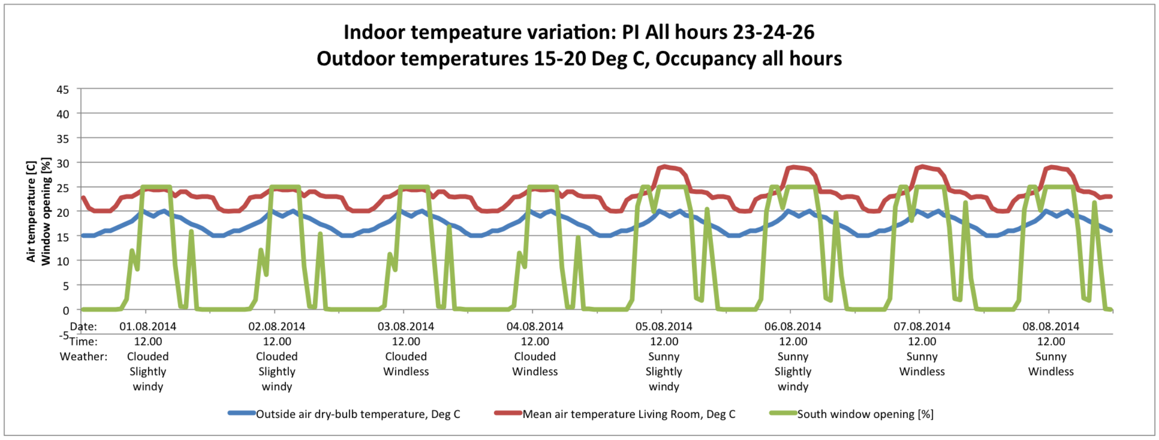


Figure 47: Window opening and indoor air temperature for the best PI all hours window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

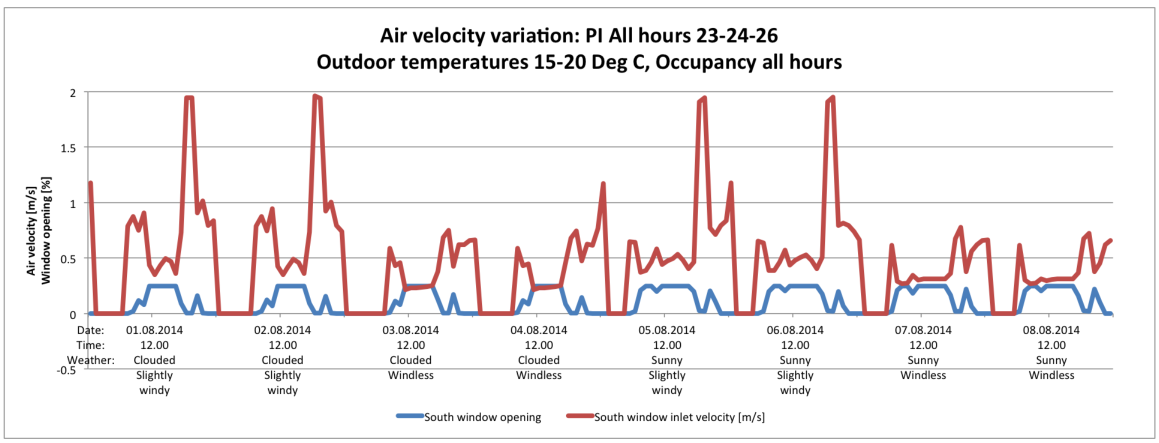


Figure 48: Window opening and indoor air velocities for the best PI all hours window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

4.1.3 Active window control period

On/off window control

To find out when the on/off window control systems should be active in operation, the results from the best set-point daytime and all hours on/off window control systems were compared. The comparison is presented in Table 32. Nighttime ventilative cooling had already been proven not to be applicable in the current building.

Table 32: Results for determination of the best on/off window control solution. Time is given as percentage of the simulation period (768 hours).

	On/off Day 24-22^{o}C		On/off All hours 24-22^{o}C	
	Result	Time [%]	Result	Time[%]
Overheating				
Max temp Living [^{o}C]	25.06		24.36	
H Tliving >24^{o}C	83	10.8	66	8.6
H Tliving >26^{o}C	0	0.0	0	0.0
H Tliving >28^{o}C	0	0.0	0	0.0
Max temp Loft [^{o}C]	25.73		25.83	
H Tloft >26^{o}C	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	25.67		25.64	
H TbedroomW >26^{o}C	0	0.0	0	0.0
H overheated	***0***	***0.0***	***0***	***0.0***
Overcooling				
Min temp Living [^{o}C]	19.97		19.97	
H Tliving <20^{o}C	49	6.4	50	6.5
Min temp Kitchen [^{o}C]	19.80		19.80	
H Tkitchen <20^{o}C	110	14.3	111	14.5
H overcooled	***1***	***0.1***	***0***	***0.0***
Air velocities				
H south window open	261	34.0	265	34.5
Average velocity [m/s]	0.39		0.39	
Average velocity 2-5% [m/s]	0.39		0.39	
Max velocity 2-25% [m/s]	0.84		0.85	
H >0.5m/s 2-25%	40	5.2	41	5.3
H >1.0m/s 5-25%	0	0.0	0	0.0
Average velocity <2% [m/s]	n.a.		n.a.	
Max velocity <2% [m/s]	0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0
H > 2.0m/s <2%	0	0.0	0	0.0

The results were almost identical. A on/off window control system would almost never operate outside of the time period 07.00-23.00. An all hours system was therefore not considered necessary. Since the results for both systems indicated good thermal comfort, the daytime solution was deemed the best solution. It is most likely to result in lower electricity use for the window control system and less frequent need for maintenance due to fewer active operating hours.

PI window control

To find out when PI window control should be active, the results from the best set-point daytime and all hours PI window control systems were compared. The comparison is presented in Table 32.

Table 33: Results for determination of the best PI window control solution. Time is given as percentage of the simulation period (768 hours).

	PI Day 23-24-26^{o}C		PI All hours 23-24-26^{o}C	
	Result	Time [%]	Result	Time[%]
Overheating				
Max temp Living [^{o}C]	25.04		29.07	
H Tliving >24^{o}C	128	16.7	88	11.5
H Tliving >26^{o}C	0	0.0	48	6.3
H Tliving >28^{o}C	0	0.0	20	2.6
Max temp Loft [^{o}C]	25.53		25.51	
H Tloft >26^{o}C	0	0.0	0	0.0
Max temp BedroomW [^{o}C]	26.41		29.47	
H TbedroomW >26^{o}C	14	1.8	52	6.8
H overheated	***0***	***0.0***	***20***	***2.6***
Overcooling				
Min temp Living [^{o}C]	19.90		19.92	
H Tliving <20^{o}C	33	4.3	33	4.3
Min temp Kitchen [^{o}C]	19.73		19.75	
H Tkitchen <20^{o}C	112	14.6	122	15.9
H overcooled	***5***	***0.7***	***5***	***0.7***
Air velocities				
H south window open	448	58.3	481	62.6
Average velocity [m/s]	0.75		0.75	
Average velocity 2-5% [m/s]	0.48		0.52	
Max velocity 2-25% [m/s]	1.19		1.94	
H >0.5m/s 2-25%	74	9.6	96	12.5
H >1.0m/s 5-25%	12	1.6	12	1.6
Average velocity <2% [m/s]	1.05		0.99	
Max velocity <2% [m/s]	2.04		2.03	
H > 1.0m/s <2%	74	9.6	70	9.1
H > 2.0m/s <2%	4	0.5	4	0.5

The results regarding overcooling and air velocities were found to be similar. The only substantial differences were found for the amount of hours below 20^{o}C in the kitchen and the maximum air velocities for large openings. Both results came out worse for the all hour solution.

The daytime solution prevented overheating. However, for the all hour solution, the risk of overheating was significantly increased. This solution resulted in high maximum indoor air temperatures and several hours with temperatures above 28^{o}C in the living room. It was unexpected that a window control system with more active operating hours would result in more overheating.

It was found that overheating for the all hours system mostly occurred on sunny days when outdoor temperatures were in the range of 15-20^{o}C. Since window control was allowed at all times, the windows were opened early in the morning. During the early morning hours there was a significant difference between outdoor and indoor temperature. The indoor air temperature was therefore quickly reduced, resulting in a reduced need for window opening. The window openings were therefore reduced at a time when both the outdoor temperature and the solar radiation was increasing. The result was a rapid increase of the indoor air temperatures. Once these high indoor temperatures were reached, it was difficult for the system to reduce them. Several hours with overheating occurred even when the window openings were increased again.

The daytime solution was, on the other hand, not activated before 07.00. The windows were therefore opened for the first time later in the morning. When they were opened, the need for cooling was only increasing. The result was constant window opening at the maximum level. Hence, the high maximum temperatures reached for the all hour solution was never reached for the daytime solution, resulting in zero hours of overheating.

Due to the unfavorable window operation causing overheating for the all hour system, the daytime system was deemed the best solution for the PI window control system.

4.1.4 Best window control solutions

An overview of the tested window control system solutions and the corresponding thermal comfort is presented in Table 34.

Table 34: Overview of the thermal comfort achieved for each tested window control solution

On/off exclusively daytime	25-20^{o}C Poor	24-20^{o}C Poor	23-20^{o}C Poor	25-22^{o}C Good	24-22^{o}C Best (chosen)
On/off exclusively nighttime	22-20^{o}C Poor	20-19^{o}C Poor			
On/off all hours	25-20^{o}C Poor	24-20^{o}C Poor	23-20^{o}C Poor	25-22^{o}C Good	24-22^{o}C Best
PI exclusively daytime	22-24-26^{o}C Poor	21-23-26^{o}C Poor	20-22-26^{o}C Poor	23-24-26^{o}C Best (chosen)	23-23-26^{o}C Good
PI exclusively nighttime	20-20-21^{o}C Poor	19-19-20^{o}C Poor			
PI all hours	22-24-26^{o}C Poor	21-23-26^{o}C Poor	20-22-26^{o}C Poor	23-24-26^{o}C Good	23-23-26^{o}C Good

Table 34 shows that the best thermal comfort was achieved for the on/off control system when using a daytime or all hour system with set-points 24-22^{o}C. The daytime system was the chosen solution. For the PI window control system the best solution was the daytime only system with set-points 23-24-26^{o}C. These two control systems were therefore used when constructing complete ventilative cooling solutions.

4.1.5 Factors most influencing the need for window opening

The window openings for all tested set-point solutions and all scenarios used were analyzed and compared. The comparison was made with regard to which scenarios caused most frequent window opening. The results are presented in Table 35. It was evident that windows were frequently opened for all set-point solutions on sunny days regardless of the occupancy, outdoor temperature and presence of wind. For cloudy days, windows were frequently used for all set-point solutions when outdoor temperatures were high and the building was occupied all hours. Only some of the set-point solutions resulted in frequent use of window openings when outdoor temperatures were high and the building was not occupied during the day. For cloudy days and low outdoor temperatures, windows were rarely open regardless of the occupancy schedule and wind.

Table 35: Window opening frequency for the different scenarios

	10-15^{o}C Occupancy exp.8-17	10-15^{o}C Occupancy all hours	15-20^{o}C Occupancy exp.8-17	15-20^{o}C Occupancy all hours
Clouded & Windy	Rarely	Rarely	Sometimes	Frequent
Clouded & Windless	Rarely	Rarely	Sometimes	Frequent
Sunny & Windy	Frequent	Frequent	Frequent	Frequent
Sunny & Windless	Frequent	Frequent	Frequent	Frequent

The factors most influencing the need for ventilative cooling in Living Lab were therefore in the following order: solar radiation, outdoor temperature and occupancy. Presence of wind did not seem to significantly influence the need for ventilative cooling. However, when comparing the air velocities when windows were fully open, it was clear that the presence of wind resulted in higher air velocities in the building. Hence, it was influential to the cooling effect.

4.2 How to apply ventilative cooling

4.2.1 Scenario used for the simulations

All complete ventilative solutions were simulated using a complete set of outdoor conditions for a whole year. This was necessary to acquire sufficient results for the evaluation of how to apply ventilative cooling. Climate data from 2014 were used for the whole year simulations. It was the most recent year with complete weather data. The weather station closest to the building is located at Værnes, only 32.2km from the building site. The climatic file from this station was therefore used. The weekday pattern was applied from Monday to Friday and the weekend pattern from Saturday till Sunday.

4.2.2 Natural, concurrent or change-over ventilative ventilative cooling

Natural ventilation was studied without use of mechanical ventilation in the previous simulations. Hence, the mechanical fans in the central AHU were turned off. The same applied when evaluating natural ventilation only as a complete ventilative cooling solution.

For the remaining simulations, the window control systems were combined with mechanical ventilation in concurrent and change-over mixed-mode systems. For the concurrent systems, the best on/off and PI window control systems were combined with constant hygienic mechanical ventilation. This level of mechanical ventilation was already implemented in IDA ICE. Hence, the only difference from the natural ventilation simulations was that all fans in the central AHU were turned on.

For the change-over systems, the best on/off and PI window control systems were combined with hygienic mechanical ventilation with the ability to decouple when ventilative cooling was utilized. All fans in the central AHU were turned on and an additional control system was constructed to turn off mechanical ventilation when windows were open. The central AHU unit in IDA ICE was not able to retrieve window signals. Therefore, the on/off mechanical ventilation system had to be implemented for each sone. It was designed so that if one or more windows in the current zone was open, the mechanical ventilation in that particular zone would be turned off. Hence, a change-over zoned system was created. A ventilation system with this design does no longer ensure a balanced mechanical ventilation system.

The construction of the on/off mechanical ventilation control system is presented in Figure 71. An example illustrating the operation of the control system is given in Figure 71. It shows that the airflow was held constant at the given level when the window was closed, and turned off when the window was open.

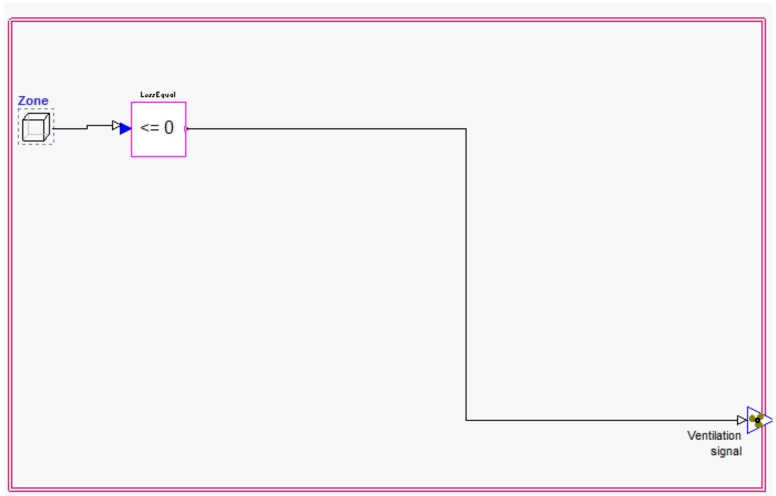


Figure 49: Illustration of the construction of the mechanical ventilation on/off control system in IDA ICE.

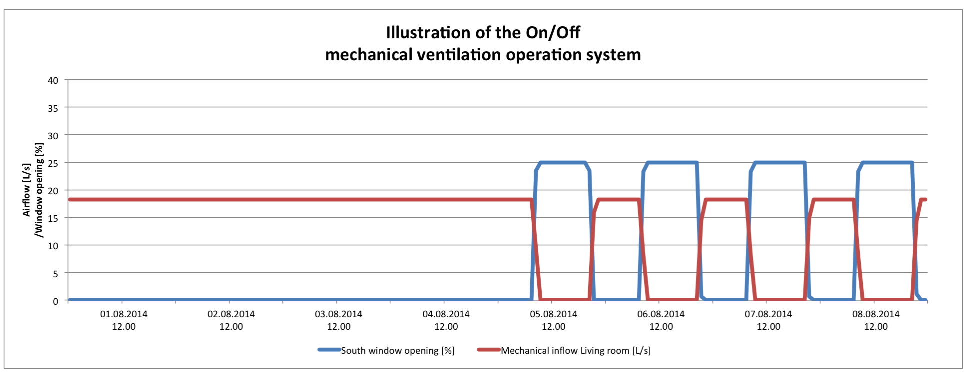


Figure 50: Illustration of the operation of the mechanical ventilation on/off control system in IDA ICE. Here: living room supply.

Ventilative cooling using on/off window control

The results from the whole year simulations of the three ventilative cooling solutions utilizing on/off window control are presented in Table 36.

The results show that all three solutions resulted in good thermal comfort. The results regarding overheating and overcooling were similar. The mixed-mode systems obtained only a few less hours of overheating and slightly more overcooling than the natural ventilation system. When there was risk of overheating, the indoor temperatures were high. When mechanical ventilation with low supply temperature was applied in this situation, the thermal conditions were slightly improved. However, when there was risk of overcooling the indoor temperatures were low. When mechanical ventilation with low supply temperature was applied in this situation, the thermal conditions were slightly worsen.

Table 36: Results from whole year simulations using on/off window control system.

	Natural ventilation: On/off window control		Concurrent: On/off window control		Change-over: On/off window control	
	Result	Time [%]	Result	Time [%]	Result	Time[%]
Overheating						
Max temp Living [^{o}C]	30.93		30.87		30.88	
H Tliving >24^{o}C	330	3.8	257	2.9	264	3.0
H Tliving >26^{o}C	40	0.5	37	0.4	38	0.4
H Tliving >28^{o}C	19	0.2	19	0.2	19	0.2
Max temp Loft [^{o}C]	32.43		32.40		32.38	
H Tloft >26^{o}C	71	0.8	65	0.7	67	0.8
Max temp BedroomW [^{o}C]	32.43		33.00		33.04	
H TbedroomW >26^{o}C	71	0.8	59	0.7	60	0.7
H overheated	***15***	***0.2***	***13***	***0.1***	***13***	***0.1***
Overcooling						
Min temp Living [^{o}C]	19.24		18.03		17.99	
H Tliving <20^{o}C	2499	28.5	2606	29.7	2676	30.5
Min temp Kitchen [^{o}C]	18.99		17.79		17.75	
H Tkitchen <20^{o}C	4681	53.4	4887	55.8	4931	56.3
H overcooled when VC	30	0.3	30	0.3	31	0.4
H overcooled	***36***	***0.4***	***48***	***0.5***	***54***	***0.6***
Air velocities						
H south window open	1160	13.2	911	10.4	948	10.8
Average velocity [m/s]	0.53		0.53		0.53	
Average velocity 2-5% [m/s]	0.53		0.53		0.53	
Max velocity 2-25% [m/s]	1.76		1.75		1.78	
H >0.5m/s 2-25%	493	5.6	386	4.4	399	4.6
H >1.0m/s 5-25%	63	0.7	54	0.6	54	0.6
Average velocity <2% [m/s]	0.00		0.00		0.00	
Max velocity <2% [m/s]	0.00		0.00		0.00	
H > 1.0m/s <2%	0	0.0	0	0.0	0	0.0
H > 2.0m/s <2%	0	0.0	0	0.0	0	0.0
Energy						
Recovery (free) [kWh]	0.0		4798.4		4769.1	
Zone heating [kWh]	4844.0		5470.0		5489.1	
AHU heating [kWh]	0.0		241.9		244.8	
Fan energy [kWh]	0.0		1094.5		1051.6	
Lighting [kWh]	1083.0		1082.8		1083.0	
Equipment [kWh]	1116.0		1115.8		1116	
Energy demand [kWh]	***7037.7***		***9005.0***		***8984.5***	

The amount of overheating was the same for the two mixed-mode systems. The change-over mixed-mode system resulted, however, in slightly more hours of overcooling compared to the concurrent zoned system. The change-over zoned system was designed to only turn off the mechanical ventilation units in the rooms that had open windows. Hence, unbalancing the mechanical ventilation and creating a more unpredictable and unstable system.

When comparing energy demand for the three solutions, it was evident that the solution utilizing natural ventilation had substantial lower energy demand than the mixed-mode systems. This was mostly due to the lack of AHU heating and energy use for fans. This solution also had the lowest energy demand for zone heating even though no heat recovery was obtained. This was due to lack of mechanical ventilation supplying cold air to the rooms during heating periods.

The differences in energy demand for the mixed-mode systems were not significant. The change-over system resulted in only slightly less energy demand than the concurrent system. Mechanical ventilation was always in operation for the concurrent system. This should have resulted in a bigger increase in energy use for fans compared to the change-over system. However, since the change over system was designed as a zoned system, it only turned off the mechanical units in some of the rooms 10.8% of the time. The difference in energy use for fan operation was therefore not substantial.

Even though it was not included in the results, the solution utilizing natural ventilation is expected to have the lowest indoor air quality. Natural ventilation was only utilized 13.2% of the time. For the remaining parts of the year there would be no air exchange except from infiltration. Infiltration is expected to be low in a highly insulated building like Living Lab. The concurrent solution is expected to have the best indoor air quality because it operates mechanical ventilation constantly. It is also considered to be less complex than the change-over system. This is because the window control system and the mechanical ventilation system operate independently. Also, the concurrent system does not jeopardize the balancing of the mechanical ventilation system. Since all solutions provided good thermal comfort and the differences in energy demand for the mixed-mode systems were not significant, the concurrent system solution was deemed the best ventilative cooling solution utilizing on/off window control.

Ventilative cooling using PI window control

The results from the whole year simulations of the three ventilative cooling solutions utilizing PI window control are presented in Table 37.

The results regarding overheating were similar to those obtained when using on/off window control. There were very few hours of overheating and they were approximately the same for all solutions. However, different results regarding overcooled hours were obtained. The results show that the risk of overcooling was significantly increased when utilizing the PI window control system. Also, the mixed-mode systems resulted in less overcooling compared to use of natural ventilation only. Since the window control system had a weaker design, applying mechanical ventilation system improved the situation. Use of mechanical ventilation reduced the need for natural ventilation, hence reducing the amount of overcooled hours.

Table 37: Results from whole year simulations using the PI window control system.

	Natural ventilation: PI window control		Concurrent: PI window control		Change-over: PI window control	
	Result	Time [%]	Result	Time [%]	Result	Time[%]
Overheating						
Max temp Living [^{o}C]	30.91		30.84		30.92	
H Tliving >24^{o}C	418	4.8	369	4.2	373	4.3
H Tliving >26^{o}C	32	0.4	31	0.4	31	0.4
H Tliving >28^{o}C	16	0.2	16	0.2	16	0.2
Max temp Loft [^{o}C]	32.53		32.48		32.48	
H Tloft >26^{o}C	84	1.0	68	0.8	68	0.8
Max temp BedroomW [^{o}C]	33.08		32.97		33.01	
H TbedroomW >26^{o}C	96	1.1	71	0.8	71	0.8
H overheated	***13***	***0.1***	***12***	***0.1***	***12***	***0.1***
Overcooling						
Min temp Living [^{o}C]	19.24		17.99		17.99	
H Tliving <20^{o}C	2587	29.5	2692	30.7	2750	31.4
Min temp Kitchen [^{o}C]	18.97		17.74		17.75	
H Tkitchen <20^{o}C	4750	54.2	4946	56.5	5000	57.1
H overcooled when VC	126	1.4	112	1.3	80	0.9
H overcooled	***130***	***1.5***	***128***	***1.5***	***103***	***1.2***
Air velocities						
H south window open	2163	24.7	1827	20.9	1769	20.2
Average velocity [m/s]	1.10		1.07		1.05	
Average velocity 2-5% [m/s]	0.81		0.80		0.81	
Max velocity 2-25% [m/s]	4.05		3.14		4.00	
H >0.5m/s 2-25%	695	7.9	580	6.6	621	7.1
H >1.0m/s 5-25%	233	2.7	200	2.3	214	2.4
Average velocity <2% [m/s]	1.31		1.28		1.28	
Max velocity <2% [m/s]	4.47		4.45		4.41	
H > 1.0m/s <2%	830	9.5	688	7.9	615	7.0
H > 2.0m/s <2%	130	1.5	88	1.0	86	1.0
Energy						
Recovery (free) [kWh]	0.0		4808.50		4137.4	
Zone heating [kWh]	4862.0		5470.4		6038.3	
AHU heating [kWh]	0.0		243.0		177.9	
Fan energy [kWh]	0.0		1094.9		943.7	
Lighting [kWh]	1083.1		1083.0		1083.0	
Equipment [kWh]	1116.0		1116.0		1116.0	
Energy demand [kWh]	***7060.9***		***9007.0***		***9358.9***	

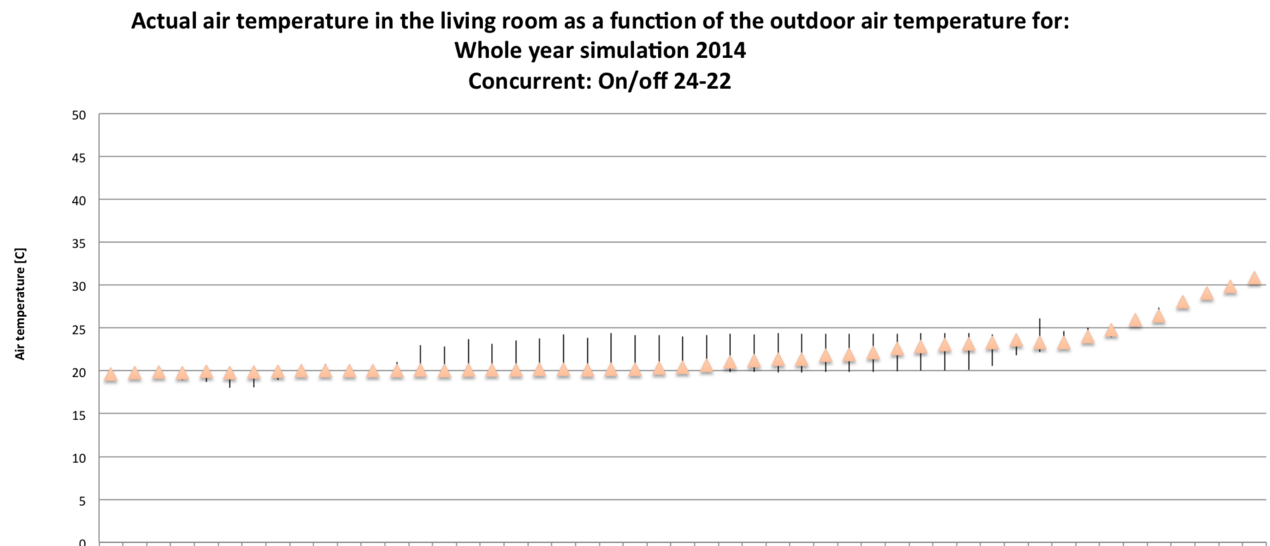
As for the on/off systems, the natural ventilation system obtained the lowest energy demand of the three solutions. The difference in energy demand for the two mixed-mode systems was not significant in this case either. However, opposed to the on/off solutions, the change-over system obtained slightly higher energy demand than the concurrent system. In this case, the increase in energy use for zone heating was bigger than the decrease in energy use for fans.

The concurrent system was also deemed the best solution for ventilative cooling utilizing PI window control. All three solutions provided approximately equally thermal comfort and the differences in energy demand between the mixed-mode systems were not significant. As explained for the on/off solutions, the concurrent system is expected to provide better indoor air quality. It is also less complex than the change-over system.

4.2.3 Best ventilative cooling solution

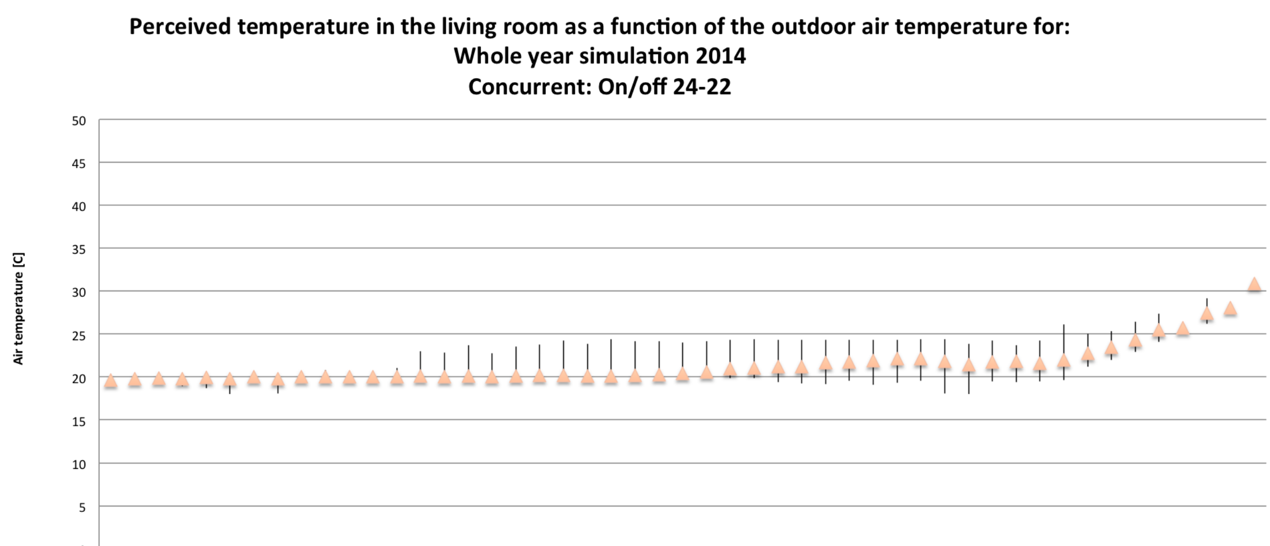
It is evident that ventilative cooling utilizing the on/off window control system would provide better thermal conditions than ventilative cooling utilizing the PI control system. Ventilative cooling using on/off window control would prevent overheating, but not so much at the expense of overcooling. Hence, the best solution for ventilative cooling in Living Lab is a concurrent mixed-mode solution utilizing a on/off window control system. The window control system should be designed to open the south and skylight windows to maximum opening when indoor air temperatures exceed 24^{o}C and close them when 22^{o}C is reached.

To give a better illustration of the thermal conditions that could be expected in Living Lab if applying this control system, the results from the simulations are presented in two figures. Both figures illustrate the different indoor temperatures that could be expected in the living room for various outdoor temperatures. Figure 51 presents the actual indoor temperatures as a function of outdoor temperatures. Figure 52 presents the indoor temperatures as expected to be perceived when taking the air velocities into account. The same temperature corrections described in Section 3.3.3 were applied. Both figures show that good thermal comfort will be achieved if this solution is implemented.



	-18	-17	-16	-15	-14	-13	-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Maximum temperature	20	20	20	20	20	20	20	20	21	21	20	20	21	23	23	24	23	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	26	25	25	25	26	27	29	30	30	31
Minimum temperature	20	20	20	19	19	18	18	19	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	21	22	22	23	23	24	26	26	27	29	30	31
Average temperature	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	21	21	21	21	21	22	22	22	23	23	23	23	23	24	23	23	24	25	26	26	28	29	30	31

Figure 51: Indoor air temperatures presented as a function of the outdoor air temperature for the best ventilative cooling solution



	-18	-17	-16	-15	-14	-13	-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Maximum temperature	20	20	20	20	20	20	20	20	21	21	20	20	21	23	23	24	23	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	26	25	25	26	27	26	29	28	31
Minimum temperature	20	20	20	19	19	18	20	18	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	19	19	19	20	19	19	20	18	18	19	19	20	20	21	22	23	24	25	26	28	31
Average temperature	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	21	21	21	21	21	22	22	22	22	22	22	21	22	22	22	22	23	23	24	25	26	27	28	31

Figure 52: Expected perceived indoor air temperatures presented as a function of the outdoor air temperature for the best ventilative cooling solution

4.3 The effect of ventilative cooling

The six solutions from the previous section were compared to two reference cases for the evaluation of the effect of ventilative cooling. The first reference case represented hygienic mechanical ventilation and the second enhanced mechanical ventilation. For the reference cases, all windows were held closed.

4.3.1 Compared to hygienic or enhanced mechanical ventilation

As described in Section 3.2.6, the hygienic mechanical ventilation supplies and extracts constantly 144m^3/h. Enhanced mechanical ventilation has the ability to supply and extract up to 260m^3/h. The intended use of enhanced mechanical ventilation is when additional extraction is needed due to cooking and frequent use of showering. However, this level could also be used when increased air exchange is needed due to high indoor air temperatures.

23^{o}C was assumed to be a reasonable set-point for switching from hygienic to enhanced mechanical ventilation. Hence, a ventilation control system was designed in IDA ICE to supply and extract 144m^3/h when indoor air temperatures were below 23^{o}C and 260m^3/h when indoor air temperatures exceeded 23^{o}C. The central AHU unit in IDA ICE was not able to retrieve indoor temperature signals. Individual control systems had to be defined for each zone with ventilation ducts. A ventilation system with this design would not ensure a balanced mechanical ventilation system. It was decided that the enhanced level of ventilation should be allocated to the different zones with the same weighing as for the hygienic mechanical ventilation. The initial ventilation airflow rates in the base model of Living Lab had to be changed to acquire these airflow rates. The living room and the two bedrooms were assigned a maximum of 10L/sm^2 supply air and 0L/sm2 extract air. For the bathroom and kitchen a maximum of 0L/sm2 supply air was assigned, and 10L/sm2 extract air. Further, the individual control systems were designed to send out a signal between 0 and 1 resulting in the right amount of supply or extraction for that particular zone when multiplied with 10L/sm2. An example of the construction of a enhanced mechanical ventilation control system in IDA ICE is illustrated in Figure 71. A more detailed description of how the control system works is provided in the Appendix Section A.8.3. Figure 54 illustrates the operation of the control system.

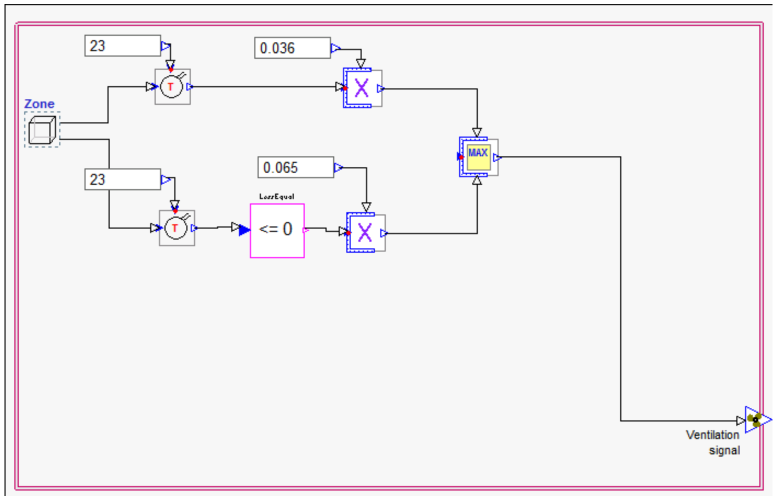


Figure 53: Example illustrating the construction of the enhanced mechanical ventilation control system in IDA ICE. Here: living room supply.

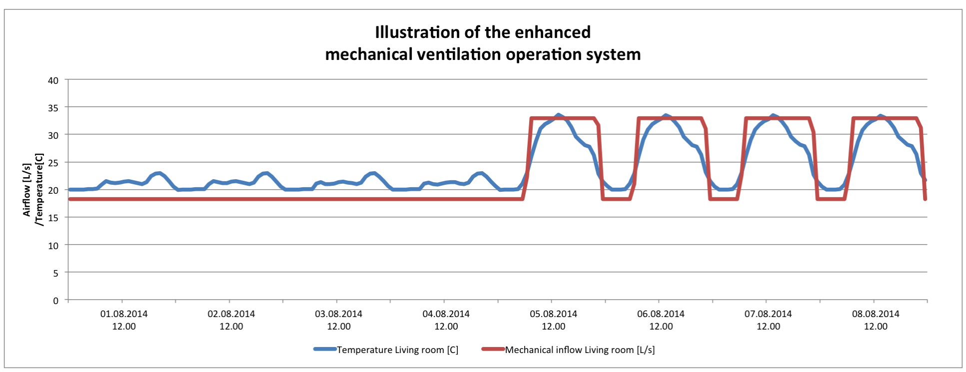


Figure 54: Example illustration of the operation of the enhanced mechanical ventilation control system in IDA ICE. Here: living room supply air.

Table 38 presents the results from the whole year simulations of the two reference cases. The results show that high indoor air temperatures would frequently occur when windows were not utilized (neither manually or automatically) to cool the indoor air. The situation was slightly improved when utilizing enhanced ventilation airflows. However, unacceptable thermal conditions were frequent for this case also. Both cases resulted in few hours of uncomfortably low temperatures. Hence, there was little risk of overcooling when only utilizing mechanical ventilation. The results regarding energy demand were found to be similar for both cases. The energy demand for the enhanced mechanical ventilation case was only slightly higher than for the hygienic mechanical ventilation case. This was due to increased energy use for fan operation and AHU heating. Also, more frequent air exchanges resulted in a small increase in energy use for zone heating.

Table 38: Results from the whole year simulations using mechanical ventilation only.

	Mechanical ventilation: Hygienic		Mechanical ventilation: Enhanced	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	44.99		42.25	
H Tliving >24^{o}C	1924	22.0	1657	18.9
H Tliving >26^{o}C	1465	16.7	1096	12.5
H Tliving >28^{o}C	1061	12.1	694	7.9
Max temp Loft [^{o}C]	47.93		45.79	
H Tloft >26^{o}C	1619	18.5	1354	15.5
Max temp BedroomW [^{o}C]	46.74		43.91	
H TbedroomW >26^{o}C	1438	16.4	1082	12.4
H overheated	***1420***	***16.2***	***1057***	***12.1***
Overcooling				
Min temp Living [^{o}C]	18.01		18.02	
H Tliving <20^{o}C	2630	30.0	2637	30.1
Min temp Kitchen [^{o}C]	17.76		17.78	
H Tkitchen <20^{o}C	4875	55.7	4870	55.6
H overcooled when VC	0	0.0	0	0.0
H overcooled	***19***	***0.2***	***19***	***0.2***
Energy				
Recovery (free) [kWh]	4810.3		5108.7	
Zone heating [kWh]	5418.3		5450.00	
AHU heating [kWh]	241.4		252.8	
Fan energy [kWh]	1094.6		1099.0	
Lighting [kWh]	1082.9		1083.0	
Equipment [kWh]	1116.0		1116.0	
Energy demand [kWh]	***8953.2***		***9000.8***	

Ventilative cooling using on/off window control

The ventilative cooling solutions utilizing on/off window control are compared to the reference cases in Table 39.

The results show that all three solutions almost eliminated the overheating that occurred when not applying natural or mixed-mode ventilation. Reductions of approximately 99% were recorded for all three solutions. The increase in overcooling was below 36 hours for all solutions. When combining the amount of overheated and overcooled hours, the total hours with uncomfortable thermal conditions were obtained. The results revealed reductions in uncomfortable hours over 93% for all ventilative cooling solutions. The effect of ventilative cooling on the thermal conditions was therefore positive and significant.

When only natural ventilation was utilized, a reduction in energy demand compared to mechanical ventilation was recorded due to zero use of fan energy or AHU heating. Also, a significant reduction in zone heating was obtained due to lack of cold supply air during heating periods. The effect of mixed-mode ventilative cooling on the energy use turned out to be small. A slight increase was recorded when compared to hygienic mechanical ventilation due to increased heating demand. When compared to enhanced mechanical ventilation, a small reduction in energy use was recorded for change-over solution. For the concurrent system, a marginal increase was recorded.

Table 39: Comparing results from the whole year simulations using the on/off window control system with use of mechanical ventilation only.

	Natural ventilation: On/off		Concurrent: On/off		Change-over: On/off	
	Increase [/year]	Increase [%]	Increase [/year]	Increase [%]	Increase [/year]	Increase [%]
Compared to hygienic mechanical ventilation						
Overheating [h]	-1405	-98.9	-1407	-99.1	-1407	-99.1
Overcooling [h]	17	89.5	29	152.6	35	184.2
Uncomfortable thermal condition [h]	-1338	-96.5	-1378	-95.8	-1372	-95.3
Energy demand [kWh]	-1915.5	-21.4	52	0.6	31.3	0.3
Compared to enhanced mechanical ventilation						
Overheating [h]	-1042	-98.6	-1044	-98.8	-1044	-98.8
Overcooling [h]	17	89.5	29	152.6	35	184.2
Uncomfortable thermal conditions [h]	-1025	-95.3	-1015	-94.3	-1009	-93.8
Energy demand [kWh]	-1963.1	-21.8	4	0.0	-16.3	-0.2

The results show that when utilizing a well designed window control system, ventilative cooling would significantly improve the thermal conditions without having a significantly negative effect on the energy use.

Ventilative cooling using PI window control

The ventilative cooling solutions utilizing PI window control are compared to the reference cases in Table 39.

The results regarding overheating were similar to the results obtained for the on/off window control system. Reductions of approximately 99% were recorded here as well. However, the increase in overcooling was above 83 hours for all solutions, hence fairly substantial. Nevertheless, the total amount of hours with uncomfortable thermal conditions were reduced with over 85% for all ventilative cooling solutions. The effect of these ventilative cooling solutions on thermal comfort was therefore also positive.

When only natural ventilation was utilized, a substantial reduction in energy use compared to mechanical ventilation was obtained. The effect of mixed-mode ventilative cooling on the energy use was not significant for these solutions either. Only a slight increase was recorded.

Table 40: Comparing results from the whole year simulations using the PI window control system with use of mechanical ventilation only.

	Natural ventilation PI		Concurrent: PI		Change-over: PI	
	Increase [/year]	Increase [%]	Increase [/year]	Increase [%]	Increase [/year]	Increase [%]
Compared to hygienic mechanical ventilation						
Overheating [h]	-1407	-99.1	-1408	-99.2	-1408	-99.2
Overcooling [h]	111	584.2	109	573.7	84	442.1
Uncomfortable thermal conditions [h]	-1296	-90.1	-1299	-90.3	-1324	-92.0
Energy demand [kWh]	-1892.3	-21.1	54	0.6	405.7	4.5
Compared to enhanced mechanical ventilation						
Overheating [h]	-1044	-98.8	-1045	-98.9	-1045	-98.9
Overcooling [h]	111	584.2	109	573.7	84	442.1
Uncomfortable thermal conditions [h]	-993	-86.7	-936	-87.0	-961	-89.3
Energy demand [kWh]	-1939.9	-21.6	7	0.1	358.1	4.0

The results show that when utilizing a PI window control system, ventilative cooling would improve the thermal conditions without having a significantly negative effect on the energy use.

5 Discussion

The following section provides a discussion of the results presented in Section 4. The main findings are listed and explained. Also, possible sources of error are discussed. The results from Living Lab provide indications on the applicability of ventilate cooling in general. However, the results have to be seen in a broader perspective to draw general conclusions. The main findings are therefore compared to previous findings from the literature survey presented in Section 2.3.

5.1 How to apply ventilative cooling

5.1.1 The evaluation process

It was decided that a mixed-mode concurrent system utilizing a on/off window control system programmed to open at 24^{o}C and close at 22^{o}C would be the best solution for Living Lab. However, the evaluation only included some aspects of the thermal environment. Local temperature variations were not evaluated. Neither was the air velocities from the mechanical system or from other windows than the south window. Humidity was also omitted from the study. Also, the evaluation of air velocities was only based on estimates. The evaluation of indoor air quality and complexity was based on assumptions. If the omitted aspects of thermal comfort had been included and a more in depth study of air velocities, draught situations, air quality and complexity had been conducted, the outcome regarding the best solution could have been different.

When evaluating energy demand, energy for domestic hot water and pumps were omitted from the evaluation. This resulted in a total energy demand less than what should be expected. However, the energy demand for hot water and pumps would have been the same for all solutions. Including these aspects would therefore not have changed the outcome.

5.1.2 The design process

Only a few solutions of the many that were tested were suitable for Living Lab. The best solutions resulted in acceptable or good thermal conditions. The weaker solutions resulted in poor thermal comfort. Living Lab is initially exposed to overheating. If a poorly designed window control system is not operating the windows frequently enough, sufficient cooling will not be supplied and the high indoor air temperatures will not be reduced to a comfortable level. On the other hand, the outdoor air temperatures are often low in Trondheim and the wind velocities are often high. If a poorly designed window control system is causing window operation at the wrong time or too often, the low supply temperatures and the increased air velocities are likely to cause overcooling.

These results indicate that a careful design process is needed to develop a suitable ventilative cooling system. It also indicates that a weak design is likely to cause severe risks of uncomfortable warm and/or cold indoor temperatures. Pellegrini et al. (2012) studied the application of the same ventilative cooling systems in similar buildings in different climates. Good thermal comfort was obtained in some of the cases while others obtained poor thermal comfort. Hence, supporting the statement that a careful design process is required and showing that the process should be individual for each building and climate.

5.1.3 Set-point evaluation

The simulations performed to determine how to apply window control in Living Lab were conducted without mechanical ventilation. This was done to limit the study and to distinct the effect of the window control systems. If set-point analyses had been conducted also for the concurrent and change-over systems, the decision process would have been more complete. It might also have resulted in the mixed-mode systems obtaining better thermal comfort. However, it is evident that the chosen concurrent system will provide good thermal comfort in Living Lab. With the chosen set-points, the opening temperature is low enough to keep the temperatures in all rooms below the maximum acceptable temperature for most of the time. The closing temperature is high enough not to cause frequent overcooling. Also, the difference between the opening and closing temperature is big enough not to cause the windows to open and close too frequently.

5.1.4 Nighttime ventilative cooling

The results revealed that nighttime ventilative cooling had no positive effect on the thermal environment in Living Lab. The indoor temperatures could not be significantly reduced at night because the building was occupied at this time. Living Lab is a lightweight building. It does not contain enough thermal mass for a small reduction of indoor temperatures at nighttime to have a significant effect on the daytime temperatures.

This study is therefore indicating that thermal storage ventilative cooling is not a good solution for low-energy dwellings. However, several studies reviewed in the literature survey stated the opposite. Previous studies have concluded that overheating in low-energy dwellings can be prevented with use of nighttime ventilative cooling in combination with strict solar shading (Thomsen et al., 2005), (Mlakar and Strancar, 2011) or in combination with increased air velocities during daytime. (Pellegrini et al., 2012). The most obvious explanation for the differences in result is that these buildings contained more thermal mass than Living Lab. Other factors could be different climates, different design of the nighttime ventilative cooling system and different combinations of other passive solutions. Nevertheless, it is evident that even though nighttime ventilative cooling might not be applicable in lightweight dwellings, it can be effective in buildings with more thermal mass.

5.1.5 Daytime versus all hour window control

The results also revealed that a daytime window control system would be equally effective in Living Lab as a window control system designed to operate both day and night. This is due to thermal storage ventilative cooling not being effective. Also because there is no solar radiation and the outdoor temperatures are low outside of the chosen daytime period. Hence, applying ventilative cooling only during the day is sufficient in Living Lab as long as it is coinciding with the expected outdoor temperature and solar radiation patterns.

Previous research comparing use of daytime to all hour ventilative cooling could not be obtained. However, nighttime ventilative cooling has been proven to be more effective in other buildings. It is therefore likely that a daytime ventilative cooling system would not always result in equal thermal comfort as an all hour system.

5.1.6 Stack ventilation through on/off control versus stack and cross-flow ventilation through PI control

The on/off window control system was considered a better choice than the PI window control system. This was not anticipated. A more advanced window control system was expected to result in better thermal conditions. The set-point testings had indicated that both window control systems could obtain almost equal thermal comfort. However, the results from the whole year simulations revealed that the chosen on/off control system would provide better thermal comfort than the chosen PI control system. This could mean that the process of designing the window control system might not have been adequate for the PI system. The PI system utilized cross-flow ventilation in addition to stack ventilation. The on/off system utilized only stack ventilation. Cross-flow ventilation is more difficult to control and predict. This could explain why the PI system provided weaker and more inconsistent results. The PI control system also had more components and was more complex than the on/off control system. This could be another explanation to why it was more difficult to find a fitting design for PI control. Also, the PI control system might have resulted in better thermal conditions if a time-delay had been utilized. Nevertheless, the on/off window control system is considered a better choice for window regulation in Living Lab. It is less complex and would therefore result in less installation and maintenance costs. It is also is expected to have better compliance between simulated operation and real life operation than the PI system. The on/off design has only one opening and one closing temperature. Also, the windows are either opened to the maximum opening degree or held completely closed. The PI window control system on the other hand, is composed of several regulators. For this system to work as implied by the simulations, these PI regulators have to work in the exact same way as they are designed in IDA ICE. This system also caused continuos changes of the window openings. It is not likely that a real window control system would change the window openings as frequently.

The results are indicating that the more complex natural ventilation system, the more accurate and careful design process is required. One of the studies reviewed in the literature survey revealed that measurements in a low-energy dwelling in Denmark displayed far higher indoor temperatures than predicted. It concluded that the main reason was that it was difficult to create cross-flow ventilation in the house. (Thomsen et al., 2005). Hence, supporting the statement that a window control system utilizing cross-flow ventilation requires a more accurate and careful design process.

5.1.7 Mechanical window control versus manual window control

The on/off window control system was designed to represent a simple automatic system or how the occupants could manually operate windows. This system was activated by a timer. Hence, manually window operation was not properly tested. The PI window control system on the other hand, was initially activated by presence. When this system was set to only operate when the occupants were present and awake, undesired thermal conditions were created. This was due to high solar radiation in the mornings before the occupants got out of bed. Once the high indoor air temperatures were reached, it took several hours to reduce them to an acceptable level. Hence, indicating that manually operated ventilative cooling would not be sufficient in Living Lab. Even though the chosen window control system has a simple design, it is unlikely that the occupants will operate the windows exactly this way. A mechanical window control system is therefore necessary for ventilative cooling to provide sufficient thermal comfort in Living Lab.

The results are indicating that ventilative cooling should not be dependent on the occupants to operate the system. A previous study came to the same conclusion. It revealed that ventilative cooling intended to be operated by the occupants resulted in poor thermal comfort. This study concluded that automatic window control would have led to better performance. (Thomsen et al., 2005). However, another study revealed that manually operated ventilative cooling provided sufficient thermal comfort. (Mlakar and Strancar, 2011). From the literature survey, it was also found that thermal comfort in naturally ventilated buildings is strongly influenced by the higher levels of perceived control. (de Dear and Brager, 2002). It seems that automatic window control is more likely to result in the desired thermal comfort. However, to achieve the advantages of personal control, the occupants should be able to overrule the automatic system.

5.2 The effect of ventilative cooling on the thermal environment

5.2.1 Range of acceptable temperatures

The results revealed that the lower range of the initial desired indoor temperatures could not always be accepted when applying ventilative cooling in Living Lab. Increased air velocities resulted in expected perceived air temperatures lower than the actual air temperatures. The method for predicting perceived temperatures was based estimates of air velocities and the temperature corrections provided by standard NS-EN 15251 (2007). However, different individual perceived temperatures are likely to occur. Nevertheless, it is clear that ventilative cooling caused high indoor air velocities, which are likely to shift the desired temperature range.

The results are indicating that the acceptable indoor air temperatures for mechanically ventilated buildings might not be applicable when ventilative cooling is utilized. A previous study concluded that occupants of naturally ventilated buildings prefer a wider range of conditions. (de Dear and Brager, 2002). Another study revealed that higher air velocity under personal control make the indoor environment acceptable at higher air temperatures. (Cattarin et al., 2012). This assignment mostly found the need to increase the lower range of acceptable temperatures. Previous studies have found the need to increase the upper range of the acceptable temperatures. Nevertheless, both results support the statement that the initial range of acceptable temperatures might have to be adjusted for buildings intended to utilize ventilative cooling.

5.2.2 The issue of overheating

The whole year simulations of the reference cases revealed that extremely high indoor air temperatures are likely to occur if passive or active cooling measures are not taken. Living Lab is particularly exposed to solar radiation due to large windows on all facades. The building is also designed with low infiltration and U-values. Once heat is obtained, it will therefore not easily escape through the building body. However, several assumptions were made when performing simulations on Living Lab. It was assumed that four people lived in the building. It was further assumed that they were present all the time on weekends and all the time except from 08.00-17.00 on weekdays. These assumptions were made to simplify the occupancy schedules and to include the worst case scenario in terms of internal heat gains. However, it is likely that the building will be occupied less than this. Also, when performing the reference case simulations, all windows were held closed. This assumption was made to distinct the effect of ventilative cooling. However, it is not likely that the occupants would never open any windows. Due to the use of worst case scenarios the most extreme high indoor air temperatures might not be realistic. Nevertheless, it is clear that uncomfortably high indoor air temperatures are likely to occur frequently in Living Lab if not any form of cooling is utilized.

The results from this study are indicating that the end-user of low-energy dwellings are likely to be dissatisfied if not any cooling measures are taken. The literature survey both supports and rejects this statement. A previous study recorded good thermal comfort in low-energy residents even without active or passive cooling solutions. (Feist et al., 2005). A few studies recorded general satisfaction but occurrences of overheating during summer. (Mlecnik et al., 2012), (Kleiven, 2007). Another study recorded general dissatisfaction with the thermal environment due to overheating in summer. (Samuelsson and Luddeckens, 2009). The differences in recorded results could be explained with climate variations, different building design and also individual expectations and preferences. However, it is clear that overheating in low-energy dwellings can be an issue. It must therefore be addressed during the design process.

5.2.3 Preventing overheating with ventilative cooling

The current study revealed that overheating can be prevented in Living Lab if the right ventilative cooling system is applied. The outdoor air temperatures in Trondheim are often below the lower range of acceptable indoor temperatures. Large amounts of free cooling are therefore available. Living Lab is equipped with large openable windows which enables rapid air exchanges of the building. The results revealed that comfortable indoor air temperatures could be achieved even when only using natural ventilation through the south and skylight windows. It is therefore evident that the window opening areas in Living Lab is not a limiting factor for ventilative cooling.

This study is indicating that overheating in low-energy dwellings can be prevented by applying ventilative cooling. This statement is supported by the literature survey. Other low-energy dwellings have also prevented overheating due to ventilative cooling alone (Oropeza-Perez and Østergaard, 2014) or in combination with extensive solar shading. (Thomsen et al., 2005), (Fjoldberg et al., 2011).

5.2.4 The issue of overcooling

The results revealed that overcooling caused by ventilative cooling could not be completely eliminated. Prevention of overcooling was found to be equally challenging as prevention of overheating. This is due to the uncertainties related to the use of natural ventilation. Outdoor temperatures, wind velocity and wind directions are unpredictable. So are the supply air temperature and the air velocities caused by window openings. However, the results showed that overcooling could be held at an acceptable level in Living Lab.

The current study is indicating that overcooling is an issue that should be considered for low-energy dwellings utilizing ventilative cooling. Only one of the studies reviewed in the literature survey was found to address this issue. This study revealed that use of daytime ventilative cooling in combination with night cooling provided good thermal comfort in a building in Athens. However, the same combination caused overcooling in Rome, Berlin and Copenhagen. (Pellegrini et al., 2012). Hence, this study is supporting the statement that overcooling can be an issue. It should therefore be addressed when designing dwellings intended to utilize ventilative cooling.

5.2.5 Factors influencing the need for ventilative cooling

This study revealed that the factors most influencing the need for ventilative cooling in Living Lab were in the following order: Solar radiation, outdoor temperatures and occupancy. The dwelling is designed with large window areas on all facades but it is only equipped with venetian blinds on the south window. It is also located in a rather open landscape and is therefore particularly exposed to solar radiation. High outdoor temperatures increase the need for ventilative cooling due to infiltration and exterior surface heating. Low outdoor temperatures decrease the need for ventilative cooling due to infiltration, exterior surface cooling and because indoor temperatures can be reduced more quickly when utilizing natural ventilation. Occupancy is the least influencing factor because there are only four people expected to reside in the building. Also, the internal heat gains due to occupancy are moderate. Presence of wind did not seem to significantly affect the need for ventilative cooling. The building is designed with low infiltration values. It is therefore protected against the influence of wind. High wind velocities did, however, cause increased air velocities in the building when windows were open. Hence, it is influential to the effect of ventilative cooling. Previous research studying factors influencing ventilative cooling could not be obtained.

5.3 The effect of ventilative cooling on energy use

5.3.1 Natural ventilative cooling

The natural ventilation system was found to have substantial lower energy demand than the reference cases. It also had the lowest energy demand out of the three ventilative cooling solutions. This system did not utilize mechanical ventilation. It had therefore zero energy demand for fans and AHU heating. The natural ventilation system also obtained less energy demand for zone heating than the reference cases and the mixed-mode systems. This was unexpected considering natural ventilation does not obtain any heat recovery. However, the mechanical ventilation was set to supply air at a low temperature. For the reference cases and the mixed-mode systems, zone heating had to compensate for this during non-cooling periods. The natural ventilation system did, however, not have to compensate for the frequently mechanical supplied air with low temperature. If a higher supply air temperature had been used during the heating period, energy demand for zone heating would have been reduced for the mixed-mode solutions and the reference cases.

This study is indicating that utilizing natural ventilation instead of mechanical ventilation can result in reduced energy use. One of the studies reviewed in the literature survey stated that energy savings were recorded when utilizing natural ventilation instead of mechanical ventilation due to reduction in electric fan use. (Oropeza-Perez and Østergaard, 2014). Another study recorded decreased electric energy use, but increased space heating when utilizing natural ventilation instead of mechanical ventilation. The result was almost zero change in primary energy consumption. (Simonson, 2005). Differences in results regarding increase or decrease in space heating when utilizing natural ventilation instead of mechanical ventilation can be explained with differences in supply air temperature and supply airflows. However, it is evident that utilizing natural ventilation instead of mechanical ventilation will eliminate energy use for fans and AHU heating.

5.3.2 Concurrent ventilative cooling and change-over ventilative cooling

The mixed-mode systems resulted in either a small increase or a slight decrease in energy demand compared to use of only mechanical ventilation. The change-over system did not reduce energy demand as much as expected. This was because it was designed as a zoned system. Better results could have been obtained if it had been designed as a centrally change-over system with a delay. For example, turning off all the mechanical ventilation units for one hour if natural ventilation was utilized. This would have resulted in less use of energy for fans and AHU heating. Also, it would not have unbalanced the mechanical ventilation system.

This study is indicating that mixed-mode ventilative cooling could in best case reduce energy demand and in worst case only slightly increase the demand compared to use of only mechanical ventilation. One study reviewed in the literature survey revealed that utilizing increased air velocities during the day and/or nighttime cooling lead to a consistent reduction in energy use. (Pellegrini et al., 2012). Another study showed that natural ventilation in combination with mechanical ventilation had a significant positive effect on the use of energy. (Fjoldberg et al., 2011). Differences in results can be explained with different reference points. If ventilative cooling is compared to use of mechanical cooling, it is more likely to result in decreased energy consumption than if compared to use of only mechanical ventilation. However, utilizing ventilative cooling could result in reduced energy use for the building. In worst case it will not have a significant effect on the energy consumption.

Previous research comparing the effect on energy use for natural ventilative cooling, concurrent and change-over ventilative cooling could not be obtained.

6 Conclusion

The results from the simulations implied that there will be a severe risk of overheating in Living Lab if no active or passive cooling techniques are applied. The results showed nonetheless that ventilative cooling can prevent overheating without significantly increasing the energy demand. Due to the uncertainties related to increased air velocities, it was not possible to eliminate the risk of overcooling caused by ventilative cooling completely. However, the study showed that the amount of hours with overcooling could be held at an acceptable level. The simulations revealed that nighttime ventilative cooling had no positive effect on the thermal environment in Living Lab. This study also showed that the factors most influencing the need for ventilative cooling were in the following order: solar radiation, outdoor temperature and occupancy. Presence of wind did not significantly affect the need for ventilative cooling. However, it did influence the cooling effect. The openable window areas were found to be sufficient.

The study found that the best way to apply ventilative cooling in Living Lab would be to implement a concurrent mixed-mode system where the window control system is only active during the day. It should be designed to open the south and skylight windows to maximum opening when indoor air temperatures exceed 24^oC and close them when indoor air temperatures drops below 22^oC. The results revealed that this system would reduce the number of overheated hours recorded when not utilizing ventilative cooling with 99%. The number of overcooled hours would be kept at a moderate level, 48 hours/year. Utilizing this ventilative cooling system resulted in increased energy demand of 52 kWh/year and 4 kWh/year compared to use of only hygienic mechanical ventilation and only enhanced mechanical ventilation, respectively.

This assignment and previous research show that overheating in low-energy dwellings is often an issue. It should therefore be addressed during the design process. Overheating in low-energy dwellings can be prevented with ventilative cooling. Ventilative cooling can have a significant positive effect on the thermal environment without having a significant negative effect on the use of energy. In some cases, energy consumption can even be reduced when applying ventilative cooling. Overcooling can be an issue when utilizing ventilative cooling. A careful design process is required for ventilative cooling to have the desired effect. The process should be individual for each building and climate. A more complex natural ventilation system requires a more accurate and careful design process. Also, the acceptable indoor temperatures for mechanically ventilated buildings often have to be adjusted for buildings intended to utilize ventilative cooling. Even though nighttime ventilative cooling is not applicable in certain lightweight dwellings, it has proven to be effective in other buildings with more thermal mass. An automatic window control system is often necessary for ventilative cooling to achieve the desired thermal environment. However, to secure the advantages with personal control, the occupants should be able to overrule the automatic system.

7 Suggestions for further work

This study of ventilative cooling in Living Lab could be taken further. A more in depth study of the PI window control system could be conducted. Determining whether it is poor design or that it utilizes cross-flow ventilation that is causing the inconsistent results would be useful to the development of a better control system. Also, a central change-over mixed-mode system could be created and analyzed to see if the energy demand would be reduced compared to a change-over zoned system. The study could also be expanded to include additional occupancy schedules and include evaluation of air humidity, indoor air quality, air velocities from the mechanical ventilation system, air velocities from other windows than the south window, draught situations and local temperature variations. More extensive CFD-simulations should also be conducted.

When the building is constructed, measurements should be performed to determine how the system works in real life. Also, an end-user evaluation should be conducted after the building has been occupied for some time. If ventilative cooling from the mixed-mode system turns out to not work as well as this assignment indicates, the possibilities of providing external shading on the remaining windows should be looked into. The west bedroom seems particularly exposed to solar radiation. In addition, the possibilities of installing a PCM air heat exchanger and utilizing the ground source heat pump for cooling could be evaluated.

Similar studies on how to apply and evaluate the effect of ventilative cooling should be conducted for other low-energy buildings to see if the findings are consistent. It seems that the current literature lacks results regarding factors most influential to the need for ventilative cooling and comparison between natural, concurrent and change-over ventilation. Also, more studies examining the applicability of cross-flow ventilation and studies devoted to the challenges of overcooling when applying ventilative cooling in colder climates should be conducted.

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A Appendix

A.1 Thermal comfort equations

A.1.1 PMV

$$\begin{aligned}PMV = (0.303e^{-0.036M} + 0.028)\{(M-W) - 3.05\times10^{-3}[5733 - 6.99(M-W) - p_a]\\ -0.42[(M-W) - 58.15] - 1.7\times10^{-5}M(5867 - p_a) - 0.0014M(34 - t_r)\\ -3.96\times10^{-8}f_{cl}[(t_{cl}+273)^4 - (t_{mrt}+273)^4] - f_{cl}\alpha_c(t_{cl}-t_r)\}\end{aligned}$$

Where:

$$\begin{aligned}t_{cl} = 35.7 - 0.028(M-W) - I_{cl}f_{cl}\alpha_c(t_{cl}-t_r)\\ -I_{cl}\{3.96\times10^{-8}f_{cl}[(t_{cl}+273)^4 - (t_{mrt}+273)^4)]\}\end{aligned}$$

$$\begin{aligned}\alpha_c &= 2.38(t_{cl}-t_r)^{0.25} \quad \text{for} \quad 2.38(t_{cl}-t_r)^{0.25} > 12.1\sqrt{v_{ar}}\\ \alpha_c &= 12.1\sqrt{v_{ar}} \quad \text{for} \quad 2.38(t_{cl}-t_r)^{0.25} < 12.1\sqrt{v_{ar}}\end{aligned}$$

$$\begin{aligned}f_{cl} &= 1.00 + 1.290Icl \quad \text{for} \quad Icl < 0.078m^{2o}C/W\\ f_{cl} &= 1.05 + 0.645Icl \quad \text{for} \quad Icl < 0.078m^{2o}C/W\end{aligned} \tag{12}$$

$M =$ Metabolic rate, W/m^2 body surface
$W =$ External work, W/m^2 body surface
$I_{cl} =$ Thermal resistance of clothing, m^2K/W
$f_{cl} =$ Ratio of a persons surface area while clothed,
to a persons surface area while nude
$t_r =$ Room air temperature, ^oC
$v_{ar} =$ Air velocity, m/s
$p_a =$ Partial water vapors pressure, Pa
$\alpha_c =$ Convective heat transfer coefficient, W/m^2K
$t_{cl} =$ Surface temperature of clothing, ^oC

A.1.2 PPD

$$PPD = 100 - 95e^{(-0.03353.PMV^4 - 0.2179.PMV^2)} \tag{13}$$

A.1.3 Draught

$$DR = (34 - t_a)(v_a - 0.05)^{0.62}(3.14 + 0.37\sigma_{v_a})$$

$$v_a = \text{Mean air velocity (3min)}, m/s \qquad (14)$$
$$\sigma_{v_a} = \text{Standard deviation of air velocity (3min)}, m/s$$
$$t_a = \text{Air temperature},^o C$$

A.2 Short description of the analysis used for verification

The building used in the analysis was a square building with a floor area of $100m^2$ and a height of 4m. It had unopenable windows on all facades (in order to account for solar radiation) in addition to two openable windows; one in the center of the south facade ($9{,}87m^2$), and one at the roof center ($4{,}8m^2$). The building was assigned U-values and g-values corresponding to Living Lab values and internal heat gain variations according to standard NS-EN 13792. An illustration of the building is presented in Figure 55.

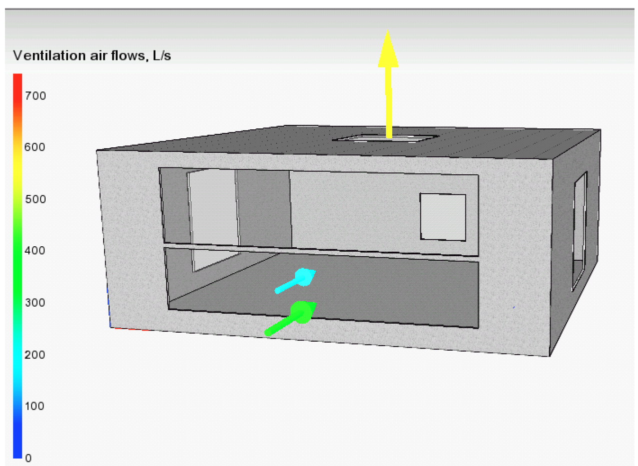


Figure 55: Illustration of the model used for verification of the IDA ICE model (Kirkøen, 2014)

The simplified building was studied for four different scenarios combining two different system solutions (with or without mechanical cooling) with two sets of outdoor conditions, one corresponding to a regular warm summer day in Trondheim (July 17th 2014, temperatures reaching 20.9^{o}C) and the other corresponding to the warmest summer day in Trondheim during the year 2014 (July 9th 2014, temperatures reaching 29.9^{o}C). The scenarios applicable to this verification are the two cases studying natural ventilation without mechanical cooling. All scenarios aimed at keeping the indoor temperature at set-point, 21^{o}C.

The calculation method was an eight-step procedure combining several well-established mathematical equations. First, the outdoor and indoor air densities were calculated using a formula by Orlando et al. (2004). Second, the total heat surplus was calculated using methods from NS-EN-13792 (2012) and formulas found in Stensaas (2001) and Stensaas (1980). Further, it was determined whether cooling was needed and cooling by natural ventilation was feasible. Using formulas found in Stensaas (2001) and Liddament (1986), the total driving pressure was calculated. By combining formulas from Novakovic (2007) and Stensaas (2001), the necessary window opening area and resulting airflow through the window were determined. Last, combining all the equations in a spreadsheet in Excel, the resulting indoor air temperature was determined. The calculations were conducted with time-steps of one hour. (Kirkøen, 2014).

The same simplified model was built in IDA ICE, and the same weather profiles and window openings were implemented. Simulations were conducted, resulting in hourly values for indoor air temperature.

A.3 Living Lab

A.3.1 Floor plan

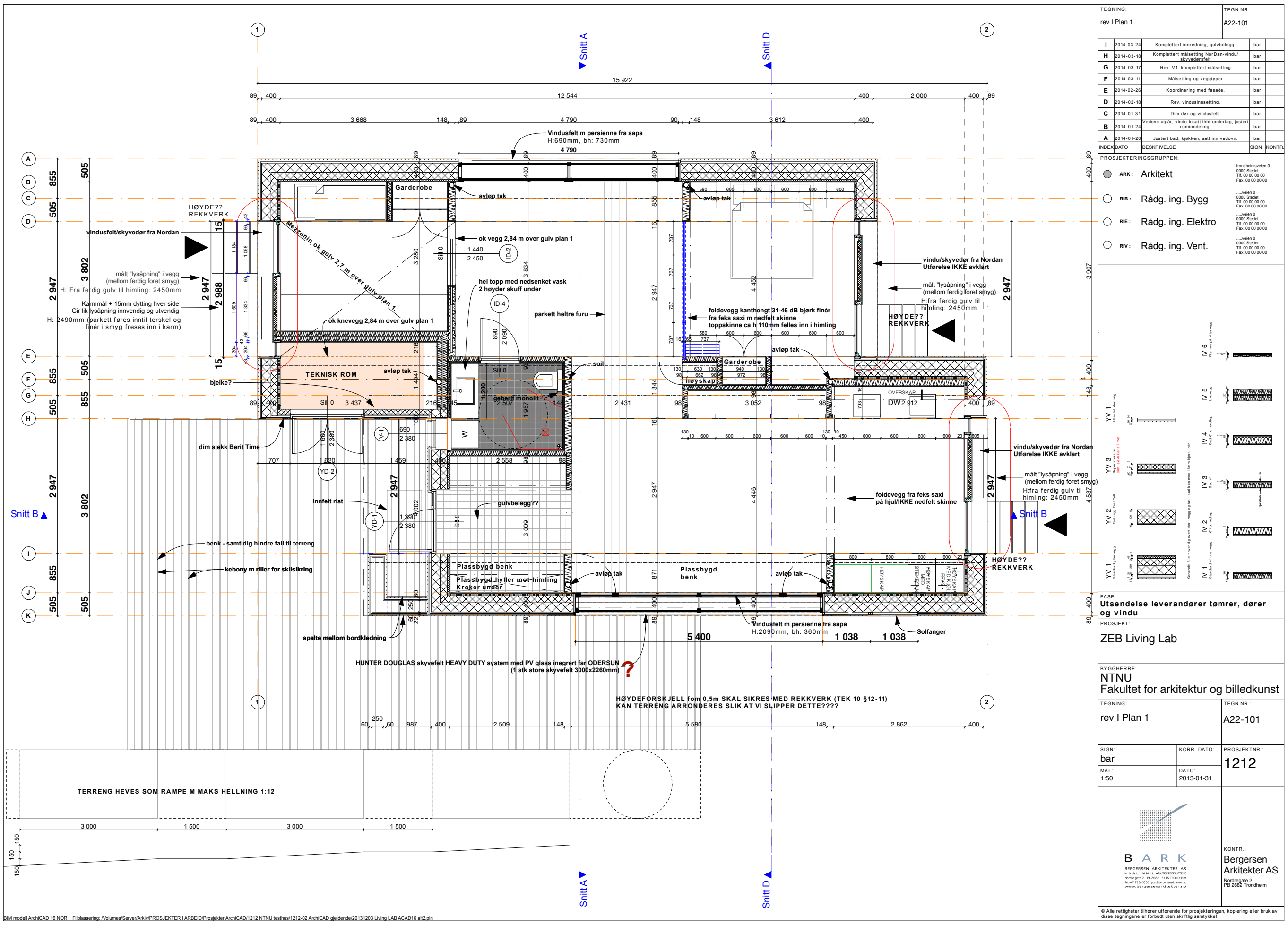


Figure 56: Floor plan of Living Lab, Floor 1

A.3.2 Sectional drawing

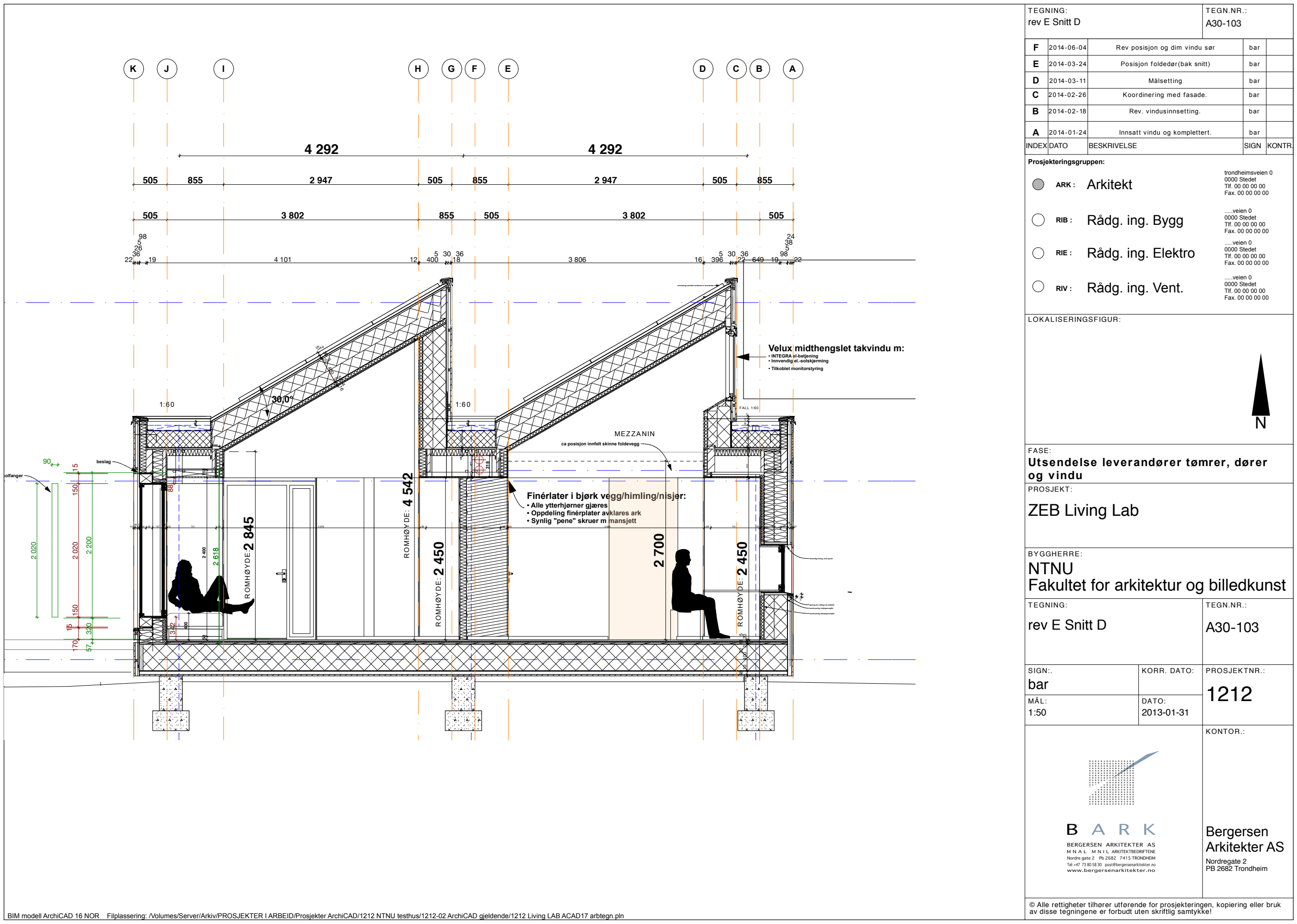


Figure 57: Sectional drawing of Living Lab, Section D

A.3.3 Supply and extract airflows

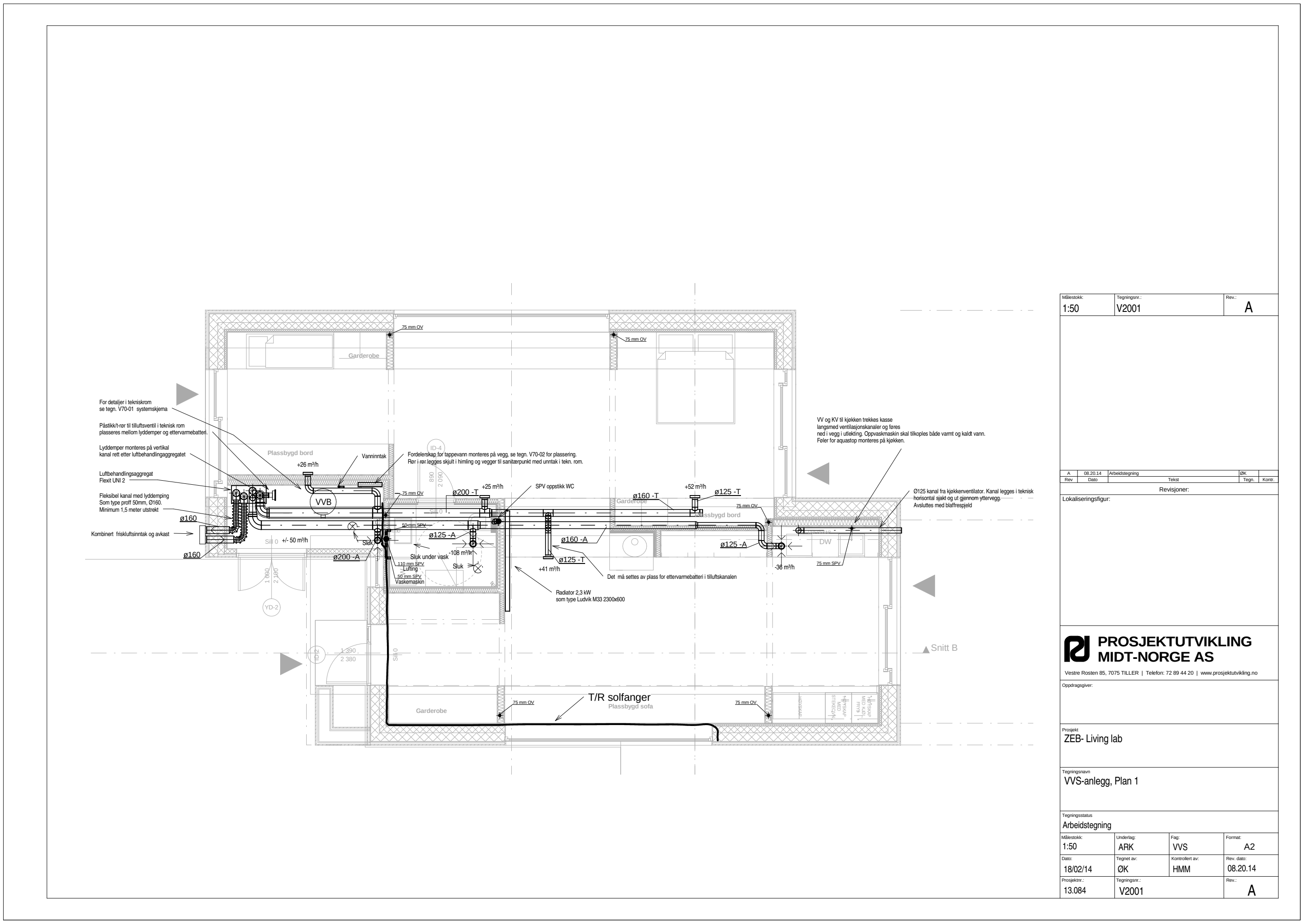
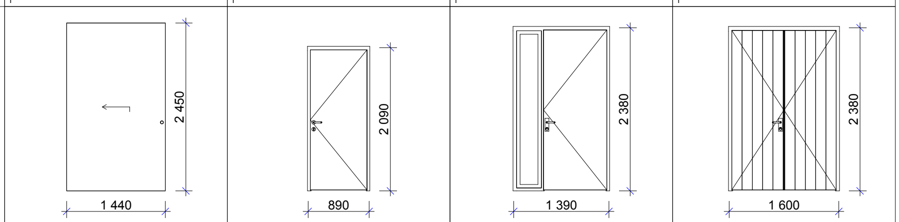


Figure 58: Ventilation drawing showing supply and extract units

A.3.4 Doors

ID		ID-2	ID-4	YD-1	YD-2
Slagretning		V	H	H	V
Rev.					
Antall		1	1	1	1



Størrelse	B=	1 440	890	1 390	1 600
	H=	2 450	2 090	2 380	2 380
Brannklasse		-	-	-	Ikke avklart ?????
Lydkrav		-	-	40dB	40dB
U-verdi		-	-	0,6 - 0,7	0,6 - 0,7
Sparkeplate		---	Uten	Uten	Uten
Terskel		Av	Våtromsterskel	Anslagterskel	Anslagterskel
Overflate		Bjørk finér - lakket	Bjørk finér - lakket	Kvistfri furu. Fastkarm: NCS S 5502-Y (RAL 9007) Vinduskarm og dørblad NCS S 8500-N (RAL 7021)	Kvistfri furu. Fastkarm: NCS S 5502-Y (RAL 9007) Vinduskarm og dørblad NCS S 8500-N (RAL 7021)
Glass		-	-	Energiglass. Sikkerhetsglass.	-
Hengsler		Ihht leverandør - skinne i topp	Ihht leverandør	Ihht leverandør	Ihht leverandør
Låssystem		Ref Låsgruppen AS v/ Sommerbakk	Ref Låsgruppen AS v/ Sommerbakk	Ref Låsgruppen AS v/ Sommerbakk	Ref Låsgruppen AS v/ Sommerbakk
Nøkkelside		Begge sider	Begge sider	Begge sider	Begge sider
Beslag		Alle beslag, skilt, vridere mm i børstet, syrefri RF stål. Forøvrig ref Låsgruppen AS v/ Sommerbakk	Alle beslag, skilt, vridere mm i børstet, syrefri RF stål. Vridere type: TrioVing VR 86101. Forøvrig ref Låsgruppen AS v/ Sommerbakk	Alle beslag, skilt, vridere mm i børstet, syrefri RF stål. Vridere type: TrioVing VR 86101 m/ langskilt. Forøvrig ref Låsgruppen AS v/ Sommerbakk. (Tilkobling klokke, alarm ?)	Alle beslag, skilt, vridere mm i børstet, syrefri RF stål. Vridere type: TrioVing VR 86101 m/ langskilt. Forøvrig ref Låsgruppen AS v/ Sommerbakk. (Tilkobling klokke, alarm ?)

BIM modell ArchiCAD 16 NOR Filplassering: /Volumes/Server/Arkiv/PROSJEKTER I ARBEID/Prosjekter ArchiCAD/1212 NTNU testhus/1212-02 ArchiCAD gjeldende/20131203 Living LAB ACAD16 alt2.pln

TEGNING: Dørskjema
TEGN.NR.: A61-103

INDEX	DATO	BESKRIVELSE	SIGN	KONTR.
A	2014-03-17	Rev. fargekoder	bar	

Prosjekteringsgruppen:

ARK : Arkitekt — trondheimsveien 0, 0000 Stedet, Tlf. 00 00 00 00, Fax. 00 00 00 00
RIB : Rådg. ing. Bygg — .....veien 0, 0000 Stedet, Tlf. 00 00 00 00, Fax. 00 00 00 00
RIE : Rådg. ing. Elektro — .....veien 0, 0000 Stedet, Tlf. 00 00 00 00, Fax. 00 00 00 00
RIV : Rådg. ing. Vent. — .....veien 0, 0000 Stedet, Tlf. 00 00 00 00, Fax. 00 00 00 00

LOKALISERINGSFIGUR:
N

FASE: Arbeidstegninger

PROSJEKT: ZEB Living Lab

BYGGHERRE: NTNU Fakultet for arkitektur og billedkunst

TEGNING: Dørskjema
TEGN.NR.: A61-103

SIGN:. bar
KORR. DATO:
PROSJEKTNR.: 1212
MÅL: 1:50
DATO: 2014-02-07

BARK
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PB 2682 Trondheim

Figure 59: Schematic of doors for Living Lab

A.3.5 Windows

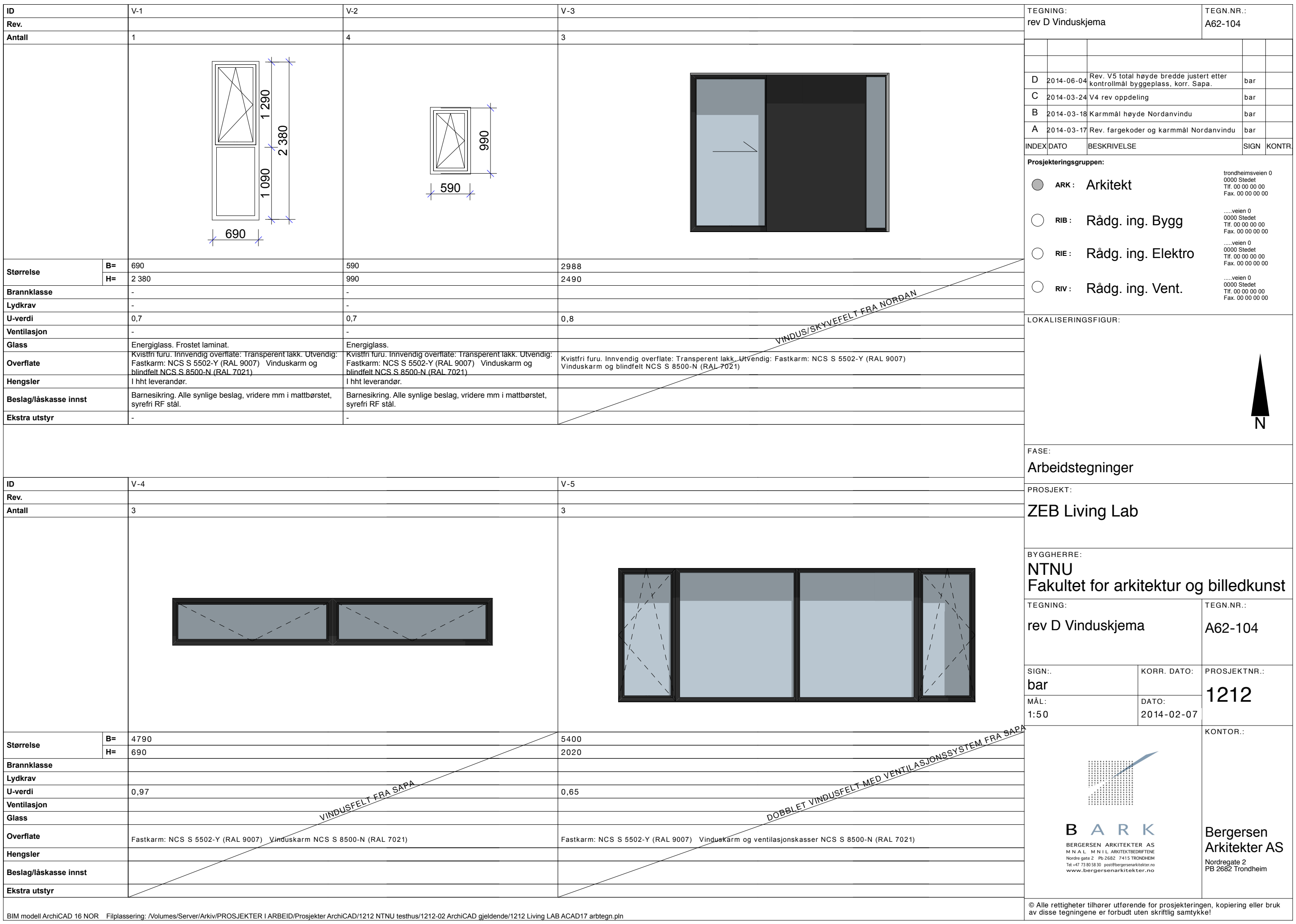

Figure 60: Schematic of windows for Living Lab

A.3.6 Vertical section south window

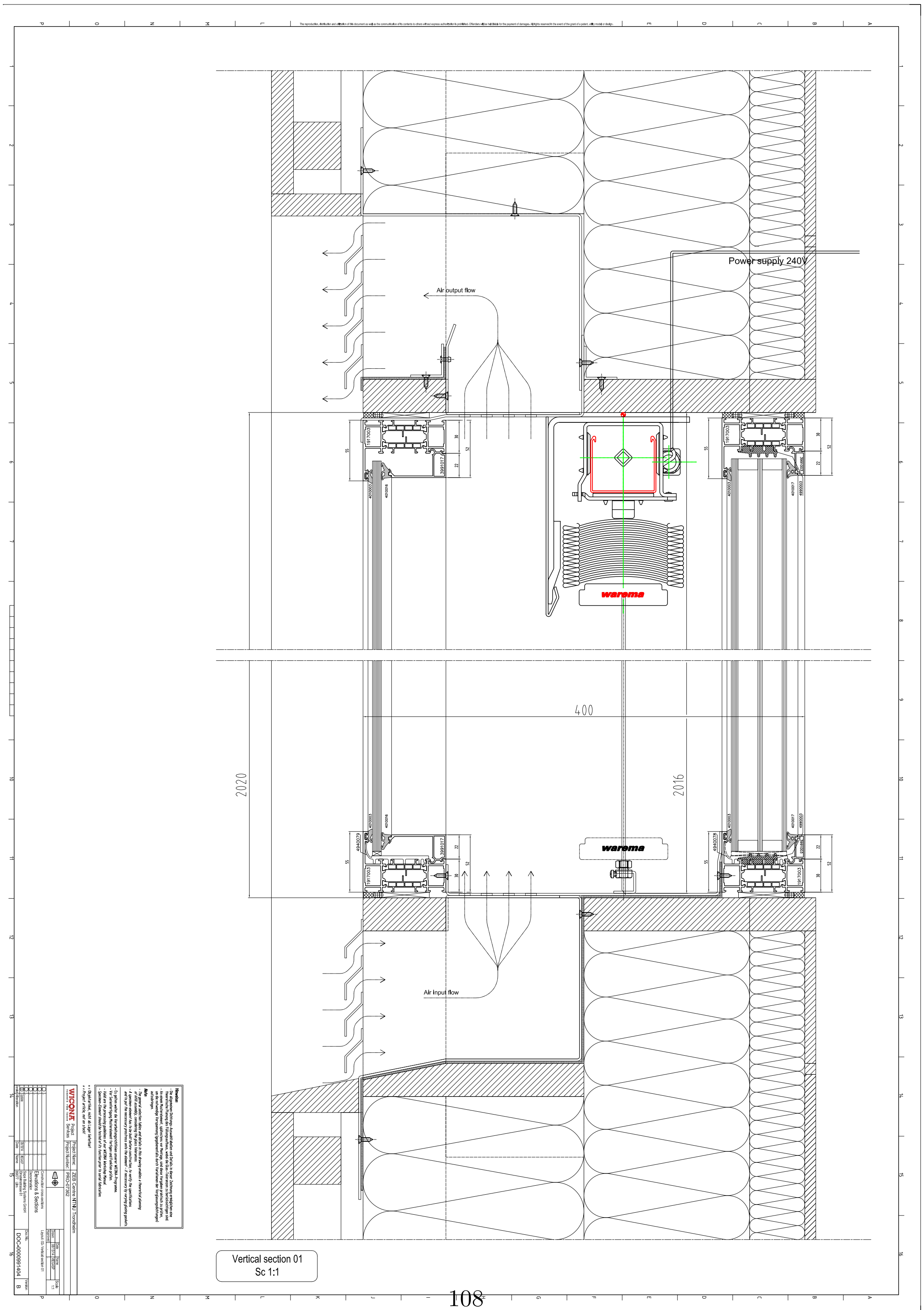

Figure 61: Vertical section of the south ventilated window

A.3.7 System form

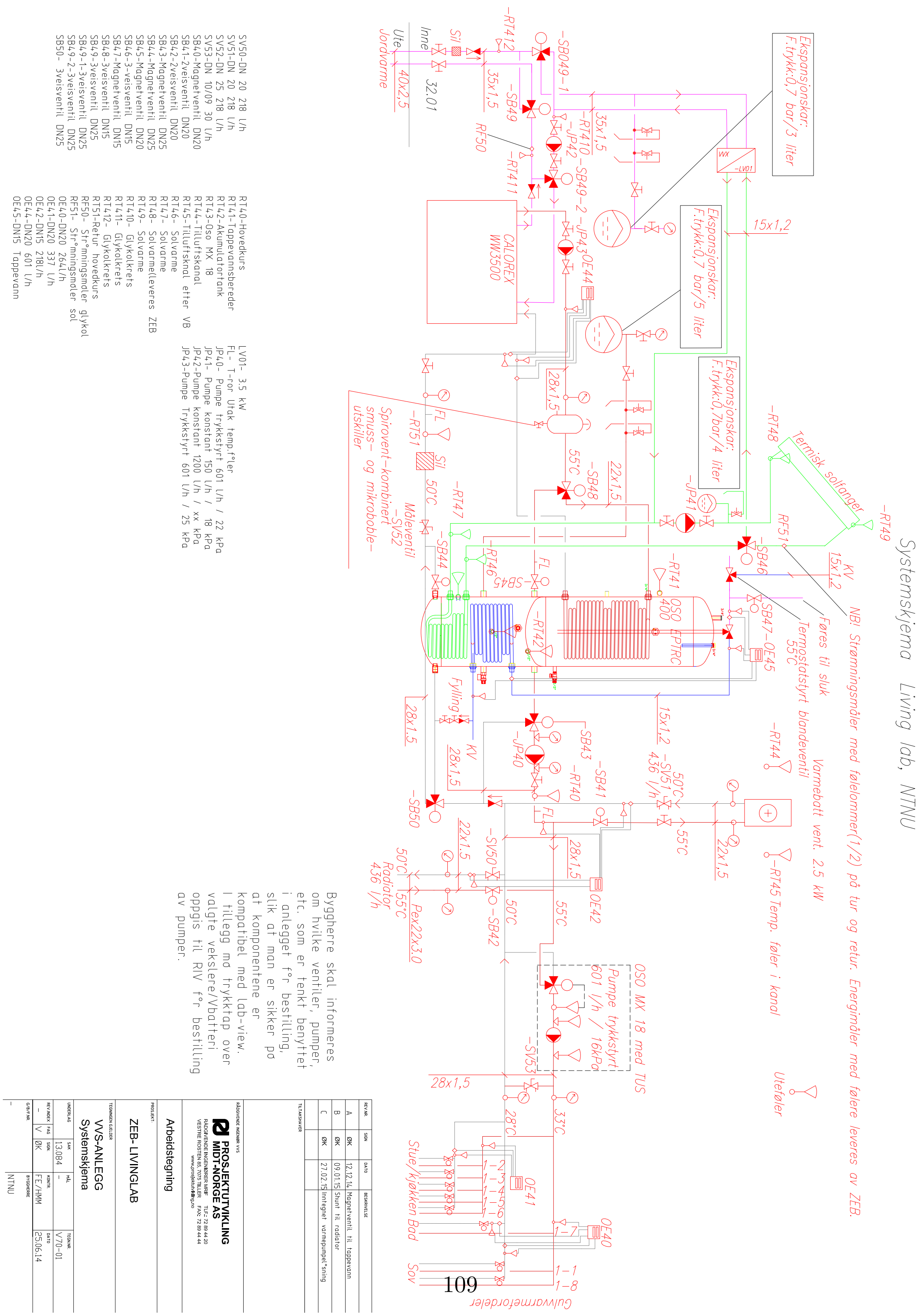

Figure 62: System form

A.3.8 Floor heating

TEKNISK SPESIFIKASJON Side 1 14.04.2014

Uponor Gulvvarme - Tilbuds.Nr. 153409-6907 - NTNU TR.Heim Zeb Living Lab

Fordeler 1

Navn	Rør dim mm	Installasjon løsning	Gulv løsning	CC	Romtemp. °C	Effektbehov W/m²	Sløyfelengde m	Vannmengde l/s	Trykfall kPa	Ventil omdreininge
Loop 1-1	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	68	0,02	2,24	2
Loop 1-2	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	54	0,016	1,18	1,5
Loop 1-3	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	79	0,023	3,4	2,5
Loop 1-4	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	96	0,028	5,84	3,6
Loop 1-5	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	92	0,027	5,19	3,2
Loop 1-6	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	63	0,018	1,81	1,8
Loop 1-7	17-system	Gulvvarmesponplate 1800x600	Betong	200	20,0	30	28	0,008	0,19	0,6
Loop 1-8	17-system	Gulvvarmesponplate 1800x600	Parkett 14mm	200	20,0	30	98	0,028	6,19	5

Dim. Trykkfall:	6,99 kPa
Dim, vannmengde:	0,167 l/s
Dim. turvannstemperatur:	35 °C
Dim. gulvoverflatetemperatur :	23 °C

Totalt varmebehov:	3,468 kW
Dim. Trykkfall:	7,35 kPa
Dim, vannmengde:	0,167 l/s
Dim. turvannstemperatur:	35 °C
Dim. gulvoverflatetemperatur :	23 °C
Vann volum i rør:	75,6 l
Delta T:	5 °C

UPONOR PEPEX Q&E RØR 17x2,0 MM HVIT, KVEIL 240M (240 m)

Length	Manifold	Loop
79,0	Fordeler 1	Loop 1-3
98,0	Fordeler 1	Loop 1-8
63,0	Fordeler 1	Loop 1-6

UPONOR PEPEX Q&E RØR 17x2,0 MM HVIT, KVEIL 240M (240 m)

Length	Manifold	Loop
96,0	Fordeler 1	Loop 1-4
68,0	Fordeler 1	Loop 1-1
54,0	Fordeler 1	Loop 1-2

Uponor A/S P.O.Box 23 T +47 64 95 66 00 Org nr 960 253 108
Uponor VVS 1541 Vestby F +47 64 95 31 20
Norge W www.uponor.no

Figure 63: Information about the floor heating system

A.4 Modelling Living Lab in IDA ICE

A.4.1 Characteristics of materials used

Table 41: Living Lab model in IDA ICE: Characteristics of materials used

Material characteristics			
Material	Heat conductivity [W/mK]	Density [kg/m3]	Spesific heat [J/kgK]
Plywood	0.15	670	2.5
Insulation	0.037	100	1.03
Massive pinewood	0.147	420	2.5
Wood HDF	0.015	900	1.7
Vapor barrier	0.8	910	1.8
Rockwool	0.06	20	1.03
Paper	0.2	192	0.33
Wind barrier	0.5	910	1.8
Airgap	0.0357	1.2	1.005
Cladding	0.7	500	2.5
Water proof barrier	0.22	910	1.8
Tiles	1.3	2300	9.84
Concrete	1.75	2000	1

A.4.2 Floor area zones

Table 42: Floor area zones

Zone areas	
Zone	Floor area [m2]
Bedroom West	14.98
Bedroom East	20.50
Living Room	50.63
Kitchen	19.32
Bathroom	5.52
Entrance	11.03
Loft	14.98
Technical	7.39

A.4.3 Light and equipment schedules

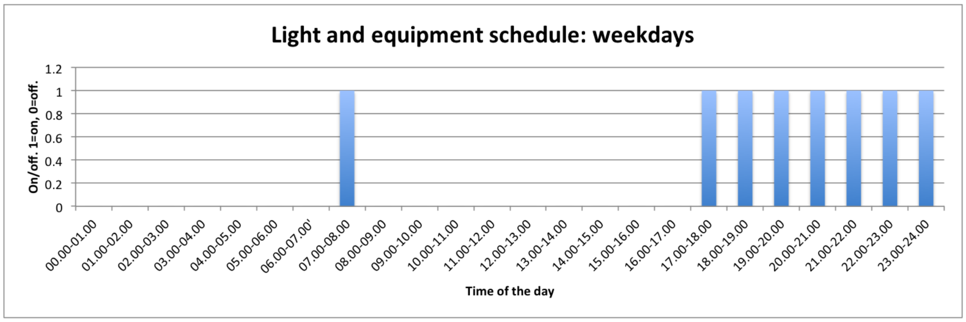

Figure 64: Living Lab model in IDA ICE: Light and equipment schedule on weekdays

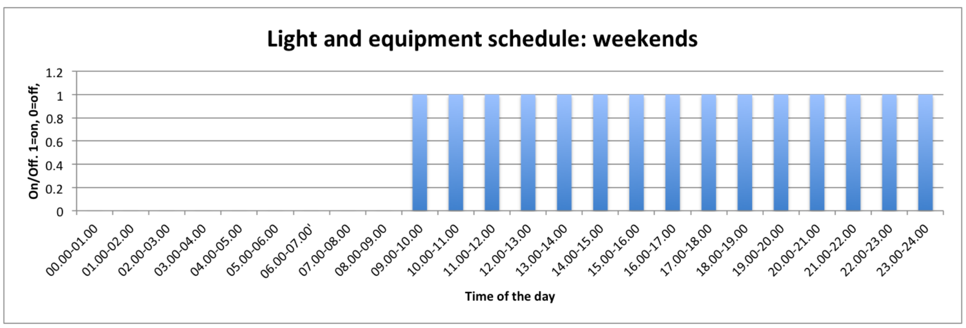

Figure 65: Living Lab model in IDA ICE: Light and equipment schedule on weekends

A.4.4 U-values doors

Table 43: Living Lab model in IDA ICE: U-values internal and external doors

U-values doors	
Door	U-value [W/m^{2}K]
Entrance door	0.65
External door technical room	0.65
Internal doors	1.05

A.5 Basic formulas

A.5.1 Saturation pressure

$$ln(p_{sat}) = \sum_{i=1}^{4} g_i \times T^{i-2} + g_7 \times lnT \tag{15}$$

$$g_1 = -6096,9385$$
$$g_2 = 21,2409642$$
$$g_3 = -0,02711193$$
$$g_4 = 1.67395 \times 10^{-5}$$
$$g_7 = 2.433502$$

A.6 Calculating maximum window opening %

A.6.1 South window

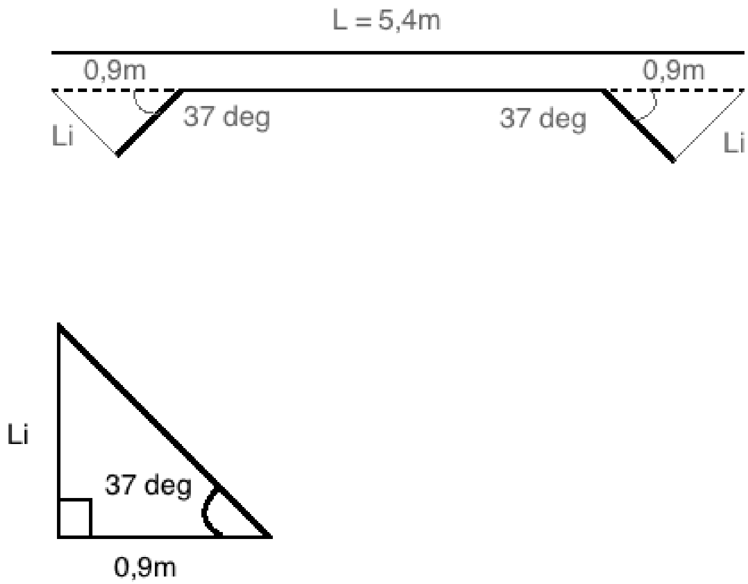

Figure 66: Illustrations explaining how the maximum percentage window opening was calculated for the south window

$$\begin{aligned}
tan\theta &= \frac{mkat}{hkat} \\
mkat &= hkat \times tan\theta \\
L_i &= 0,9 \times tan(37^o) \\
L_i &= 0,68m \\
x &= \frac{2 \times L_i}{L} \\
x &= \frac{2 \times 0,68}{5,4} \\
x &= 0,25
\end{aligned} \tag{16}$$

A.6.2 North window

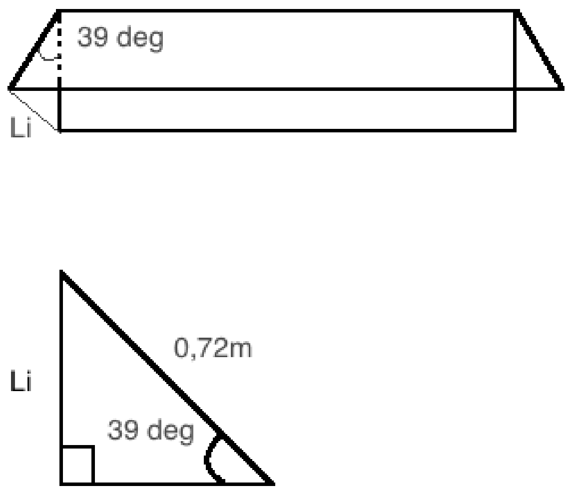

Figure 67: Illustrations explaining how the maximum percentage window opening was calculated for the north window

$$\begin{aligned}
sin\theta &= \frac{mkat}{hyp} \\
mkat &= hyp \times sin\theta \\
L_i &= 0,72 \times sin(39^o) \\
L_i &= 0,45m \\
x &= \frac{L_i}{L} \\
x &= \frac{0,45}{0,72} \\
x &= 0,625
\end{aligned} \tag{17}$$

A.6.3 Skylight windows

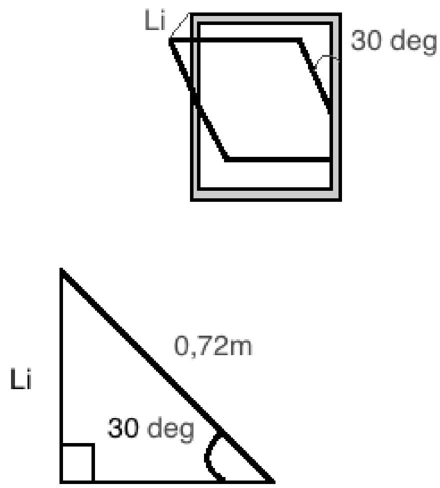

Figure 68: Illustrations explaining how the maximum percentage window opening was calculated for the skylight windows

$$
\begin{aligned}
sin\theta &= \frac{mkat}{hyp} \\
mkat &= hyp \times sin\theta \\
L_i &= 0,55 \times sin(30^o) \\
L_i &= 0,275m \\
x &= \frac{2 \times L_i}{2 \times L} \\
x &= \frac{2 \times 0,275}{2 \times 0,55} \\
x &= 0,5
\end{aligned} \tag{18}
$$

A.7 Climate files

Table 44: Values used when constructing the climate files

Hour	**Temperature**		**Humidity**	**Wind direction**
	10-15 Deg	15-20 Deg		
1	10	15	83	180
2	10	15	83	180
3	10	15	84	180
4	10	15	85	180
5	11	16	86	180
6	11	16	88	90
7	11	16	86	270
8	12	17	85	360
9	12	17	84	360
10	13	18	80	360
11	13	18	79	360
12	15	20	78	360
13	15	20	80	360
14	14	19	80	260
15	14	19	78	260
16	15	20	85	260
17	15	20	74	260
18	14	19	77	260
19	14	19	82	90
20	13	18	83	360
21	13	18	84	360
22	12	17	83	360
23	12	17	85	90
24	11	16	92	90

Table 45: Values used when constructing the climate files

Hour	**Wind speed**		**Direct radiation**		**Diffuse radiation**	
	Windless	Slightly windy	Clouded	Sunny	Clouded	Sunny
1	0.4	2.8	0	0	0	0
2	0.6	2.6	0	0	0	0
3	0.8	2.2	0	0	0	0
4	0.5	1.4	0	0	0	0
5	0.9	0.5	0	0	0	0
6	0.9	0.8	55	220	14	55
7	0.9	0.8	122	486	30	122
8	0.3	1.3	162	648	41	162
9	0.6	1.7	178	712	45	178
10	0.7	2.3	172	689	43	172
11	0.9	2.8	148	592	37	148
12	0.3	2.7	134	534	33	134
13	0.3	2	155	621	39	155
14	0.2	2.9	163	651	41	163
15	0.2	3.9	155	621	39	155
16	1.1	4.2	134	534	33	134
17	1	2.6	148	592	37	148
18	1.2	2.7	172	689	43	172
19	1.3	5.1	178	712	45	178
20	1	5.7	162	648	41	162
21	1.8	4.2	122	486	30	122
22	1.2	3.8	55	220	14	55
23	0.5	2.2	0	0	0	0
24	0.9	2	0	0	0	0

A.8 Control systems

A.8.1 On/off window control

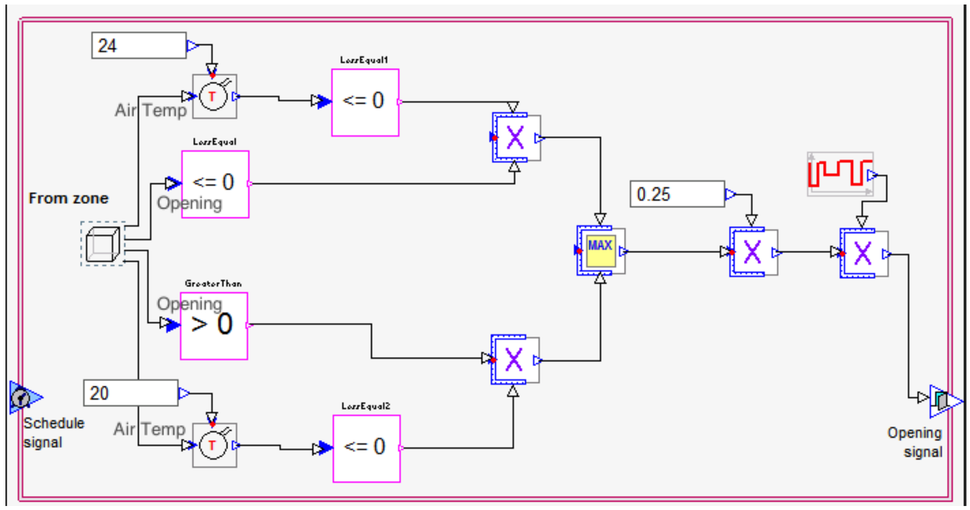

Figure 69: Example of construction of a on/off window control system in IDA ICE. Here: south window open at 24^{o}C, close at 20^{o}C

The window control system begins with a zone sensor sending out signals containing indoor air temperature and the current opening signal. If the indoor temperature is above 24^{o}C and the window is closed the top circuit sends a signal equal to 1 into the MAX block, if not it sends signal 0. If the temperature is above 20^{o}C and the window is open, the bottom circuit sends signal equal to 1 into the MAX block, if not 0. If either the window is closed and temperatures are above 24^{o}C, or the window is open and temperatures above 20^{o}C the MAX block sends out signal 1, which is multiplied with the maximum allowed opening of the window, in this case 25% for the south window. This signal is then multiplied with a schedule which depends on whether it is daytime, nighttime or all-hours ventilative cooling that is being applied. The result is that if allowed by the schedule, the window will open to 25% when indoor temperatures exceed 24^{o}C, and close when 20^{o}C is reached.

A.8.2 PI window control

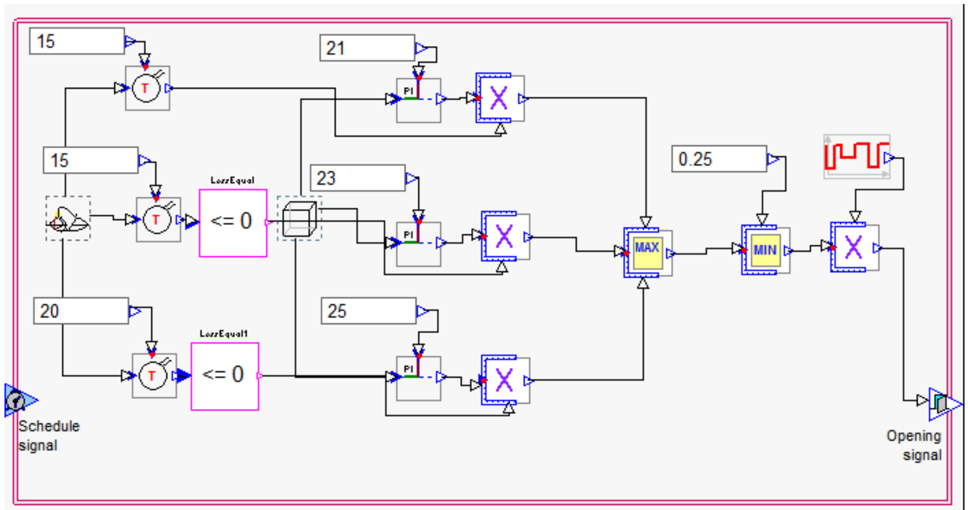

Figure 70: Example of construction of a PI window control system in IDA ICE. Here: south window keep 21^{o}C when below 15^{o}C outside, keep 23^{o}C when 15-20^{o}C outside, keep 25^{o}C when above 20^{o}C outside

The window control system begins with temperature sensors receiving signals from the ambient sensors containing outdoor air temperatures. If the outdoor temperature is below 15^{o}C, the upper circuit sends a signal equal to 1 to the top multiplying box, if not it sends 0. If the outdoor temperature is above 15^{o}C, the middle circuit sends a signal equal to 1 to the middle multiplying box, if not it sends 0. The bottom circuit sends out signal 1 if the outdoor temperature is above 20^{o}C, and 0 if its below. The PI regulators receives signals from the zone sensor containing air temperature and from the constant blocks containing set-point temperatures. The regulators sends out a signal between 0 and 1 indicating the opening that is needed to keep the indoor temperatures below the set-point temperature. All PI regulators were set to cooling mode, hence only signaling window opening if there is a risk of indoor air temperatures above the set-point temperature. When the signals from the PI regulator is multiplied with the signals from the outdoor temperature sensors, the outgoing signal is the one needed to keep the set-point temperature if the outdoor temperature is in the corresponding range, 0 if not. The MAX block then sends out a signal equal to the biggest input signal from the multiplying boxes. Hence, the top PI regulator is controlling if the outdoor temperature is below 15^{o}C, the middle is controlling if outdoor temperatures are between 15-20^{o}C and the bottom PI regulator is controlling if outdoor temperatures are above 20^{o}C. The outgoing signal from the MAX block is then reduced to the maximum allowed opening, 25% for the south window, if the initial signal was larger than this value, and kept if its below. This signal is then multiplied with the applied schedule, the same way as for the on/off system.

A.8.3 Enhanced mechanical control

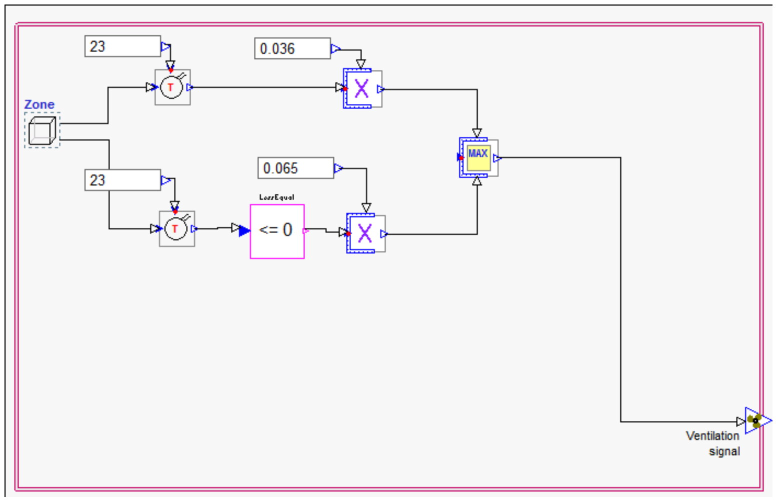

Figure 71: Example illustrating the construction of the enhanced mechanical control system in IDA ICE. Here: living room supply. All days are simulated as weekdays.

For hygienic ventilation 18% (26m^3/h of 144m^3/h) was supplied to bedroom west, 46% (66m^3/h of 144m^3/h) to the living room and 26% (52m^3/h of 144^3/h) to bedroom east. Using the same percentages resulted in supply of 47m^3/h, 119m^3/h and 94m^3/h for bedroom west, living room and bedroom east respectively. Extraction for hygienic ventilation was divided between 75% (108m^3/h of 144m^3/h) in the bathroom and 25% (36m^3/h of 144m^3/h) in the kitchen. Using the same percentages resulted in extraction of 195m^3/h and 65m^3/h in the bathroom and kitchen respectively. To acquire these airflow rates, the initial ventilation airflow rates in the base model of Living Lab had to be changed. The living room and the two bedrooms were assigned a maximum of 10L/sm^2 supply air and 0L/sm2 extract air. For the bathroom and kitchen a maximum of 0L/sm2 supply air was assigned, and 10L/sm2 extract air. Further, the control system was designed to send out a signal between 0 and 1 resulting in the right amount of supply or extraction when multiplied with 10L/sm2.

The enhanced mechanical ventilation control system begins with the zone sensor sending out air temperature signals to the temperature control block. The top circuit sends out the right signal for hygienic ventilation if temperatures are below 23^{o}C, 0 if above. The bottom circuit sends out the right signal for enhanced ventilation if temperatures are above 23^{o}C, 0 if below. The MAX block ensures that the controlling signal is used. For the living room this results in a signal of 0.036 if temperatures are below 23^{o}C and 0.065 if temperatures are equal to or above 23^{o}C. These signals represent 3.6% and 6.5% of the maximum supply and extraction airflows, which are set to 10L/sm2 and 0L/sm2 respectively. The results are supply airflows of 0.36L/sm^2 if temperatures are below 23^{o}C and 0.65L/sm^2 if temperatures are above 23^{o}C. Multiplied with the zone area of 50.6m^2 and converted to m^3/h the results are 66m^3/h and 119^3/h which coincides with hygienic and enhanced mechanical ventilation respectively.

A.9 Additional simulation results

A.9.1 On/off nighttime window control

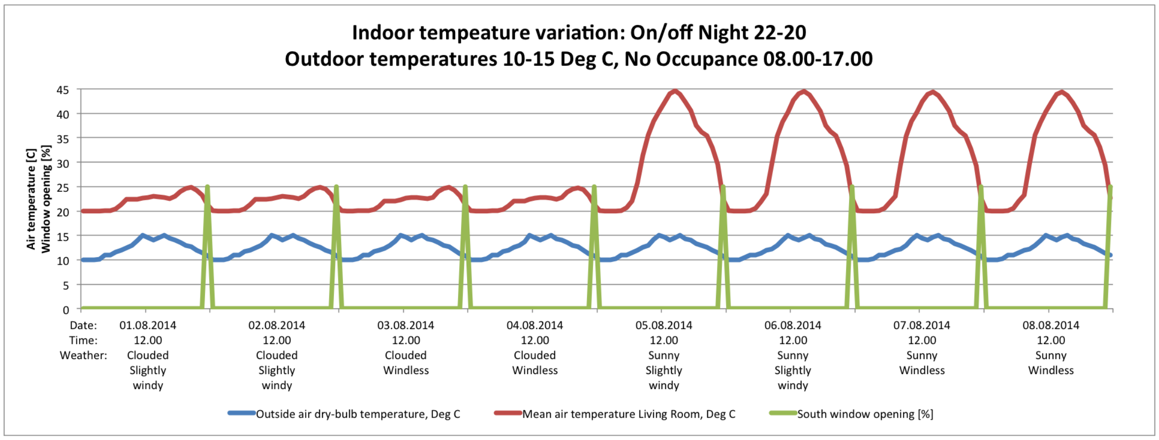

Figure 72: Window opening and indoor air temperature variation for the 20-22^{o}C on/off nighttime window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

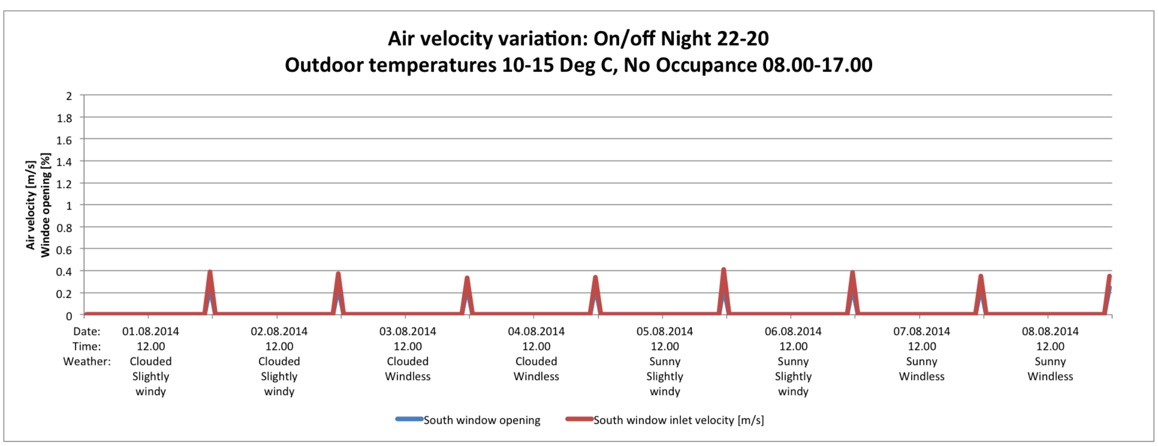

Figure 73: Window opening and indoor air velocities for the 20-22^{o}C on/off nighttime window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

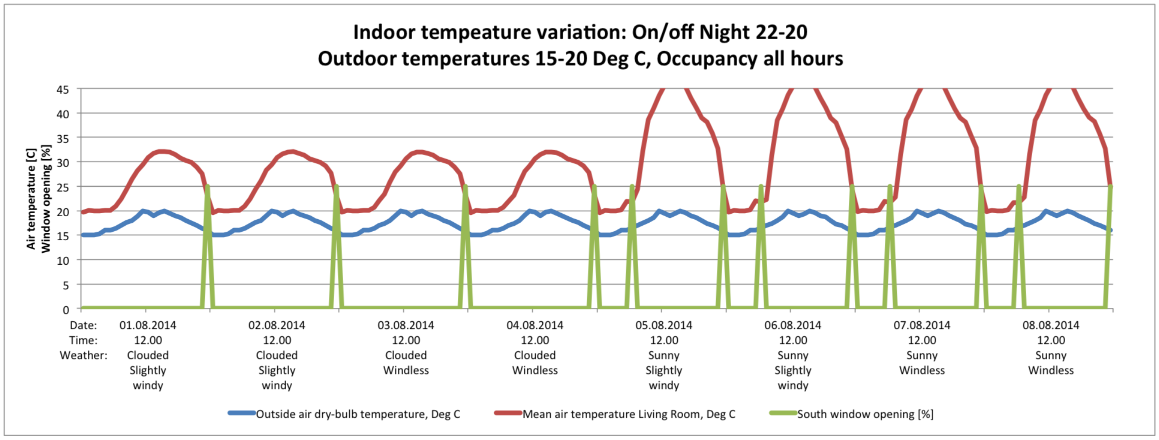

Figure 74: Window opening and indoor air temperature variation for the 20-22^{o}C on/off nighttime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

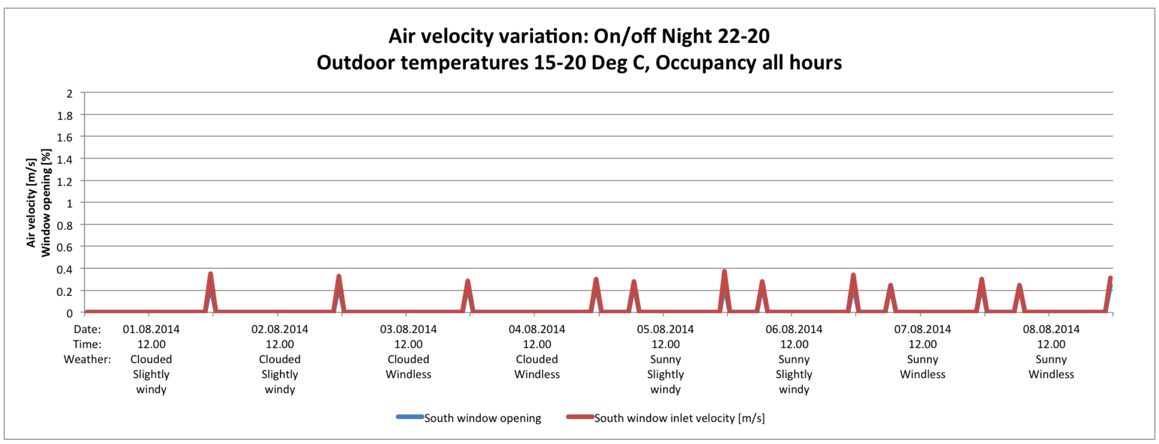

Figure 75: Window opening and indoor air velocities for the 20-22^{o}C on/off nighttime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

A.9.2 PI nighttime window control

Table 46: Results for the set-point temperature testing for the PI nighttime window control system. Time is given as percentage of the simulation period (768 hours).

	PI Night 20-20-21		PI Night 19-19-20	
	Result	Time [%]	Result	Time [%]
Overheating				
Max temp Living [^{o}C]	48.29		48.43	
H Tliving >24^{o}C	414	53.9	414	53.9
H Tliving >26^{o}C	367	47.8	363	47.3
H Tliving >28^{o}C	298	38.8	297	38.7
Max temp Loft [^{o}C]	47.44		47.56	
H Tloft >26^{o}C	386	50.3	382	49.7
Max temp BedroomW [^{o}C]	48.65		48.77	
H TbedroomW >26^{o}C	383	49.9	381	49.6
H overheated	***348***	***45.3***	***346***	***45.1***
Overcooling				
Min temp Living [^{o}C]	19.54		18.69	
H Tliving <20^{o}C	35	4.6	242	31.5
Min temp Kitchen [^{o}C]	19.44		18.57	
H Tkitchen <20^{o}C	180	23.4	251	32.7
H overcooled	***231***	***30.1***	***224***	***29.2***
Air velocities				
H south window open	304	39.6	304	39.6
Average velocity [m/s]	0.66		0.59	
Average velocity 2-5% [m/s]	0.58		0.53	
Max velocity 2-25% [m/s]	1.16		1.17	
H >0.5m/s 2-25%	65	8.5	90	11.7
H >1.0m/s 5-25%	12	1.6	9	1.2
Average velocity <2% [m/s]	0.73		0.57	
Max velocity <2% [m/s]	1.23		1.20	
H > 1.0m/s <2%	20	2.6	12	1.6
H > 2.0m/s <2%	0	0.0	0	0.0

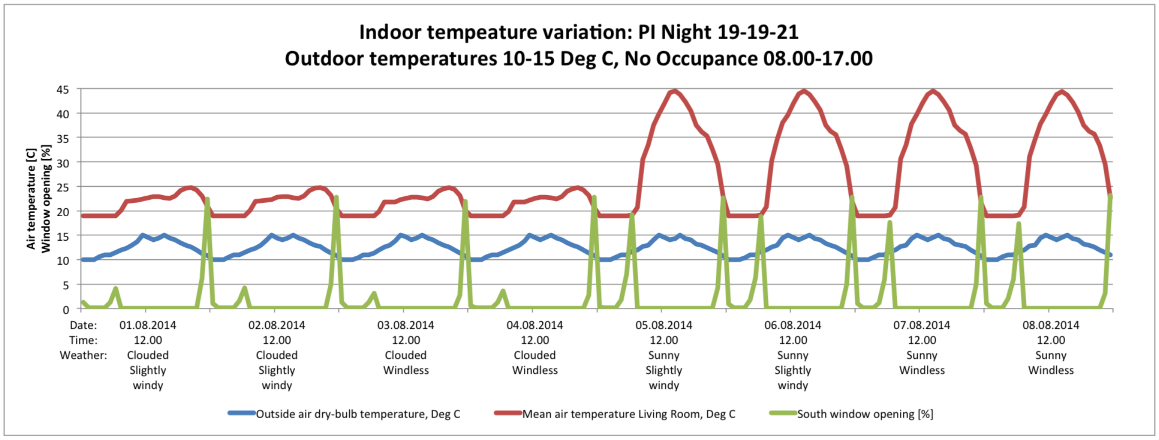

Figure 76: Window opening and indoor air temperature variation for the 19-19-21^{o}C PI nighttime window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

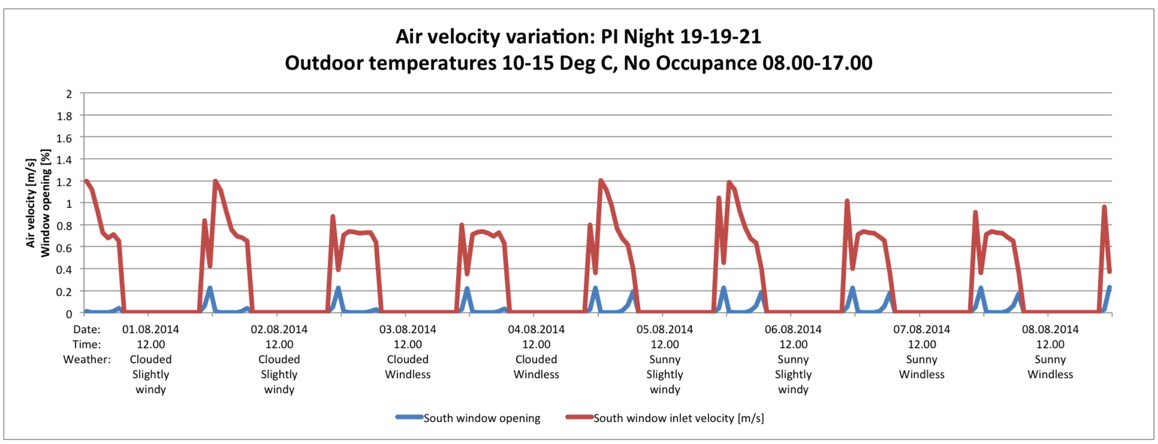

Figure 77: Window opening and indoor air velocities for the 19-19-21^{o}C PI nighttime window control set-point solution using outdoor temperatures 10-15^{o}C and occupancy all hours except from 08.00-17.00.

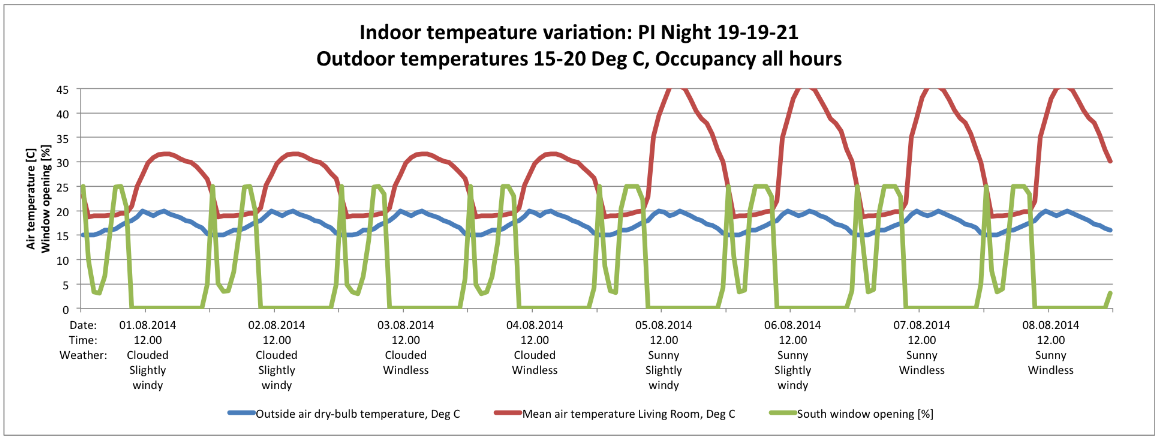

Figure 78: Window opening and indoor air temperature variation for the 19-19-21^{o}C PI nighttime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.

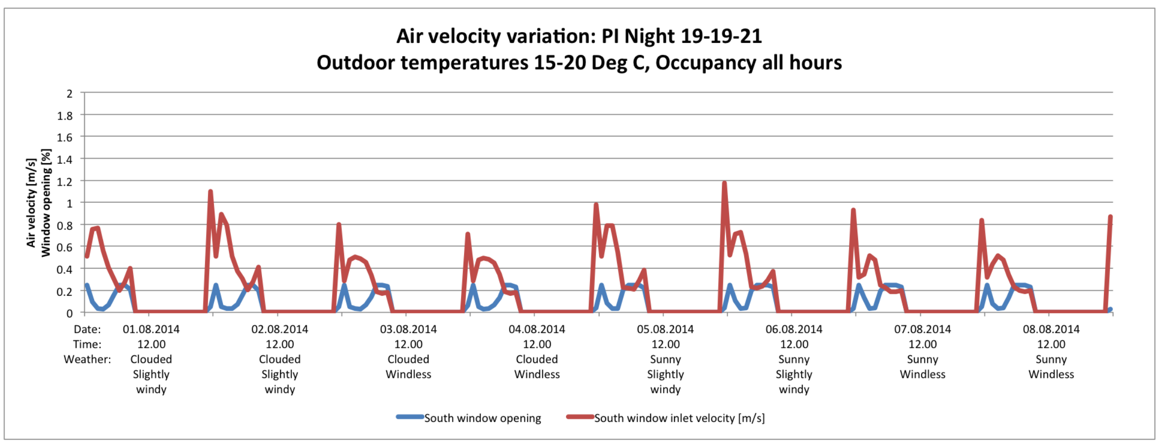

Figure 79: Window opening and indoor air velocities for the19-19-21^{o}C PI nighttime window control set-point solution using outdoor temperatures 15-20^{o}C and occupancy all hours.