Martin Sande

Ventilative Cooling Potential of the ZEB Laboratory

Based on simulations performed with IDA ICE

Master's thesis in Energy and the Environment Engineering
Supervisor: Hans Martin Mathisen
June 2021

Master's thesis

NTNU
Norwegian University of Science and Technology
Faculty of Engineering
Department of Energy and Process Engineering

NTNU
Kunnskap for en bedre verden

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Acknowledgments

With this thesis, I conclude the two-year MSc program, *Energy and the Environment Engineering* at the Department of Energy and Process Engineering, at Norwegian University of Science and Technology. The master thesis has been written in conjunction with the subject TEP4920, *Energy supply and air conditioning in buildings* during the spring of 2021 and accounts for 30 ECTS credits.

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Trondheim, June 11^{th}, 2021

Signature: Martin Sande

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Abstract

Well-insulated buildings, such as *Zero Emission Buildings* (ZEB), are subjected to a high occurrence of overheating during the cooling season. Cooling becomes necessary to achieve a good thermal environment. However, mechanical cooling has a high energy consumption and is not permitted by the Norwegian Standard criteria for passive house and low-energy buildings. Passive cooling becomes a necessary strategy in achieving a satisfactory thermal environment. The ZEB Laboratory is an office and educational building located in Trondheim Norway. It is an arena where new and innovative components and solutions are developed, investigated, tested, and demonstrated in mutual interaction with the occupants of the building. This master thesis investigates the possibility of using natural ventilation to supply ventilative cooling to the ZEB Laboratory. A control algorithm was created to supply ventilative cooling through windows during unsatisfactory thermal or atmospheric conditions. The controller setpoints were specified to ensure hygienic ventilation that provides thermal comfort to occupants. The potential of fan power reduction from mechanical ventilation use was also explored by investigating the potential for ventilating the ZEB Laboratory with clean natural ventilation.

The results of the conducted simulations conclude that a good thermal environment could be provided in the ZEB Laboratory through the cooling season, using ventilative cooling. A cooling demand of $1141\,\mathrm{kWh}$, corresponding to a power demand of $0.821\,\mathrm{kWh/m^2}$ was entirely removed with the implemented window control algorithm. The draught risk was evaluated and deemed minimal as ventilative cooling would mainly be utilized in periods with high indoor temperatures, periods where draught is assumed to provide a comfortable cooling effect.

Ventilating the ZEB Laboratory with clean natural ventilation resulted in an unacceptable indoor environment. The window control algorithm prohibits window operation during periods of an unacceptable ambient condition resulting in periods where the building was not ventilated. With the implementation of a mechanical ventilation control algorithm, a total of $68\,\%$ of hours of mechanical ventilation could be replaced with natural ventilation while achieving an good indoor environment, resulting in a fan power requirement of $3.58\,\mathrm{kWh}$. Clean natural ventilation of the ZEB Laboratory under the presented conditions was not an acceptable ventilative solution.

Due to uncertainty regarding draught risk in occupied zones and the potentially short-circuiting of the ground and third floor of the ZEB Laboratory, the hybrid ventilation solution with ventilative cooling supplied by window operation was chosen as the best ventilative cooling strategy for the ZEB Laboratory through the summer season.

Norwegian Summary

Velisolerte bygninger, som *Zero Emission Buildings* (ZEB), utsettes for en høy forekomst av overoppheting i kjølesesongen. Kjøling blir nødvendig for å oppnå et godt termisk miljø, men mekanisk kjøling har et høyt energiforbruk og er ikke tillatt i henhold til Norsk standardkriterier for passivhus og lavenergibygg. Passiv kjøling blir en nødvendig strategi for å oppnå et tilfredsstillende termisk miljø.
ZEB-laboratoriet er et kontor- og utdanningsbygg i Trondheim, Norge. Det er en arena der nye og innovative komponenter og løsninger utvikles, undersøkes, testes og demonstreres i gjensidig samhandling med beboerne i bygningen. Denne masteroppgaven undersøker muligheten for å bruke naturlig ventilasjon for å levere ventilasjonskjøling til ZEB-laboratoriet. En kontrollalgoritme ble opprettet for å levere ventilativ kjøling gjennom vinduer under utilfredsstillende termiske eller atmosfæriske forhold. Kontrollalgoritmens settpunkt verdier ble spesifisert etter å sikre hygienisk ventilasjon som gir termisk komfort til beboerne. Potensialet for reduksjon av viftebruk fra den mekanisk ventilasjonen ble også undersøkt ved å undersøke potensialet for å ventilere ZEB Laboratory med ren naturlig ventilasjon.

Resultatene av de utførte simuleringene vise til at et godt termisk miljø kan oppnås i ZEB Laboratory gjennom kjølesesongen ved bruk av ventilativ kjøling. Et kjølebehov på $1141\,\mathrm{kWh}$, tilsvarende et effektbehov på $0.821\,\mathrm{kWh/m^2}$, ble helt fjernet med den implementerte vinduskontrollalgoritmen. Trekkrisiko ble evaluert og ansett som minimal da ventilativ kjøling hovedsakelig ville bli brukt i perioder med høye innetemperaturer, perioder der trekk antas å gi en behagelig kjølende effekt.

Ventilering av ZEB Laboratory med ren naturlig ventilasjon resulterte i et uakseptabelt inneklima. Vinduskontrollalgoritmen forbyder vindusdrift i perioder med uakseptabel omgivelsestilstand, noe som resulterer i perioder der bygningen ikke ble ventilert. Med implementeringen av en mekanisk ventilasjonskontrollalgoritmen, kan totalt $68\,\%$ timer med mekanisk ventilasjon erstattes med naturlig ventilasjon mens det oppnås et godt inneklima, noe som resulterer i et vifteeffektbehov på $3.58\,\mathrm{kWh}$. Ren naturlig ventilasjon av ZEB Laboratory under de presenterte forholdene kan ikke anses som en akseptable løsning for ventilering av bygget.

På grunn av usikkerhet angående trekkrisiko i okkuperte soner og potensiell kortslutning av først of fjerde etasjene av ZEB Laboratory, ble hybridventilasjonsløsningen med ventilativ kjøling levert av vindusstyring valgt som den beste ventilative kjølestrategien for ZEB Laboratory i sommersesongen.

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Nomenclature

A	Area (m^2)
C	Convection
C_p	Pressure coefficient $(-)$
D_h	Hydraulics diameter (m)
E	Evaporation
K	Flow coefficient $(-)$
M	Heat gained from activity level
N_k	Total distinct leakages of air $(-)$
P	Pressure (Pa)
Q	Airflow rate (m^3/h)
R	Radiation
Re	Reynolds number $(-)$
S	Heat storage of the body
U	Air speed (m/s)
V	Volume (m^3)
W	External work rate
α_c	Heat exchange coefficient for convection $(-)$
α_r	Heat exchange coefficient for radiation $(-)$
ϵ	Ration between two openings on a facade $(-)$
ν	Kinematics viscosity (m^2)
ρ	Mass density (kg/m^3)
e	Euler's number $(-)$
g	Gravity (m^2/s)
h	Heat transfer coefficient $(-)$
m	Mass flow rate (kg/h)
n	Air exchange rate (h^{-1})
q	Heat exchange by convection $(-)$
t_a	Air temperature $(°C)$
t_o	Operative temperature $(°C)$
t_r	Radiant temperature $(°C)$
z	Height (m)

Ar	Archimedes Number $(-)$
$\mathrm{C_{res}}$	Respiratory convective heat loss
$\mathrm{E_{res}}$	Respiratory evaporate heat loss
$\mathrm{Q_v}$	Ventilation rate (m^3/h)
ϵ^{c}	Containment removal effectiveness $(-)$
$\langle \mathrm{c} \rangle$	Mean contaminant concentration $(-)$
$\langle \overline{\tau} \rangle$	Mean age of air $(-)$
$\langle \varepsilon_{\mathrm{a}} \rangle$	Air change efficiency $(-)$
$\overline{\tau_{\mathrm{r}}}$	Residence time of air $(-)$
τ_{n}	Nominal time constant $(-)$
τ_{n}	Shortest possible time of air exchange $(-)$
τ_{r}	Actual time of air change $(-)$
$\mathrm{c_e}$	Contaminant level in the exhaust $(-)$
l	Characteristic length (m)
x	Separation distance (m)

Abbreviations

AHU	Air Handling Unit
AI	Artificial Intelligence
CAV	Constant Air Volume
CBE	The Center for the Built Environment
DCV	Demand Control Ventilation
FME	Environment-friendly Energy Research
HVAC	Heating Ventilation and Air Condition
IAQ	Indoor Air Quality
IDA ICE	IDA Indoor Climate and Energy
IEA EBC	The International Energy Agency Energy in Buildings and Communities
IESNA	Illuminating Engineering Society of North America
IEQ	Indoor Environmental Quality
MMV	Mixed Mode Ventilation
MPC	Model Predictive Control
NTNU	Norwegian University of Science and Technology
nZEB	Nearly Zero Emission Building
PMV	Predicted Mean Vote
PPD	Predicted Percentage Dissatisfied
SBS	Sick Building Syndrome
TEK	Byggtekniske forskrifter
VAV	Variable Air Volume
WHO	World Health Organisation
ZEB	Zero Emission Building

1. Introduction

Global climate change has become more noticeable over the past decade. Effects of global warming that scientists have predicted are starting to occur: rising sea levels, longer and more intense heatwaves in different parts of the world, and an increasing loss of sea ice [1]. To counteract this increasing threat, a reduction in greenhouse gas emission must be met, which means; reduction of energy consumption and a more considerable focus and use of renewable energy sources.

Increased efficiency of energy use in Norway has helped limit the total energy consumption over the past years. To be within the limits of the Paris agreement, Norway must reduce the total CO_2 emission by 55% by 2030, compared to values recorded in 1990. This is equivalent to 23,2 million tons of greenhouse gas [2]. The recorded greenhouse gas emission of the past 30 years and the emission reduction suggested to meet the Paris agreement is illustrated in Figure 1.1. 40% of the total energy consumption of Norway is used by the building sector and is utilized mainly for heating, cooling, and ventilating of buildings [3]. Indicating that the reduction in energy consumption and an increase in the energy efficiency of the building sector will be crucial to fulfill the Norwegian obligations to the UN [4].

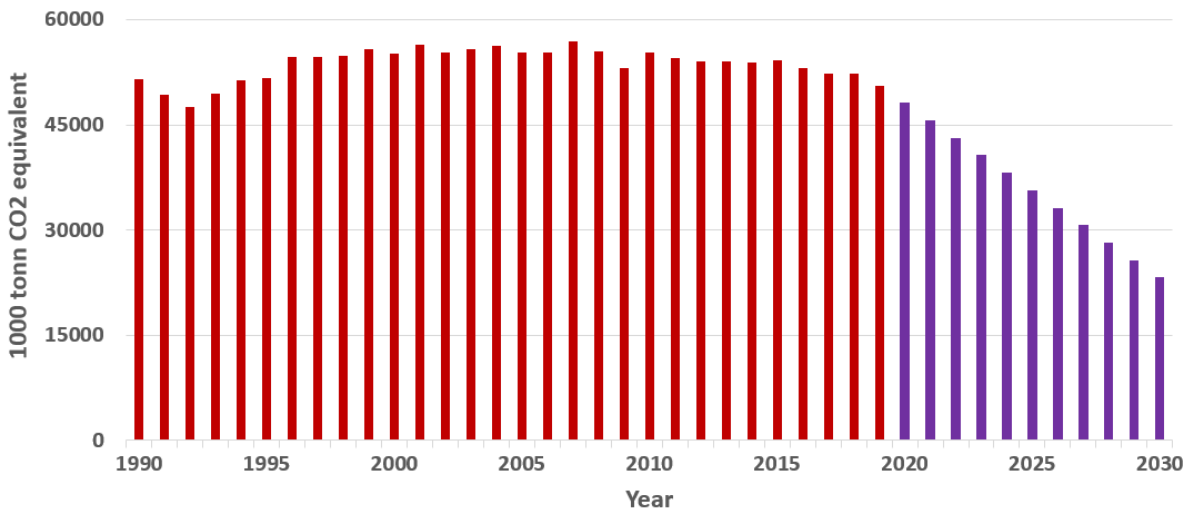
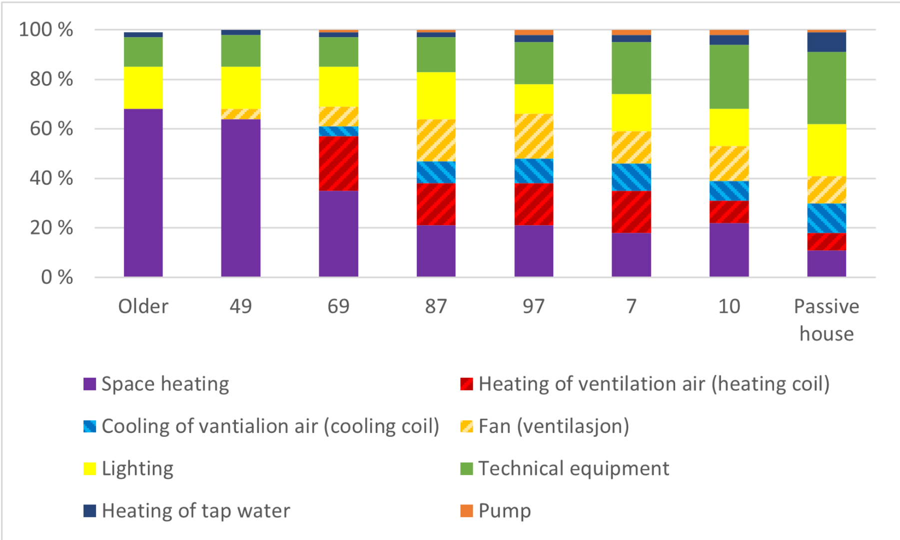


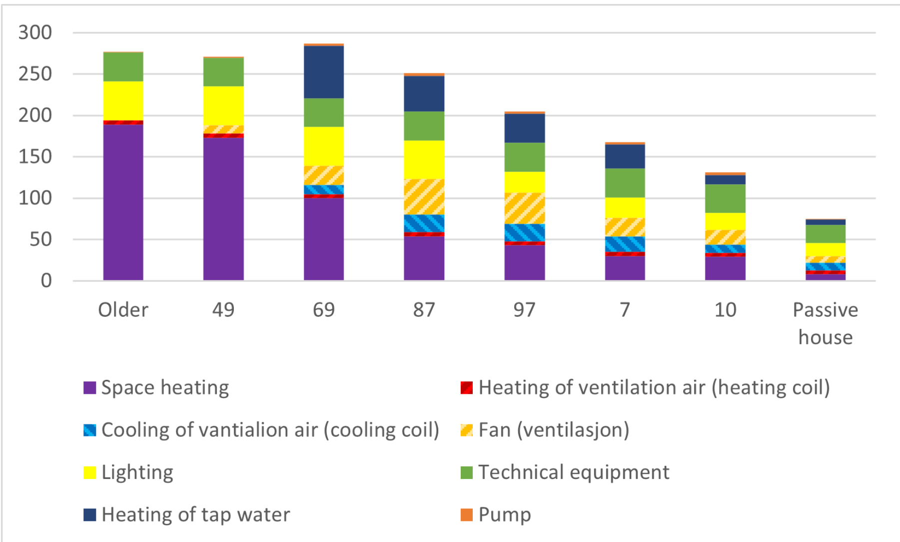
Figure 1.1.: *Greenhouse gas emission in Norway of the last 30 years (red bars), with suggested reduction in emission to meet the Paris agreement by 2030 (purple bars) [2]*

A transition to energy-efficient buildings is among the most profitable measures to decrease greenhouse gas emissions. Zero Emission Buildings aim to produce an amount of on-site renewable energy that compensates for the greenhouse gases emitted throughout the building's lifetime [5]. A reduction of six million tons of greenhouse gases annually can be achieved with development towards energy-efficient buildings, such as ZEB [4].

The graphs presented below in Figure 1.2 show the expected energy consumption in different sectors of a building, for all past versions of TEK, the guidance on technical requirements for buildings. The values are presented in both as a percentage of the total energy consumption and $\mathrm{kWh/m^2}$. The graphs indicate that space heating has been the focus of energy reduction over the past versions of TEK, resulting in a higher percentage of energy consumption of the other sectors. A steady increase in energy requirements for cooling is also shown as buildings become increasingly insulated and air-tight. It is shown in Figure 1.2a that the expected energy consumption from ventilating buildings (The areas of the columns marked with strips) with passive house standards is about 29% of the total energy requirement, counting heating and cooling of ventilated air and fan operation. A natural step for continued improvement and reduction of energy consumption of modern buildings would be to streamline the ventilation system. [6]



(a) *Energy use in percent*



(b) *Energy use in* $\mathrm{kWh/m^2}$

Figure 1.2.: *Energy use in office buildings divided into different sectors, expressed in percentage, Figure (a) and in* $\mathrm{kWh/m^2}$, *Figure (b) [6]*

1.1. Background

SINTEF is one of Europe's leading research institutes, with multidisciplinary expertise in technology, natural sciences, and social studies [7]. They have a close collaboration with NTNU, and together they focus on the development of strategies and increased insight into how to reduce energy consumption while taking climate and environmental considerations into account. *The Research Centre on Zero Emission Buildings* is an example of one of the FME - projects (The Research Centers for Environmental Friendly Energy) they have created [8]. The ZEB Laboratory will be a central part of a new FME - project. *Research Centre on Zero Emission Neighbourhoods in Smart Cities (ZEN Centre).*

The ZEB Laboratory is an arena where new and innovative components and solutions are developed, investigated, tested, and demonstrated in mutual interaction with the building's occupants. It is located in Trondheim at NTNU Gløshaugen campus, close to the existing facilities of SINTEF Community and NTNU Department of Civil and Environmental Engineering. The building will form a living laboratory that continuously collects experimental data while the building is used as an ordinary office building. The building's facades, components, and technologies can be modified and replaced, which gives the building the adaptability to investigate different building configurations, technologies, and usages that can be implemented in other designs and constructions for Zero-emission buildings [9].

ZEBs are usually highly insulated, which leads to a naturally high occurrence of overheating. Cooling becomes required to meet acceptable levels of thermal comfort for occupants in the building. Mechanical cooling has a high energy requirement and is not permitted by the Norwegian Standard criteria for passive house and low energy buildings - Residential buildings [10]. Passive cooling strategies like ventilative cooling becomes a necessary strategy in achieving an acceptable thermal environment. Northern climates have a considerable ventilative cooling potential due to the generally colder ambient conditions. However, the low temperatures may lead to local thermal discomfort from draught if natural ventilation is not utilized correctly. A carefully designed ventilative cooling system becomes essential.

The Project Thesis, *Natural Ventilation in ZEB Laboratory*, was completed during the fall of 2020. This report contains preliminary work to this master thesis and researched ventilation strategies and mathematical models of natural ventilation while investigating IDA Indoor Climate and Energy (IDA ICE) as a potential suitable simulation tool for the planned work of the master thesis. The report concluded that IDA ICE should be the chosen simulation tool for the master thesis. The project thesis literature study overlaps with this master thesis. Therefore, a large part of the presented theory and literature is based on literature and knowledge obtained during the previously completed Project thesis.

1.2. Scope

The goal of the master thesis is to investigate the ventilative cooling potential of the ZEB Laboratory. No mechanical cooling is installed in the building, and ventilative cooling is chosen as the cooling strategy. The primary goal is to achieve a healthy indoor environment that provides thermal comfort for occupants and hygienic ventilation. A window control algorithm has been created that optimally utilizes natural ventilation when the mechanical ventilation system cannot achieve a good thermal environment. The second goal is to investigate the potential fan power reduction by reducing the use of the mechanical ventilation system when natural ventilation can provide satisfactory indoor conditions. An earlier master thesis by Maren Elise Leinum, investigating hybrid ventilation of the ZEB Laboratory concluded that the ZEB Laboratory could achieve a good indoor environment using clean natural ventilation through the summer season [11]. When investigating the possibility of fan power reduction, the potential of clean natural ventilation of the ZEB Laboratory was investigated first.

An earlier master student at NTNU, Andrea Elisabeth Holltrø Søraas has constructed a realistic IDA ICE model of the ZEB Laboratory as a part of her master thesis. The model was controlled check and edited by SINTEF throughout the work of the project thesis and was ready for use in the early stages of the spring of 2021. Initially, there was an intention to implement the proposed ventilative cooling strategy presented in the thesis and test its performance with the realistic model. However, due to the model only working in the 5.0 version of IDA ICE, combined with technical difficulties, this was not possible.

The ZEB Laboratory model created for this thesis was modeled as a replica of the real building but with limited zone division to lower computation time and complexity of the conducted simulations. Due to the unrestricted airflow nature of the ZEB Laboratory floors, this simplification was deemed acceptable. However, this decision was chosen with the intention of implementing the resulting strategy to the realistic model. A model with more realistic zone division would have been created and utilized for the thesis if the complication was discovered earlier in the master thesis work period.

The research question of the master thesis is as follows:
To what degree can ventilative cooling reduce the cooling demand of the ZEB Laboratory while maintaining an acceptable and satisfactory indoor environment, and is clean natural ventilation an acceptable ventilation strategy for the ZEB Laboratory?

1.3. Structure of the Report

This report is split into three main parts. The report starts with presenting the finding of the theory and literature review. The purpose of this is to establish a theoretical basis of indoor climate and thermal comfort, different ventilation strategies, and the basic fluid mechanics of natural ventilation. The literature review explores earlier findings of the adaptive thermal comfort model, window automation, and energy-efficient ventilation strategies focusing on ventilative cooling. The presented literature will act as the foundation for the evaluated results and discussion. Further, information regarding the ZEB Laboratory will be presented. Including a theoretical basis of ZEB, building body, occupancy, and the implemented ventilation strategies.

Secondly, a presentation of the modeled ZEB Laboratory and the method of the conducted simulations. This includes an introduction to the chosen simulation tool IDA ICE, a description of the structured model, the parameter that defines its operation, the constructed window opening algorithm for ventilative cooling of the ZEB Laboratory, and a summary of how the results will be presented and evaluated.

Furthermore, the results and discussion of the simulations will be presented. The results will be evaluated based on the thermal and atmospheric environment's resulting quality and energy consumption. A short discussion will be presented with the results of the conducted simulation scenario, and a more general discussion of the best solution will be presented in its chapter.

Lastly, a conclusion that summarizes the most important results of the literature review and conducted simulations, and presents the best solution for ventilative cooling of the ZEB Laboratory will be presented. Further work will be presented at the end of this thesis.

2. Indoor Environment

A building's primary purpose is to provide shelter from the weather and give a comfortable environment for occupants. Humans spend up to 90% of their life indoors, so the indoor environment must be of high quality. A poor indoor climate may have a negative effect on health, well-being, and the productivity of occupants. According to the World Health Organisation (WHO), the indoor environment is defined as the thermal, atmospheric, acoustic, actinic, mechanical, aesthetic, and psycho-social environment [12]. The first five factors mentioned affect how occupants perceive the indoor climate of a building. These factors with their most relevant traits are presented in Table 2.1 below. [13, 14, 15]

Table 2.1.: *The factor that influence indoor climate*

Thermal Environment	Heat balance, draught and humidity
Atmospheric Environment	Pollutants, air quality and fresh air volume
Acoustic Environment	Noise and the perception of speech and sound
Actinic Environment	Lighting, radiation, electric fields and magnetic fields
Mechanical Environment	Ergonomics, anti-slip and vibration

This chapter of the report will mainly present how the thermal and atmospheric environments affect occupants. These factors directly influence the thermal comfort of occupants and how they perceive the Indoor Air Quality (IAQ), and are directly affected by ventilation.

2.1. Thermal Environment

A thermal environment of high quality is essential for the well-being of occupants. The human body is comfortable within a small range of core body temperatures and is therefore quite sensitive to the influence of heat from the surroundings. In this section, the concept of and the parameters that affect thermal comfort will be presented.

2.1.1. Thermal Comfort

Human thermal comfort is defined in NS-EN ISO 7033:2005 as:

"The condition of mind that expresses satisfaction with the thermal environment". [16]

Thermal comfort is an individual experience, and thus, the conditions required for an occupant to achieve comfort may vary from person to person. Physical demand, physiological status, and psychological attitude must be considered. Comfort is a state of mind which makes it difficult to classify the factors that affect thermal comfort. [17]

2.1.2. Impact on Human Heat Balance

Sustaining a stable core temperature involves an energy balance between the body's heat loss and heat production. When the energy balance is in equilibrium, heat is gained at the same rate as it is lost. This can be expressed mathematically using the first law of thermodynamics, the law of conservation. *"The total energy of an isolated system is constant"* [18]. Equation 2.1 expresses the human body's thermal balance. [17]

$$\mathrm{S} = \mathrm{M} - \mathrm{W} - (\mathrm{C_{res}} + \mathrm{E_{res}} \pm \mathrm{C} \pm \mathrm{R} + \mathrm{E}) \tag{2.1}$$

Where S is the heat storage of the body, M is the heat gained from the activity level of the person, W is the external work rate, $\mathrm{C_{res}}$ and $\mathrm{E_{res}}$ is the respiratory convective and evaporative heat loss, C and R is the heat gain or loss from convection and radiation and E is the heat loss from evaporation of sweat.

Fanger established in the book *"Thermal Comfort. Analysis and applications in environmental engineering"* that six fundamental factors define the human thermal environment [19]. Air temperature, thermal radiation, air velocity, and humidity are defined as fundamental environmental factors, and clothing and metabolic rate are defined as behavioral factors. [14, 17, 20]

Air temperature

Air temperature is defined as the measured temperature of air without factoring radiance from the surroundings. This is called the dry-bulb temperature and can be measured using a dry-bulb mercury thermometer. According to the *"Veiledning om Klima og luftkvalitet på arbeidsplassen"* also called *"Guidance 444"*, the recommended air temperature should not exceed $22\,^{\circ}\mathrm{C}$, especially during heating seasons [21]. The effects air temperature has on occupant's performance and well-being has been widely studied by, among others, David P. Wyon in numerous articles [22, 23, 24]. Stating that both too high or too low air temperature may reduce the performance of occupants and increase the possibilities of mistakes and accidents within a workplace. The gradient temperature is the rate of change in temperature with distance. A temperature difference larger 3 - 4 $^{\circ}\mathrm{C}$ between the head and feet of an occupant will cause a sensation of displeasure. [14, 20]

Radiant temperature

Radiant temperature is the exchange of sensible heat between a person and surrounding surfaces. The radiance exchange is dependent on the temperature difference between the human body and the surface. Heat transfer will occur as long as $\Delta\mathrm{T}$ is larger than zero. Occupants will experience heating or chilling sensation depending on if the surrounding surfaces are hotter or colder than that of the human body. This local heating or cooling may cause discomfort. [14, 20]

Humidity

Humidity can be expressed as relative or absolute humidity. The relative humidity is the amount of water vapor as a function of the dry air and absolute humidity being the amount of water vapor. Absolute humidity impacts heat loss through evaporation for a person and is expressed as the water vapor pressure in the air. Relative humidity will have little to no effect on the perceived indoor climate if it stays between 20 - $60\,\%$. At relative humidity levels below $20\,\%$, occupants may experience irritated mucous membranes, dry eyes, and skin. In addition, dehydration of materials may occur over time. For relative humidity levels above $70\,\%$, an occupant's sweat production increases, and the chances for mold in buildings increase. In addition, the operative temperature of the air will be perceived as higher. [14, 20]

Draught

Draught is defined as "an undesired cooling of the human body caused by air movement". Draught can also be a combination of air velocity, low air temperature, and radiance from surfaces with low temperatures. The air velocity in an occupied zone should generally not exceed $0.30\,\mathrm{m/s}$ to avoid discomfort, although this may vary with indoor temperature. The placement of air supply should be carefully considered to minimize the air velocity within occupied areas. Draught can also be caused by air descending after being exposed to cold surfaces like windows. Tactical placement of a heat supply like a radiator can be used to combat chilled descending air. [14, 20]

Clothing Insulation

Clothing has a large impact on how occupants experience indoor climates. Clothing essentially works as insulation for the body and helps reduce and regulate energy transfer between the body and the surroundings. The level of insulation can be described by the unit of *clo*, where on *clo* equals, $0.115\,\mathrm{m^2K/W}$. Isolation values for different clothing combinations are presented in Table 2.2. Small changes in insulation levels can greatly impact human comfort because humans are susceptible to changes in the body's heat balance. [14, 20]

Table 2.2.: *Heat resistance in different sets of clothing [20]*

Clothing	**Insulation**	
	$\mathrm{m^2 \cdot K/W}$	**clo**
Shorts, underwear, t-shirts, socks, sandals	0,005	0,30
Lightweight dress with sleeves, petticoat, tights, panties	0,070	0,45
Light trousers, short-sleeved shirt, underpants, light socks, shoes	0,08	0,50
Skirt, short-sleeved shirt, panties, tights, sandals	0,095	0,6
Skirt, sweater with round neck, shirt, panties, thick knee socks	0,140	0,9
Jacket, trousers, shirt, underpants, socks, shoes	0,155	1,0
Coat, jacket, vest, trousers, shirt, short underwear, socks, shoes	0,230	2,50

Activity Level

The heat production of the human body is dependent on the activity level of the person. It is called the metabolic rate and is measured in the unit *met*, which is equivalent of $58\,\mathrm{W/m^2}$ and is the heat production of the human body when in a sedentary or relaxed state. As shown in Table 2.3 the heat production increases with the intensity of the activity. The presented values are not exact and may vary with age, health, gender, and from person to person. [14, 20]

Table 2.3.: *Heat production during various activities [20]*

Activity	**Heat production**	
	$\mathrm{W/m^2}$	**Met**
Laying down resting	46	0,8
Sedentary, relaxed	58	1,0
Sedentary activities (Office, school, lab)	70	1,2
Standing, light activity (Store, light industry)	93	1,6
Standing, medium activity (Industry)	116	2,0
Walking speed:		
- 2 km/h	110	-
- 3 km/h	140	-
- 4 km/h	165	-

2.1.3. Operative Temperature

Operative temperature is a constructed temperature measurement based on air temperature and radiation exchange between an occupant's body and surrounding surfaces. Operative temperature is the experienced temperature of an occupant, and it depends on the factors that impact the heat balance of the human body presented earlier in this section. Equation 2.2 below shows how operative temperature is calculated. [14, 20]

$$\mathrm{t_0} = \frac{\alpha_\mathrm{c} \cdot \mathrm{t_a} + \alpha_\mathrm{r} \cdot \mathrm{t_r}}{\alpha_\mathrm{c} + \alpha_\mathrm{r}} \tag{2.2}$$

Where $\mathrm{t_0}$ is the operative temperature, $\mathrm{t_a}$ is the air temperature, $\mathrm{t_r}$ is the radiation temperature, α_r is the heat exchange coefficient for radiation and α_c is the heat exchange coefficient for convection.

If the air velocity is lower than $0.2\,\mathrm{m/s}$ or the difference between the radiance temperature and the air temperature is less than $4\,^\circ\mathrm{C}$. The difference between air and operative temperature becomes so small that there will be no significant difference. In most cases, the value of convection can be assumed equal to the radiation value without substantial errors occurring, in which case, $\alpha_\mathrm{c} = \alpha_\mathrm{r}$. Equation (2.3) is assumed. [14]

$$\mathrm{t_0} = \frac{\mathrm{t_a} + \mathrm{t_r}}{2} \tag{2.3}$$

The operative temperature is primarily influenced by clothing and the heat production of the body. "*Byggteknisk forskrifter*, TEK17 recommends that the air temperature be kept below 22 °C as far as possible during the heating season. The temperature should be adapted to the function and use of the room, and possibilities for individual control options be available. Table 2.4 below presents the recommended values for operative temperature for workloads.

Table 2.4.: *Values of operative temperature for different work loads recommended by TEK17*

Activity Group	Light work load	Medium work load	Heavy work load
Operative temperature (°C)	19 - 26	16 - 26	10 - 26

Indoor operative temperatures exceeding the maximum limit are only found acceptable during hot summer periods were the ambient temperature exceeds 24 °C. Up to 50 hours in an ordinary year where indoor temperatures exceed 26 °C are accepted outside these conditions.

NS-EN 15251:2007 purposes a range of acceptable 'summer' indoor temperatures (cooling season) for buildings without mechanical cooling systems. It is stated that the range of operative temperatures presented in Figure 2.1 are valid for office buildings. Other buildings of a similar type are used mainly for human occupancy with mainly sedentary activities and dwellings. There is easy access to operable windows, and occupants may freely adapt their clothing to indoor or outdoor thermal conditions. The ZEB Laboratory is seen as a category II building, as will be explained in Section 2.1.4.

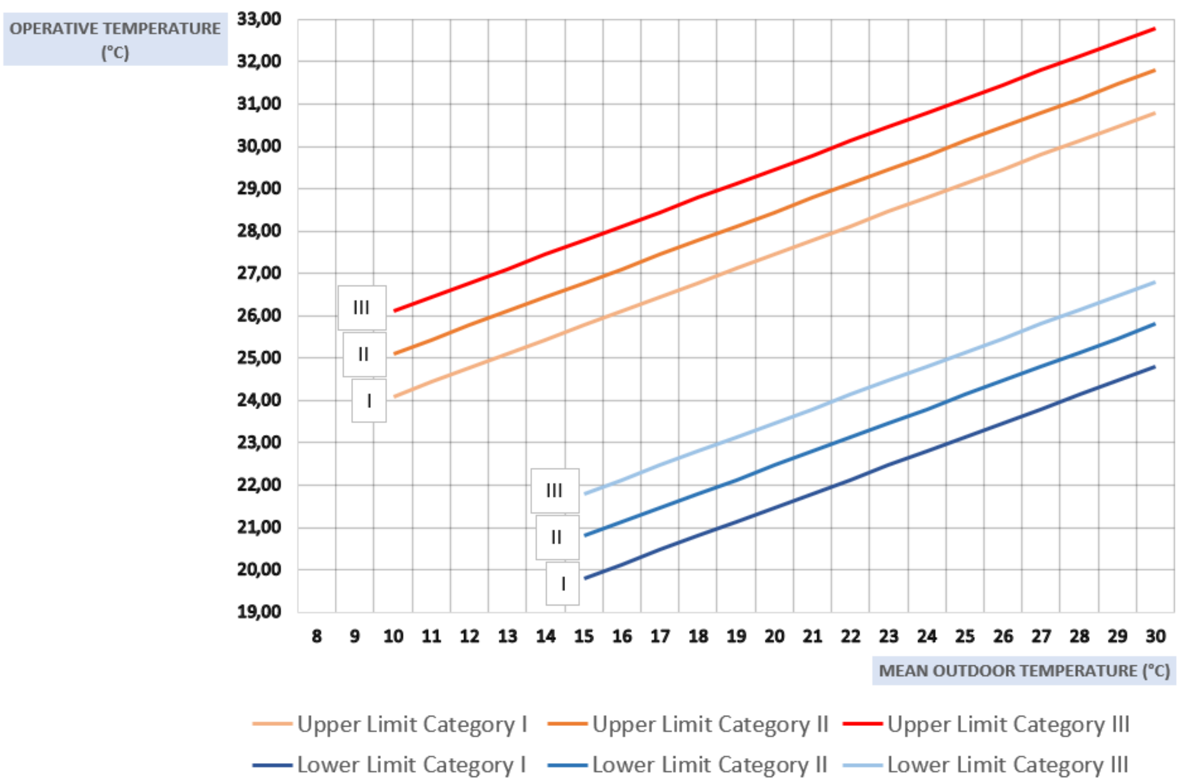


Figure 2.1.: *Acceptable indoor temperatures during summer/cooling season for buildings without mechanical cooling systems.*

It is important to note that for landscaped (open plan) offices, the temperature limits presented above may not be entirely accurate as not all occupants will have the same access to the operable windows, resulting in a lower sensation of control over the environment and a narrower temperature range of comfort.

2.1.4. Thermal Comfort Assessment

Fanger's Model for Thermal Sensation

The work of Fanger [19] defines the heat balance of an occupant in an indoor environment as personal and environmental parameters affecting the heat exchange between the human body and the surrounding area. PMV (Predicted Mean Vote) and PPD (Predicted Percentage Dissatisfied) are the two comfort indexes used to indicate occupant's thermal perception in a randomly chosen group of people in a given thermal environment. These indexes are based on empirical examination and are seen as the standardized method to evaluate indoor climate. The primary international standards for determination of indoor comfort conditions [16, 25, 26] refers to this method for assessing the quality of a thermal environment.

PMV-index uses a scale where individuals in a group can pinpoint their satisfaction or dissatisfaction with the thermal environment. It is a seven-step scale from -3 to 3, where 0 expresses thermal neutrality. -1 / +1 expresses slight dissatisfaction caused by light cooling or light heating while the others (-3,-2, 2, 3) express dissatisfaction with increased proportions.

PPD is an index that estimates the probable percentage of occupants that are dissatisfied with the thermal environment with a given activity level (*met*) and insulation level (*clo*). With known PMV-values, the PPD can be calculated using Equation 2.4, given by NS-EN 7730. Note that with a PMV-value of 0, there will still be an expected 5% dissatisfaction among occupants. [14, 16, 20, 27]

$$PPD = 100 - 95 \cdot \exp^{(0{,}03353 \, \cdot \, PMV^4 \, - \, 0{,}2179 \, \cdot \, PMV^2)} \tag{2.4}$$

The acceptable level of PMW and PPD is dependent on the *category* of the building. Table 2.5 presents the different building categories as given by NS-EN 15251. The ZEB Laboratory falls under category II.

Table 2.5.: *Building categories (NS-EN 15251:2007)*

Category I	Highest level of expectation. For building and zones used by fragile or sensitive people like young children, sick people, or elderly.
Category II	Standard comfort class. Used in new or rehabilitated buildings.
Category III	Acceptable in already existing buildings.

NS-EN 15251 specifies that a building will have met the criteria for their respected building category when rooms that constitute $95\,\%$ of the hours of occupancy do not stray from the acceptable indoor parameters for more the $3\,\%$ of the time of occupancy every day, week, month and year. Table 2.6 presents the acceptable time for deviation.

Table 2.6.: *Allowed deviation of indoor environment parameters (NS-EN 15251:2007)*

	Daily	**Weekly**	**Monthly**	**Yearly**
Allowed deviation time	43 minutes	5 hours	22 hours	259 hours

Though Fanger's model is one of the most widely used models for comfort prediction, it is not always the most accurate tool for every situation. Difficulties regarding real clothing and activity level estimations cause discrepancies between the actual and predicted thermal sensation. Different studies have also discovered that the model was better for predicting thermal sensation for mechanically conditioned buildings rather than naturally ventilated ones, as the model does not factor in outdoor temperature psychological effects of opportunities for adaptation for occupants [28]. The model was developed in controlled laboratory conditions where occupants were considered passive subjects of the climate, which left little consideration of the possibility that people may naturally adapt to a broader range of thermal environments in a more realistic real-life setting [29].

Adaptive Model

For the past two decades, an increasing number of thermal comfort research and citations have been registered [30]. With the ever-increasing focus on climate change and decarbonizing of the building sector, the way thermal comfort is delivered to occupants is changing. Voluntary or mandatory greenhouse mitigation strategies av been rolled out in various jurisdictions around the world. There has been a shift away from the physical-based determinism of Fanger's comfort model towards an adaptive model. The adaptive model predicts that contextual factors and past thermal history (weather, solar radiation, wind) will modify occupant's expectations and preferences towards thermal comfort in buildings. De Dear and Brager [31] brought the model into the mainstream in 1998 with the "ASHRAE Transactions paper", which is to date one of the most cited papers on the topic of adaptive thermal comfort. The paper concludes that occupants in naturally ventilated buildings were tolerant of and often preferred a significantly broader range of indoor temperatures. This was explained by a combination of both behavioral adjustments and psychological adaptations by occupants. Research has shown that the occupant's ability to interact with the building and its systems is a significant determinant for the occupant's satisfaction and response to the thermal environment. Accounting for these broader adaptive mechanisms allows for optimized design and operation of buildings that increase thermal comfort and reduce energy use [29]. Occupancy control is now included in the indoor environment quality section of various sustainability tools, such as the US and Australian Green Building Council's rating tool, LEED and Green Star, and the UK's BREEAM.

Human Response to Air Movement

There has also been a shift in how air movement/draught is related to thermal comfort over the past 20 years. At the start of the period, relatively large air movement was considered harmful in most situations, described as a draft. The draught model of Fanger's is frequently used to assess occupant's reaction to draught but suffers from the same problems as the thermal comfort model. It can be confidently applied to situations where the occupants are wearing regular indoor clothing, performing sedentary activities, at or near thermal neutrality, and have no personal control over the air velocity in the zone [28]. Studies conducted by, for example, Hoyt et al. [32], have given a better understanding of how air movement can be used to better thermal comfort for occupants. Showing in the study that $52\,\%$ percent of occupants out of 6148 surveys registered in a neutral to slightly warm climate requested more air movement, while $45\,\%$ wanted no change. Indicating that draught can be a helpful tool to combat overheating of occupants. Another study conducted by Toftum et al. in 2002 [33], evaluated the subject's air movement preferences under varying overall thermal sensation and temperature and if the requested increase in air movement could be verified when more air movement was provided. This was then confirmed. In general, air movement preference depended on both overall thermal sensation and temperature, and considerable inter-individual differences existed between subjects. Indicating that subjects show a more considerable acceptance towards draught when subjected to a high thermal sensation/temperatures.

NS-EN ISO 7730 presents Equation (2.5) shown below, which can be used to predict the percentage of occupants expected to express discomfort due to draught.

$$\mathrm{DR} = (34 - \mathrm{t_{a,l}})(\overline{\mathrm{v}}_{\mathrm{a,l}} - 0{,}05)^{0{,}62}\,(0{,}37 \cdot \overline{\mathrm{v}}_{\mathrm{a,l}} \cdot \mathrm{T_u} + 3{,}14) \tag{2.5}$$

Where DR is the draught rate in percentage, $\mathrm{t_{a,l}}$ is the local air temperature in $^\circ\mathrm{C}$, $\overline{\mathrm{v}}_{\mathrm{a,l}}$ is the local mean air velocity in $\mathrm{m/s}$ and $\mathrm{T_u}$ being the local turbulence intensity, standard deviation between the local air velocity and the mean local air velocity (if unknown, $40\,\%$ may be used). The model applies to light, mainly sedentary activity level and a close to neutral thermal sensation. The formula calculates predictions of draught at neck height. Calculations at a lower level of the body may result in an overestimation of the draught rate. NS-EN ISO 7730 also discusses how an increased air velocity can offset the warmth sensation caused by increased temperatures. Showing that an increase in air velocity from $0.2\,\mathrm{m/s}$ to $0.82\,\mathrm{m/s}$ can offset the increased warmth sensation created from a temperature increase of $3.0\,^\circ\mathrm{C}$ above $26\,^\circ\mathrm{C}$ for light, primarily sedentary activity.

2.2. Atmospheric Environment

When evaluating indoor climate and its effects on occupants, thermal parameters and air quality should be assessed. Air quality has an extensive impact on the well-being and health of humans and is a deciding factor when assessing the quality of an indoor environment. When assessing the atmospheric environment, both indoor and outdoor pollutants must be taken into account. This section will present the different pollutants that must be taken into consideration and how to evaluate a zone's air exchange rate.

2.2.1. Carbon Dioxide Concentration

Carbon dioxide is a taste- and colorless gas created through the combustion of organic materials and can be found in exhaled air. CO_2, even in large quantities, is not a poisonous gas but is an indicator of how much air in a room has been used through breathing. With rising CO_2 levels, it is expected that the scent of bodily odors will increase in parallel. This smell may be perceived as uncomfortable and irritating. There may be an increase in air temperature, which will affect the thermal comfort of the occupants. CO_2 levels can therefore be used to estimate/calculate the air exchange rate of the ventilation system, this will be explained in Section 2.2.3. [34]

The expected background level for CO_2 concentration is 400 - 450 ppm. NS-EN 15251 and NS-EN 13779 states that the CO_2 concentration for a given zone with maximum occupancy load should not exceed 900 - 950 ppm. [26, 35]

2.2.2. Outdoor Pollutants

When evaluating IAQ of a building, it is important to assess the outdoor conditions. Outdoor pollutants can affect indoor conditions to a large degree when brought into the building through ventilation. Outdoor air often contains concerning levels of pollutants that have a negative biological effect on humans. Small particles called particulates (PM_{10} or $PM_{2,5}$) are comprise of a wide range of solid or liquid materials that are found in the air. The toxicity of these particulates depends on their size and chemical composition. Where smaller particulates are generally considered more dangerous as they more easily penetrate the lungs. Though there have been studies debating which particulate size has the largest effect on human health [36]. Levels of ozone, carbon monoxide, sulfur oxides, and nitrogen oxide molecules must also be assessed. However, PM and NO_x are the most important pollutants in Norwegian urban centers as the concentration levels for most other components are below the EU limit values. [37, 38]

When extracting outdoor air for ventilation of a building, it is important to assess the air quality in the extraction area. Filters and other ventilation system equipment must be used if the quality does not comply with regulations. Depending on the pollution levels measured, the area will be evaluated as green, yellow, or red. Table 2.7 presents the respected pollution levels for the different zones.

Table 2.7.: *Recommended limits for air pollution and zoning of pollution degree [39]*

Component	**Pollution zones**		
	Green zone	Yellow zone	Red zone
PM_{10} - 7 days per year	< 35 $\mu g/m^3$	35 $\mu g/m^3$	50 $\mu g/m^3$
NO_2 - winter mean	< 40 $\mu g/m^3$	40 $\mu g/m^3$	40 $\mu g/m^3$

Yellow zone: People with severe respiratory and cardiovascular disease have an increased risk of worsening of the disease. Healthy people will, in most cases, not be affected by the pollutants.
Red zone: People with respiratory and cardiovascular disease have an increased risk of negative health effects. Among these, children with respiratory disorders and the elderly with respiratory and cardiovascular disorders are most vulnerable. [39]

Placement of air intake and exhaust must be carefully considered to avoid unnecessary pollutants being brought back into the building. Intake should be placed on a facade or roof with access to the best air quality. Avoid placement facing high pollution areas like parking lots, garages, and smoking areas. The exhaust air should be extracted as far away from the intake as possible to avoid air being brought back into the building via the air intake. Exhaust air should be exhausted away from potentially occupied areas. [14, 40]

2.2.3. Air Exchange Rate

To avoid the accumulation of pollutants in a building, it is important to have a sufficient air exchange rate. The air exchange rate is defined as the airflow rate Q, passing through a zone divided by the volume V, of the zone. To calculate the air exchange rate n, Equation (2.6) can be used. It expresses the period it takes for all air in a room to be replaced.

$$\mathrm{n} = \frac{\mathrm{Q}}{\mathrm{V}} \tag{2.6}$$

TEK 17 states that the fresh air supply due to pollution from occupants in light activity should be a minimum of $26\,\mathrm{m^3/hour\ per\ person}$. At higher activity levels, the supplied air must be adjusted to achieve sufficient air quality. Fresh air supply due to pollution from materials, products, and installations should be a minimum of $2.5\,\mathrm{m^3/h \cdot m^2}$ floor area when the utility unit or rooms are in use and $0.7\,\mathrm{m^3/h \cdot m^2}$ floor area when the utility unit or rooms are not in use. [41, 42]

2.2.4. Age of Air

Age of air is a concept that can be used to classify the efficiency of a ventilation system and is defined as the time frame a given amount of air enters a building until the same amount has left. Figure 2.2 illustrates a zone with one inlet and one outlet. The air passing through the zone will pass point *P* at a given time. This time period is referred to as the *local age of air*, τ_p. The *local mean age of air*, $\overline{\tau_\mathrm{p}}$, equals the mean local age of air of all air streamline passing through the room. The *nominal time constant*, τ_n, refers to the local mean age of air in the exhaust, which has a constant value as both supply and extract have a constant value. The *nominal time constant* can be calculated with Equation (2.7). [43]

$$\tau_\mathrm{n} = \frac{\mathrm{V}}{\mathrm{Q_V}} \tag{2.7}$$

where τ_n equals the room volume, V, divided by the ventilation rate, $\mathrm{Q_V}$. The *nominal time constant* is the shortest possible time for an air change to take place of an entire room [44].

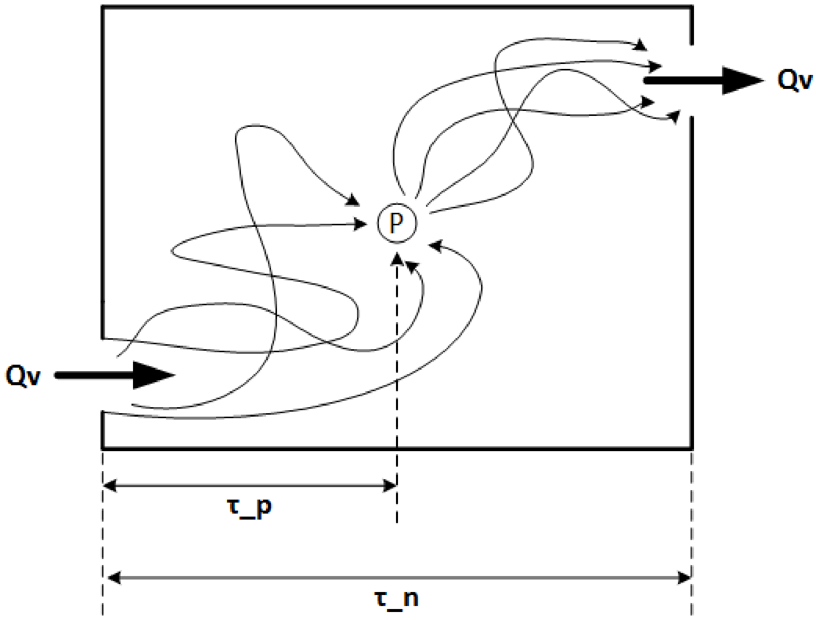


Figure 2.2.: *Age of air definition (Reproduced from Etheridge & Sandberg, 1996) [44]*

The average age of air in the exhaust is by definition equal to the *residence time of air* in the room, $\overline{\tau_{\mathrm{r}}}$ [44]. The residence time can also be expressed as half of the *mean age of air* in the room, $\langle\overline{\tau}\rangle$, as presented in Equation (2.8)

$$\overline{\tau_{\mathrm{r}}} = 2\langle\overline{\tau}\rangle \tag{2.8}$$

Air Change Efficiency

The air change of an entire room is an important indicator of the experienced air freshness [44]. The *air change efficiency*, $\langle\varepsilon_{\mathrm{a}}\rangle$, can be determined by calculating the ratio between the shortest possible time of air change, τ_{n}, and the actual time of air change, τ_{r}. Equation (2.9) presented below shows this connection.

$$\varepsilon_{\mathrm{a}} = \frac{\tau_{\mathrm{n}}}{\tau_{\mathrm{r}}} = \frac{\tau_{\mathrm{n}}}{2\langle\overline{\tau}\rangle} \tag{2.9}$$

The value of the air exchange efficiency can also be used to determine the characteristic of the airflow, and this is presented in Table 2.8. Unidirectional flow pattern, which may occur, for example, in displacement ventilation, achieves an air exchange efficiency of 0,5-1. The age of air registered in the exhaust is higher than that of the average age of air in the zone, suggesting an accumulation of contaminants in the upper layer of the zone. Perfect mixing occurs during mixing ventilation, where the concentration of the contaminants is equal throughout the zone. The *residence time of air* is equal to half of the *nominal time constant* and the air change efficiency is therefore 0,5. Short circuit characteristic achieves an air exchange efficiency of 0-0,5. The age of air passing through the exhaust is lower than that of the average age of air in the zone, suggesting that supplied air is not mixing in the zone. [45]

Table 2.8.: *Air exchange efficiency for characteristic room ventilation flow [45]*

Flow pattern	**Air exchange efficiency,** ϵ_a	**Compared with time of exchange**
Unidirectional flow	0.5 - 1.0	$\tau_n < \tau_r < 2\ \tau_n$
Perfect mixing	0.5	$\tau_r = 2\ \tau_n$
Short circuiting	0 - 0.5	$\tau_r > 2\ \tau_n$

Containment Removal Effectiveness

The efficiency of ventilation can also be indicated by the *contaminant removal effectiveness*. This is done by calculating the ratio between the *contaminant level in the exhaust*, $\mathrm{c_e}$, the *mean contaminant concentration* of the zone, $\langle \mathrm{c} \rangle$. Equation (2.10) presents the connection.

$$\epsilon^{\mathrm{c}} = \frac{\mathrm{c_e}}{\langle \mathrm{c} \rangle} \tag{2.10}$$

2.3. Results of Poor Indoor Environment

A high-quality indoor environment is important for the concentration, health, and general well-being of occupants. Irritations, feeling of discomfort, more or less serious health problems, or a reduction in work efficiency may occur if the indoor environment is not within the specified standards [23]. Literature suggests that buildings of high standard designs do not automatically guarantee the occupant's satisfaction of the indoor environment [46]. As mention though Section 2.1 and 2.2 there are many factors that affect the indoor environment. When an occupant is experiencing one or more symptoms indicating an indoor environment with low quality, it can be hard to pinpoint specifically what the cause may be. Disturbances in the thermal and atmospheric environment are most relevant for the subject of this report as the factors of these environments affect thermal comfort and air quality and are directly affected by ventilation. When assessing an indoor environment, it is important to understand the effects of the acoustic and actinic environment on human comfort as well. A general description and their potential effects on dissatisfaction will be presented continuing this chapter. [14, 41]

Some symptoms that may occur for occupants who experience a poor indoor environment are mucosal irritations, headaches, general feelings of fatigue, dizziness, concentration difficulties, and skin and eye irritations. When occupants regularly experience symptoms, then the building may be classified as a "Sick building". [14, 41]

2.3.1. Acoustic Environment

The acoustic environment for an occupant of an area is the sound from all sources that that occupant in that area can hear. Undesirable sound experienced by an occupant can be considered as noise. Poor acoustic properties of a room may reduce occupant's speech intelligibility, make them irritated, unable to concentrate, which may affect productivity, and hearing problems/permanent hearing damage, if exposed to a sufficient and prolonged load of noise [47, 48, 49]. Low-frequency noises from air supply fans are often a dominant factor of noise complaints [50]. Some studies have reported occupants turning down their ventilation systems to tolerable levels as a response to the unwanted noise or disable them entirely to prevent the perceived noise nuisance [51]. This negatively affects the adequate IAQ as modern airtight depends on an effective ventilation system, which will cause further problems as the air quality deteriorates and pollutants accumulate. A study on facade sound insulation of Italian schools show the facades of building passively works as sound insulation [52]. An increase in insulation can have positive effects on the perceived acoustic environment regarding the penetration of sound from outside the building. When implementing natural ventilation, the passive sound insulation of the facade is removed as openings for airflow are used. Thus, exterior sound must be assessed when natural ventilation is implemented to avoid a reduction in Indoor Environmental Quality (IEQ).

2.3.2. Actinic Environment

The actinic environment regards mainly lighting and illumines of an occupants surroundings. A study on the influence of indoor lighting on students in Italy found that insufficient lighting had a significant impact on the students learning ability [53]. A large study conducted on 2744 occupants collected over 1 and $1/2$ years on a green building (buildings with environmentally responsible and resource-efficient structures and application of process throughout a buildings life-cycle) in Korea showed that use of daylighting and screen-type shading device to intercept direct sunlight and reduce annoyance glare could effectively increase visual comfort [54]. Indicating that lighting has a considerable impact on the reduction of concentration and work performance.

An exciting find was discovered in a study on the effects of thermal, luminous, and acoustic environments on indoor environmental comfort in offices [48]. The experiment was conducted on 120 university students who were placed in a room and exposed to varying temperatures, the noise level from the surroundings, and light levels. The study found that the subjects found it hard to be satisfied with the environment when the temperature was dissatisfying—indicating that temperature has one-vote veto power over the satisfaction level of the indoor environment. The same was also discovered about the noise level. The subjects found it hard to be satisfied with the indoor environment regardless of temperature and lighting level when the noise level was dissatisfying. The noise level also has one-vote veto power over the satisfaction level of the indoor environment. However, this was not the case for the lighting level. In situations with a level of light outside the acceptable range, it was still possible for the entire environment to be judged acceptable. However, it should be mentioned that lighting comfort/dissatisfaction is highly dependent on the planned activity and age, preference, and ability of the occupant. The "Illuminating Engineering Society of North America (IESNA) Lighting Handbook" states. Lighting conditions must provide appropriate lighting conditions for all tasks that are completed in the space [55].

2.3.3. Sick Building Syndrome

Sick building is a description of a building where symptoms of sickness are regularly experienced by occupants that are temporal and related to working in that particular building. SBS is comprised of a group of mucous membrane symptoms related to the eyes, nose, and throat, dry skin, together with what are often called general symptoms of headache and lethargy, and is recognized as an increased occurrence of these non-specific symptoms among populations in a determined building [56]. The symptoms can affect concentration, performance, productivity, and the health of occupants. Actions should therefore be taken to reduce the occurrence of these symptoms.

Many explanations of SBS have been put forward: insufficient fresh air supply, little user impact, insufficient maintenance, noise from the ventilation system, misplaced air intake, pollution from air filters, heat recovery, heating, and cooling batteries and duct system, and more. Individuals diagnostics has proven to be a problematic issue as it is a group phenomenon and not a *syndrome* as defined in medicine. Some researchers, therefore, want to abolish the SBS concept [57], while others accept it [58]. In-office buildings, SBS may have significant economic implications [59], as a reduction in productivity from eventual symptoms of sick buildings will have a direct effect on work achieved. Another study done in 2004 suggests that a reduction in air temperature and humidity levels can counteract the reduction in ventilation rates and may alleviate symptoms of SBS in indoor environments [24]. An assuring way to reduce SBS risk is to have a good and healthy indoor environment with high air quality. [14, 60]

3. Building Ventilation

Ventilation is a necessary measure to achieve healthy and satisfactory indoor environments in buildings. It should aim to provide a high air quality to the building without compromising the health, concentration, or productivity of its users [14]. The following chapter will present a general introduction to the different strategies of building ventilation, including its control and distribution methods.

Satisfactory ventilation can be achieved by either utilizing natural driving forces such as wind and temperature differences, mechanics in the form of fans, or a combination of the two. These methods can be categories into three strategies of building ventilation. Natural, mechanical, and hybrid ventilation, respectively.

3.1. Mechanical Ventilation

The mechanisms of mechanical ventilation involve forcing air with the use of fans to be supplied and extracted from a building. The different rooms of a building are connected to the fans through a duct system [44]. Supplied air is in most cases treated in an aggregate to achieve a specific quality, either preset or controlled by the condition of the supplied zone.

Mechanical ventilation is a very reliable solution for ventilating buildings. With electronically controlled fans, the building can reliably be supplied with the right amount of air. The implementation of filters, heating and cooling batteries, heat recovery units, and humidifiers within the aggregate assure good quality of supplied air. The possibility of heat recovery from the extracted air is one of the most prominent advantages of mechanical ventilation. Bypassing supply and exhaust air parallel to each other thermal energy transfers between the two, reducing the total energy demand of the ventilation system. This is especially efficient in colder climates where $80 - 90\%$ of the annual energy demand of ventilation of building is for conditioning air. [4, 61]

Mechanical ventilation is the highest costing strategy of the three due to its reliance on energy and the investment cost and maintenance of components. Energy demand increases with heating and cooling demand, and noise from air movement in ducts and fans may reduce the quality of the indoor environment if the system is not dimensioned correctly. Lastly, cleaning of ducts and equipment will be necessary throughout the lifetime of the ventilation system. [61]

3.1.1. Control Strategy of Ventilation System

There are three main strategies to manage a mechanical ventilation system. Constant air volume (CAV), variable air volume (VAV), and demand control ventilation (DCV).

Constant Air Volume

With *CAV ventilation*, the zone is supplied with a constant airflow rate throughout the operation time, regardless of the zones occupancy. The varying energy demand of the zone is covered by the heating and cooling of the ventilated air. CAV ventilation is operated with an *On / Off* mechanism. There is an alternation between the two modes as the zone goes in and out of use. This can be set by a timer or a sensor within the room. When the system is *On* the airflow rate is chosen after the maximum pollution load dimensioned for the zone. The *Off* mode is dimensioned after the static pollution load that is generated from the materials and inventory of the zone, as explained in Section 2.2.3. [62]

Variable Air Volume

With *VAV ventilation*, the air is supplied to a zone with varying airflow rate-regulated after the demand of the zone. The supply temperature is kept constant while regulation of the airflow rate covers the heating/cooling demand. VAV ventilation operates after a set of variables and regulates after changes in these variables. Sensors that measure temperature or CO_2 levels within the zone are often used. Occupancy control is another alternative that can be monitored using motion detectors.

VAV is an effective, easy way to control ventilation based on demand. The potential of over/under ventilating is largely decreased compared to CAV. Energy demand can therefore be reduced. However, the mechanism of VAV ventilation does not allow for direct feedback to the system regarding the current state of the zone. As a result, one can not be certain if the system supplies the correct amount of air to the zone [62]. [63]

Demand Control Ventilation

DCV adapts to the conditions of the zone it is regulating. The systems supply air to a zone with variable airflow rates and temperatures and automatically regulates these variables after occupancy demand. The system utilizes a feedback loop that constantly compares the achieved air quality to the desired quality and adjusts the output. While facing this varying demand in air flow rate, it is intended to deliver adequate ventilation with minimized/optimized energy use for thermal conditioning of outside air. To achieve this, a control signal proportional to the number of occupants in the zone is required. The most widely used strategy is to measure the rise in CO_2 in the indoor air relative to the outdoor air. Assuming no additional pollution source is present in the occupied area, CO_2 generation is closely proportional to the number of occupants. [62, 63, 64]

Although CO_2 based DCV has been the primary method of demand airflow control, it is not without flaws. There is a delay inherent in the sensor. The CO_2 level must rise above a set *noise level* before the sensor signals to change the airflow rate. This is to avoid continuous fluctuation in the delivered airflow rate caused by small variations and irregularities in the registered CO_2 concentration. Additionally, depending on the air exchange rate, the rise and decay of the CO_2 concentration may vary from minutes to hours. This time delay may cause zones to be over/under-ventilated in periods of occupancy change. Lastly, using CO_2 as an indicator for the state of the indoor environment does not eliminate the possible existence of other contaminates.

In the case of no occupancy and other CO_2 creating sources and therefore, by design, minimum airflow rate, the CO_2 level should be equal to that of the outside air. In a situation like this, there is no certainty that other potential pollutants are not accumulating. [64, 65]

A report by Damiano and Dougan discusses the inaccuracies in ventilation rate control using CO_2 and points out the risk of non-compliance that the method gives [66]. A proposed alternative is an external airflow measurement system to ensure the minimum levels of outside air, as required by standards, under the different load conditions is being delivered, coupled with a people counting device. Though the report states that such systems are not widely available, more recent studies on people counting technology presents promising system solutions [67, 68].

3.1.2. Air Distribution

The characterization of how air is introduced, flows through, and extracted from a zone is called air distribution. There are generally two classifications of air distribution for Heating, Ventilation, and Air Condition (HVAC) airflow: Displacement and mixing ventilation. These two strategies operate after different mechanics and, when implemented correctly, can increase ventilation efficiency.

Displacement Ventilation

In a displacement ventilation system, conditioned air is supplied directly to the zone from a diffuser placed on the low part of the zone. The air is supplied with a temperature lower than the air temperature of the zone and will therefore distribute evenly across the floor of the zone. Sources of heat such as people, electrical equipment, and other surfaces that emit heat to their surroundings create plumes that force air upwards and away from the occupied zone. Heat and contaminants accumulate at the upper part of the room and are then removed with an exhaust duct. The air exhausted from the zone is generally several degrees above the temperature in the occupied zone.

Thermal plumes are free convection that is created by forces acting upwards, driven by buoyancy. The density of air is reduced when it is heated, and hot air will therefore rise above colder air, creating a jet-like flow above heated sources. The plume will gradually spread outwards as the air climbs higher while the air velocity decreases. The flux from the diffuser is low and has a small impact on the system, and free convection is thus seen as the main force that affects flow in displacement ventilation.

Mixing Ventilation

Mixing ventilation is achieved when air is supplied at a high impulse through diffusers placed close to the ceiling. The clean air is supplied at a high initial velocity and generates a jet flow in front of the opening. It activates air in the rest of the zone and promotes a mixing effect of the supplied and existing air in the zone. The contaminated air mixes with the newly supplied air and dilutes. It is essential to supply a sufficient amount of air to thin out the contaminants in the zone to an acceptable amount. Mixing ventilation is mainly used for cooling as hot air supplied at the top layer of a zone will create a stratification layer, possibly causing a short circuit of the ventilation system. [14]

3.2. Natural Ventilation

Natural ventilation is a ventilation strategy that is reliant on naturally created pressure differences between areas. Depending on which forces are in effect or in dominance, natural ventilation will either be wind-driven, caused by pressure difference occurring on each side of a building, or buoyancy-driven, caused by temperature difference within a zone or between zones. These forces may also work together if the right conditions are met. In contrast to mechanical ventilation, the air is not brought through ducts in a ventilation system through openings on the facades like windows, doors, valves, and leaks. [44, 69]

When implemented, natural ventilation can help reduce the energy use for cooling, reduce operating costs and carbon dioxide emissions of a building. Natural ventilation is only reliant on naturally created forces, and the operation cost is therefore, close to zero. Assuming no electrical mechanisms are used for the operation of, for example, windows and doors. It is generally considered that natural ventilation provides the advantage of "contributing to a sustainable building environment" [44]. Lastly, natural ventilation gives more control to occupants, which, as presented in Section 2.1.4, significantly affects how occupants perceive their thermal environment. [61]

Natural ventilation is heavily dependent on external conditions. The ambient air must have a distinct quality to be acceptable for ventilation as there are no filtration or air conditioning components. Therefore it is not certain that natural ventilation is adaptable to every building. In addition, the efficiency of natural ventilation varies with the uncertainties of the outside conditions, which make control and utilization of natural ventilation difficult. As a result, natural ventilation depended buildings may at times be under-ventilate resulting in overheating, over-ventilate, causing draught, or ventilate with air of unacceptable quality. Air distribution may become uneven and turbulent, causing draught and discomfort for occupants. In addition, the need for openings on the facade makes natural ventilation strategies exposed to noise from the ambient surroundings as explained in Section 2.3.1. A well designed natural ventilation system is therefore essential in achieving a acceptable indoor environment. [61]

3.2.1. Wind Driven Ventilation

Natural ventilation is influenced by several environmental conditions, the most unpredictable being wind velocity, both its speed and direction. Wind creates pressure differences around a building depending on the shape, wind direction, and presence of other objects around that building that may disturb or affect the wind. Generally speaking, the pressure created is higher on the windward side of the building and lower on the leeward side and roof of the building.

Studies on wind-driven ventilation done in wind tunnel experiments by Chia-Ren Chu et al. have determined that both the internal porosity and the length of the building contributes to a loss factor through the building that affects the total airflow rate [70, 71]. Obstacles like closed or partially closed doors and the building length as a ratio to the ceiling height decrease the ventilation rate due to increased internal friction. The first report investigates the rule of thumb for adequate wind-driven cross ventilation, which suggests that the building length L should be less than five times the ceiling height H [70]. It was discovered that buildings with aspect ratio $L/H \geqslant 5$ would overestimate ventilation rates by up to 20% more than models that did not consider the internal resistances. By investigating the placement of openings on wind and leeward facades, it was discovered that openings on opposite corners rather than on the centerline would further reduce the ventilation rate. The second report presents a resistance model that predicts the ventilation rate of wind-driven cross ventilation in buildings with internal porosity [71]. Though the model for simulation of this master thesis will use open floor zones, this should be factored in when assessing the results.

Figure 3.1 below illustrates how a general pressure profile of wind-driven ventilation changes throughout a single zoned building. Both indoor and outdoor temperatures are assumed uniform, and as a result, pressure does not vary with height. The generated pressure is shown to be at its highest on the windward side, lowest on the leeward side of the building, and the pressure gradually decreases through the building, as expected from the theory presented in Section **??**. [72]

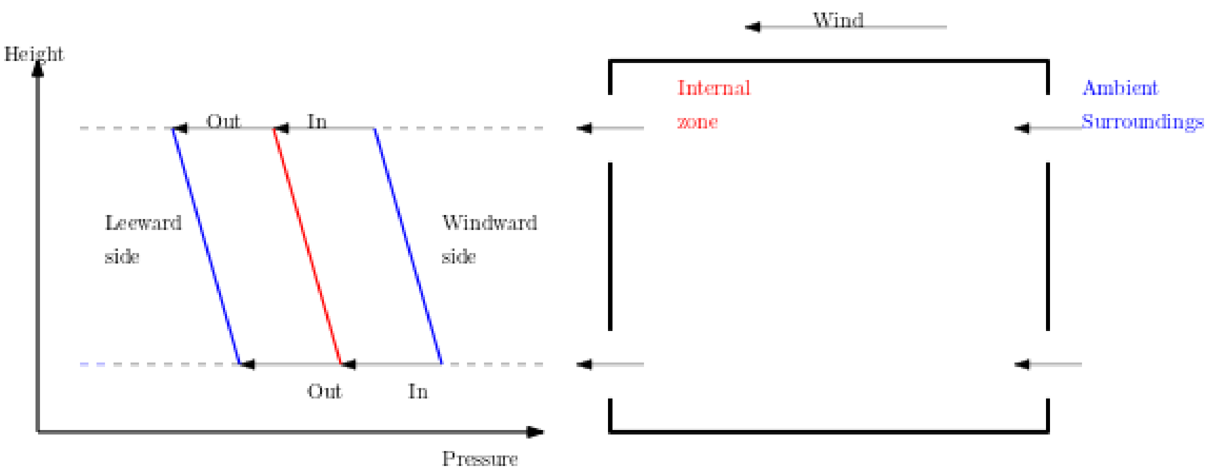


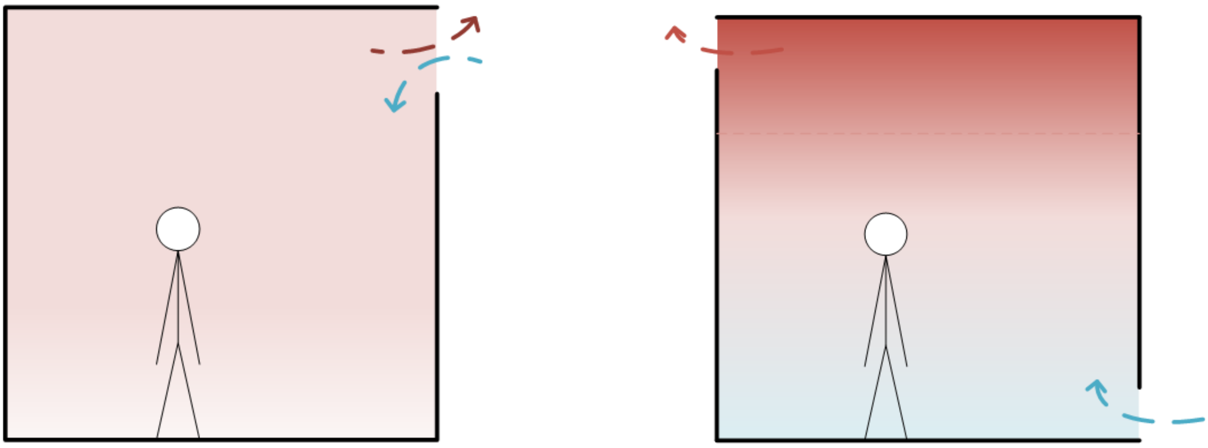
Figure 3.1.: *Resulting pressure profile through a building due to wind [72]*

3.2.2. Buoyancy Driven Ventilation

Temperature differences between different areas create buoyancy forces that drive flow. While wind-driven ventilation is driven by forces acting outside the building, stack-driven ventilation results from forces purely within the building. Buoyancy forces lead to temperature variations within the zone which creates different flow patterns depending on the placement of the fresh air inlet. Hot air naturally rises and creates stable stratification throughout the room. This is used to determine the best position for the inlet and outlet of the system.

If the air in a zone has a higher temperature than the ambient environment, then a single opening at the top layer of the room will allow an exchange of warm air outwards and cool air inwards, as shown in Figure 3.2a. As the cool air enters the zone, it quickly descends like a turbulent plume, activating and mixing with the zone. Mixing ventilation, as explained in Section 3.1.2 is then achieved. If the single opening were placed close to the floor under the same conditions, incoming air would gradually occupy the depths up to the height of the opening. Pressure between the ambient and interior at this point would become equal, and air exchange would eventually cease.

If two openings are used, then it would be most efficient to place one close to the floor and the other at ceiling level, shown in Figure 3.2b. Cool air enters through the bottom opening and flows through the building as it is being heated and out at the top layer of the zone. Displacement ventilation, as explained in Section 3.1.2 is then achieved. [11]



(a) *Buoyancy driven ventilation with use of one opening*

(b) *Buoyancy driven ventilation with use of two opening*

Figure 3.2.: *Buoyancy driven ventilation strategies*

The pressure difference occurring in a zone due to density, static pressure, and elevation difference can be depicted graphically by a pressure profile [73]. Figure 3.3 below illustrates a general pressure profile of both internal and external pressure gradient. It is assumed that the internal temperature is higher than the external temperature.

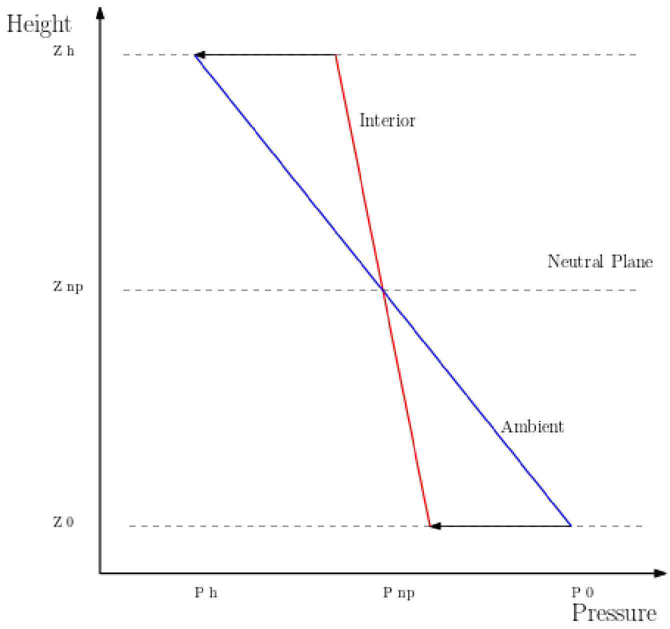
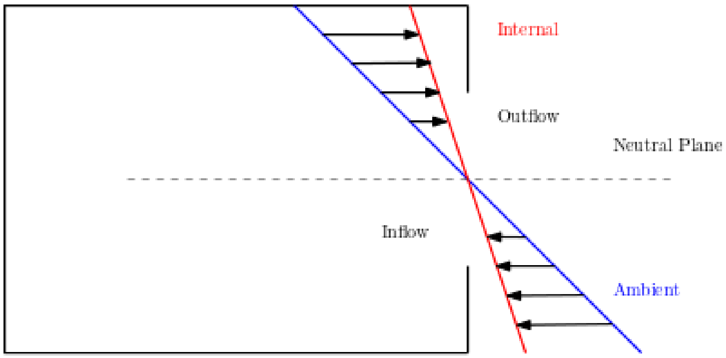


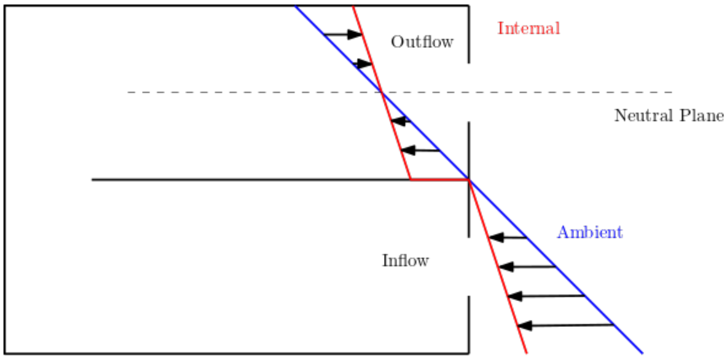
Figure 3.3.: *Example of a pressure profile created by temperature difference between the upper and lower layer of a zone*

On the lower level of the zone, the ambient pressure is higher than the internal pressure. The pressure difference forces outside air into the building. The internal pressure increases gradually per unit of height as the temperature increases. Following the y-axes, the profile shows interior pressure gradient surpasses the ambient pressure gradient at the height Z_{np}. This point is called the neutral plane and is where the ambient and interior pressure is equal. The neutral plane defines the height that separates the lower and upper openings on a facade: air flows in through openings below the neutral plane and out through openings place above. At the exact level of the neutral plane, no air exchange is achieved between the interior and exterior of the building as no forces are acting on them. [74]

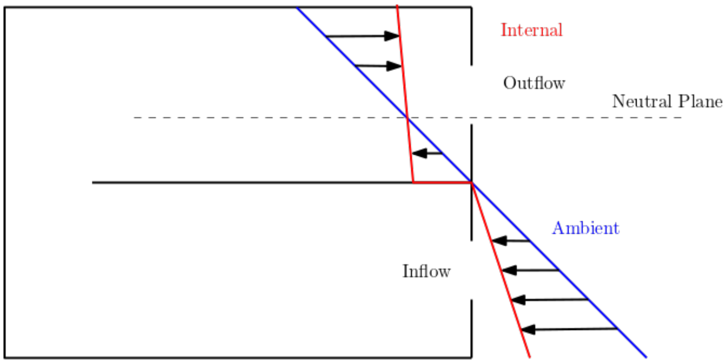
It is interesting to evaluate how pressure profiles area affected by changes to interior properties of a building, such as the presence of dividing floors, zonal division, and temperature differences across floors. The figures presented in Figure 3.4 below illustrate different cases where these factors are implemented coupled with the resulting pressure profiles. These figures are based on a research paper on high-rise residential buildings in Korea [75].



(a) *Pressure profile over a opening*



(b) *Pressure profile across dividing floors*



(c) *Pressure profile across dividing floors with higher top floor interior temperature*

Figure 3.4.: *Resulting pressure profiles across a facade with different internal geometric. Figures inspired by earlier papers [11, 73, 75]*

Figure 3.4a shows the resulting pressure profile of an open zone building. The resulting gradient is quite similar to the general pressure profile depicted in Figure 3.3. In Figure 3.4b the pressure profile has been affected by the addition of a second floor with a staircase connecting them. A considerable pressure drop occurs over the floor, and the interior pressure profile will shift, resulting in an elevation of the neutral plane. Figure 3.4c shows the same situation as b) but with a higher interior temperature on the second floor. The pressure profiles are pretty similar, but the internal profile is steeper, resulting in a neutral plane that occurs lower on the facade.

3.2.3. Wind and Buoyancy Driven Ventilation

Wind and buoyancy-driven ventilation occur when both wind and buoyancy-driven forces are acting on a zone. These forces can either reinforce one another or be in opposition depending on the ambient parameters and the design of the zone. The resulting pressure difference created by a combination of both wind and temperature difference equals the sum or difference of the pressure caused by wind, ΔP_w, and buoyancy, ΔP_b. The total pressure difference, ΔP_{total}, is presented in Equation (3.1). [69]

$$\Delta P_{total} = \Delta P_b \pm \Delta P_w \tag{3.1}$$

Research has shown that there is a Pythagorean relationship between the combined buoyancy and wind-driven velocity and the velocities of buoyancy and wind forces acting in isolation [76]. This can also be proven through calculations using the equations that will be presented in the following section.

Figure 3.5 below illustrates how a general pressure profile of wind and buoyancy ventilation changes throughout a single zoned building. The pressure profile is quite similar to the pressure profile through a building due to only wind, though the pressure level changes with height as well as through the building. It is important to note that the pressure profile created by a combination of wind and buoyancy-driven forces will vary depending on if the forces are reinforcing each other or stand in opposition.

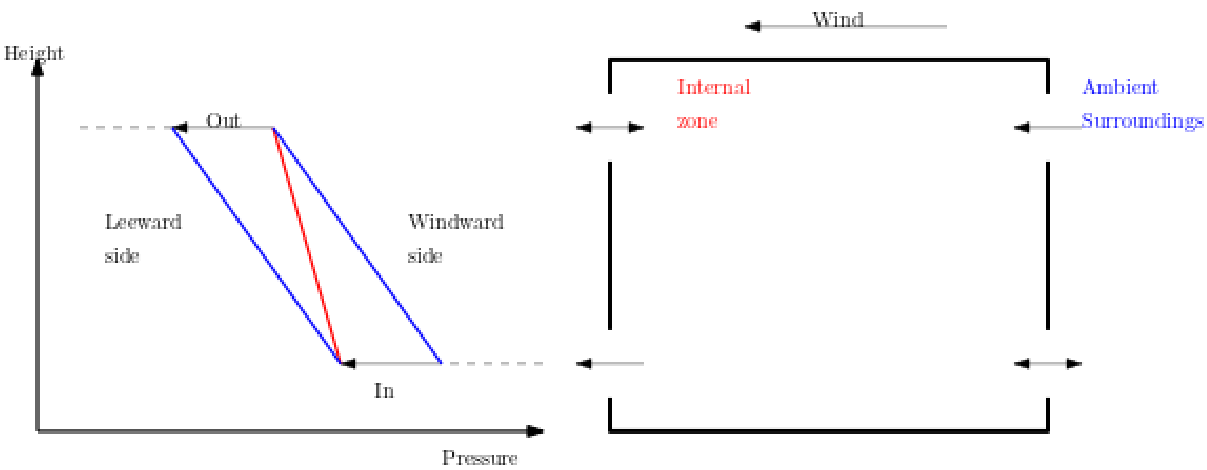


Figure 3.5.: *Resulting pressure profile through a building due to wind and buoyancy driven forces [72]*

3.3. Mathematics of Natural Ventilation

As mention in Section 3.2, one of the disadvantages of natural ventilation is its reliance on naturally generated forces. Though it is hard to control, it is crucial to utilize the freely available resources wind, and buoyancy provide to reduce the total energy cost and the $\mathrm{CO_2}$ footprint cause by building operations. To develop mechanisms for control that can provide a high-quality indoor environment, it is essential to understand the fundamental physics and mechanics of natural ventilation.

3.3.1. Fluid Mechanics

When predicting airflow through a building, it is essential to understand the fundamentals of fluid mechanics. The following section will present the fundamental physics that will help understand the mechanics of natural ventilation.

Conservation of Mass

Mass conservation within a zone of steady-state conditions must be ensured. The mass conservation Equation (3.2) expresses that the mass flow rate entering a zone must be equal to the mass flow rate exiting the same zone, resulting in the total amount of air in the zone remaining constant.

$$\sum_{\mathrm{k=1}}^{\mathrm{N_k}} \mathrm{m_k} = \sum_{\mathrm{k=1}}^{\mathrm{N_k}} \rho \cdot \mathrm{Q_k} = 0 \tag{3.2}$$

Where $\mathrm{N_k}$ is the total distinct leakages of air in and out of the zone and $\mathrm{m_k}$ is the individual mass flow rate through leakage opening k. In addition, Equation (3.2) illustrates that mass flow rate $\mathrm{m_k}$ can be expressed as the density of air, ρ, multiplied by the airflow rate, $\mathrm{Q_k}$. [69]

Archimedes Number

Archimedes number is a dimensionless number that describes the relationship between the inertia forces and the gravity forces and can be used to determine the motion of fluids due to density differences. The Archimedes number can be calculated using Equation (3.3). [63, 77]

$$Ar_o = \frac{g \cdot l \cdot |\Delta T_o|}{T_i \cdot v^2} \tag{3.3}$$

Where g is the acceleration of gravity, l is the characteristic length of the supply air device, $\Delta\mathrm{T_o}$ is the temperature difference between the supply air and the room air, $\mathrm{T_i}$ is the temperature of the indoor air and v is the air velocity based on net opening area.

The Continuity Equation

The continuity equation presented in Equation (3.4) expresses how mass flow between two points will remain constant for steady-state flow.

$$\dot{m} = \rho_1 \cdot A_1 \cdot U_1 = \rho_2 \cdot A_2 \cdot U_2 \tag{3.4}$$

In most cases, the pressure and temperature difference are small enough that the density of both points can be assumed equal. The flow rate Q can then be expressed as shown in Equation (3.5). [14, 78]

$$Q = A_1 \cdot U_1 = A_2 \cdot U_2 \tag{3.5}$$

A_1 and A_2 are the openings of the different point and U_1 and U_2 are the velocity of a fluid of gas at these points.

Bernoulli's Equation

Bernoulli's equation shows the relation between a fluid or gas pressure and velocity traveling through a pipe or an opening.

$$P_{s1} + \frac{\rho_1\, {U_1}^2}{2} + \rho_1 \cdot g \cdot z_1 = P_{s2} + \frac{\rho_2\, {U_2}^2}{2} + \rho_2 \cdot g \cdot z_2 + \Delta P_{1-2} \tag{3.6}$$

Where P_{s1} and P_{s2} are the static pressure measured in ratio to the atmospheric pressure and $(\rho \cdot U_{1,2}^2)/2$ are the dynamic pressure at their respected points. ΔP_{1-2} is the pressure difference between the two points caused by friction. [14, 78]

The equation shows that an increase in the fluid velocity from points 1 to 2 must decrease either pressure or potential energy of an inviscid fluid. The shows that the energy of the fluid is constant [79].

Reynolds Number

A fluid or gas flow can be defined as either a laminar, turbulent, or combination. A laminar flow is characterized by parallel flow lines, which generally occur with low velocities. When the flow lines start to move in the axial and radial direction, the flow can be turbulent, with high velocities. The value of Reynolds number, presented in Equation (3.7), indicates whether a flow is laminar or turbulent. [78]

$$Re = \frac{U \cdot D_h}{\nu} \tag{3.7}$$

Where U is the velocity of the flow, D_h is the internal diameter of the pipe, and ν is the kinematic viscosity of the flow. The flow is determined laminar if the value of the Reynolds number, Re, for that flow is less than 2000 and classified as a turbulent flow if the value exceeds 3500. At Reynolds numbers between about 2000 to 4000, the flow is unstable due to the onset of turbulence. This flow characteristic is referred to as a transitional flow. [14, 78]

3.3.2. Mathematics of Wind-Driven Ventilation

The pressure difference between the two sides of the building creates forces that drive flow within the building. The pressure difference over the building can be calculated with Equation (3.8).

$$\Delta P = \frac{1}{2} \cdot \rho \cdot \Delta C_p \cdot U_w^2 \tag{3.8}$$

Where ρ is the density of the air, ΔC_p is the difference in pressure coefficient between the relevant facades, and U_w is the air velocity at the reference height due to wind. The reference height is normally equal to the height of the building. The pressure coefficient C_p is a dimensionless value that describes the relative pressure throughout a flow. Every point within a fluid flow and on a facade has a distinct pressure coefficient. C_p values are in most cases in studies obtained through the use of wind tunnel experiments [80]. The pressure coefficient C_p is defined as the wind pressure at a given point on a facade divided by the dynamic pressure in the free wind. Equation (3.9) presents the relation. [81]

$$C_p = \frac{P_x - P_0}{P_d} \tag{3.9}$$

Opening on a building facade can be defined into two categories, openings that are larger or smaller than $10\,\mathrm{mm}$. The power-law equation can express the airflow rate across an opening due to pressure differences (3.10) presented below.

$$Q = K \cdot (\Delta P)^n \tag{3.10}$$

Where K is the flow coefficient that is a function of the geometry of the opening and n is the flow exponent that depends on the flow characteristics and varies in the range 0,5 to 1,0. 0,5 corresponding to a fully turbulent flow and 1.0 corresponding to a laminar flow. Equation (3.10) can be used for both openings smaller and larger than 10 mm. Only large openings will be analyzed in this report. To calculate the airflow rate through a large opening the continuity and Bernoulli's and Equations, (3.4) and (3.6), are used to derive Equation (3.11). A_2 is assumed small compared to A_1 and therefore $(1 - (\frac{A_2}{A_1}))^2$ is assumed to equal 1. Equation (3.11) can then be derived.

$$U = \sqrt{\frac{2(P_1 - P_2)}{\rho \cdot (1 - (\frac{A_2}{A_1}))^2}} \approx \sqrt{\frac{2\Delta P}{\rho}} \rightarrow Q = A \cdot \sqrt{\frac{2\Delta P}{\rho}} \tag{3.11}$$

Equation (3.11) describes an ideal situation where the effects of viscosity are neglected. The discharge coefficient $\mathrm{C_d}$ is introduced to account for *real world* effects, such as turbulent motions and swirling flows. The discharge coefficient is a function of the temperature difference, wind velocity, and opening geometry [69]. The discharge coefficient is equal to the ratio between the measured and theoretically calculated airflow. An estimation of the actual discharge coefficient can be calculated with Equation (3.12).

$$\mathrm{C_d} = (0\,4 + 0{,}0075 \cdot \Delta \mathrm{T}) \tag{3.12}$$

By adding the discharge coefficient to Equation (3.11) the airflow rate for *real world* scenarios can be calculated. [72, 74]

$$\mathrm{Q} = \mathrm{C_d} \cdot \mathrm{A} \sqrt{\frac{2\Delta \mathrm{P}}{\rho}} \tag{3.13}$$

Equation (3.8) is inserted for $\Delta \mathrm{P}$ in Equation (3.13), Equation (3.14) can then be used to calculate the air flow rate due to wind.

$$Q = C_d \cdot A \cdot U_w \sqrt{\Delta C_p} \tag{3.14}$$

3.3.3. Mathematics of Buoyancy Driven Ventilation

Buoyancy-driven ventilation is based on the simple fact that the density of air changes with temperature. Air can be assumed to be a perfect gas, and the ideal gas law presented below in Equation (3.15) can be used to calculate the density of air at a given temperature. As explained in Section 3.1.2, heated air will rise above cooler air due to the density air reducing with an increase in temperature. [69]

$$\rho_1 = \rho_0 \cdot \frac{\mathrm{T_0}}{\mathrm{T_1}} \tag{3.15}$$

Where ρ_0 and $\mathrm{T_0}$ is the reference density and temperature of the air, and $\mathrm{T_1}$ is the absolute temperature of the air that the density is being calculated. The differences in density of air in a zone create pressure differences. The pressure $\mathrm{P_1}$ at a given height $\mathrm{z_1}$ within a zone can be derived from Equation (3.16).

$$\mathrm{P_1} = \mathrm{P_0} - \rho_1 \cdot \mathrm{g} \cdot \mathrm{z_1} \tag{3.16}$$

Where $\mathrm{P_0}$ refers to the static pressure at the bottom of the zone. To calculate the pressure difference between two distinct points in a zone at different heights due to buoyancy, Equation (3.17) can be used.

$$\Delta \mathrm{P} = \mathrm{P}_{1,0} - \mathrm{P}_{2,0} + (\rho_1 - \rho_2) \cdot \mathrm{g} \cdot (\mathrm{z}_1 - \mathrm{z}_2) \tag{3.17}$$

$\mathrm{P}_{1,0}$ and $\mathrm{P}_{2,0}$ is referring to the static pressure at the different points in reference of the reference point P_0. Assuming the static pressure is constant in the zone, the equation for air flow rate due to buoyancy can be derived by inserting Equation (3.17) for $\Delta \mathrm{P}$ in Equation (3.11).

$$Q = C_d \cdot A \sqrt{\frac{2 \cdot (\rho_1 - \rho_2) \cdot g \cdot (z_1 - z_2)}{\rho_2}} \tag{3.18}$$

3.3.4. Mathematics of Wind and Buoyancy Driven Flow

To calculate the total pressure difference between openings caused by both buoyancy and wind pressure, Equation (3.8) and (3.17) are combined, and Equation (3.19) can be used.

$$\Delta \mathrm{P} = \mathrm{P}_{1,0} - \mathrm{P}_{2,0} + \frac{\rho_1 \cdot \mathrm{C}_{\mathrm{p1}} \cdot \mathrm{U}_1^2}{2} - \frac{\rho_2 \cdot \mathrm{C}_{\mathrm{p2}} \cdot \mathrm{U}_2^2}{2} + (\rho_1 - \rho_2) \cdot \mathrm{g} \cdot \mathrm{z} \tag{3.19}$$

When calculating the pressure difference between two points where one of the zones is defined outside the building then $\mathrm{U}_2 = 0$, U_1 equals the wind velocity at the reference height of the building and ρ_I, and ρ_E is the density of the internal and external air of the building. The resulting pressure difference, $\Delta \mathrm{P}_{\mathrm{ext}}$, can be calculated using Equation (3.20).

$$\Delta P_{ext} = P_0 - \frac{\rho_I \cdot C_p \cdot U_{ref}^2}{2} + (\rho_I - \rho_E) \cdot g \cdot z \tag{3.20}$$

If both points of interest are within the same zone, then the velocity difference can be neglected. The pressure difference would then be purely created by buoyancy, and Equation (3.17), presented in Section 3.2.2, can be used. Equation (3.8) for wind pressure calculation would be used if the temperature in the zone were uniform.

Assuming the static pressure of the zone is constant, the airflow rate created by both wind and buoyancy can be calculated using Equation (3.21). This is derived by implementing Equation (3.20) to Equation (3.11). Depending on the ambient conditions and how/where the openings to a zone are placed/designed, buoyancy and wind-driven forces will either work in combination or opposition. Equation (3.21) indicates this with a pm sign.

$$\mathrm{Q} = \mathrm{C}_\mathrm{d} \cdot \mathrm{A} \cdot \mathrm{U}_\mathrm{w} \sqrt{\Delta \mathrm{C}_\mathrm{p} \pm \frac{2 \cdot (\rho_\mathrm{I} - \rho_\mathrm{E}) \cdot \mathrm{g} \cdot (\mathrm{z}_2 - \mathrm{z}_1)}{\rho_\mathrm{I}}} \tag{3.21}$$

When mass balance is assumed, the pressure difference over one of two openings must be equal to half of the total pressure difference over the total building [44]. Thus the pressure difference when calculating the airflow rate for wind-driven, buoyancy-driven, and a combination of the two will be halved. Equations (3.22), (3.23) and (3.24) are then used.

$$Q = C_d \cdot A \cdot U_w \sqrt{\frac{\Delta C_p}{2}} \tag{3.22}$$

$$Q = C_d \cdot A \sqrt{\frac{(\rho_1 - \rho_2) \cdot g \cdot (z_1 - z_2)}{\rho_2}} \tag{3.23}$$

$$\mathrm{Q} = \mathrm{C_d} \cdot \mathrm{A} \cdot \mathrm{U_w} \sqrt{\frac{\Delta \mathrm{C_p}}{2} \pm \frac{(\rho_\mathrm{I} - \rho_\mathrm{E}) \cdot \mathrm{g} \cdot (\mathrm{z_2} - \mathrm{z_1})}{\rho_\mathrm{I}}} \tag{3.24}$$

3.4. Hybrid Ventilation

The disadvantages of both mechanical and natural ventilation have led to compromise between the two strategies. By utilizing the strengths of both principles, it is now realistic to satisfy the requirements of thermal comfort and high air quality in an energy-efficient manner. This is done by implementing mechanical and natural ventilation and utilizing both strategies at different times and seasons of the year. Hybrid ventilation is an intelligent system that can manually or automatically shift between/regulate natural and mechanical ventilation to optimize energy consumption in addition to maintaining an acceptable indoor environment and thermal conditions [82]. The disadvantage of using hybrid ventilation is the fact that two ventilation systems must be designed and installed. [12, 83]

There are three categories of hybrid ventilation that are defined in the book "Principles of Hybrid Ventilation" by Heidelberg [84]: Fan assisted natural ventilation, mechanical ventilation assisted by natural forces, and mixed-mode ventilation, which will be discussed further continuing this section.

3.4.1. Fan Assisted Natural Ventilation

This principle is based on a natural ventilation system that utilizes an extract or supply fan. Ventilation is covered by naturally created forces that, during periods where these forces prove insufficient, weak, or increase demand, can enhance the pressure difference through the building by a mechanical fan.

3.4.2. Mechanical Ventilation Assisted by Natural Forces

This principle is based on a mechanical ventilation system that makes optimal use of natural driving forces. A considerable amount of the necessary pressure can be accounted for by the natural driving forces and is often used in buildings where the mechanical ventilation system has minimal pressure losses.

3.4.3. Mixed - Mode Ventilation

With the ever-increasing focus on enhancing energy efficiency and maximizing occupant's thermal comfort, there has been a shift away from traditional mechanical cooling systems to more passive cooling strategies in the form of natural ventilation. As explained in Section 3.2, natural ventilation is dependent on ambient conditions and is therefore prone to uncertainties. Studies have shown that the sole use of natural ventilation for building ventilation leads in most cases to discomfort for occupants, especially in northern countries [85]. For this reason, Mixed-Mode Ventilation (MMV) has been extensively used in recent years. MMV is a hybrid ventilation strategy that combines natural ventilation from manual or automatically controlled windows/openings on the facade with mechanical ventilation that includes air conditioning and distribution equipment for distribution and cooling of air.

Mixed-mode ventilation can be utilized with different methods where classifications are based on whether natural and mechanical ventilation operates in different or the same space and at the same or different periods. It is essential to understand that the different methods apply to different situations and different parts of a building. Furthermore, designers must understand that a different design philosophy must be planed from the early design stage to correctly utilize the strategy and reduce the energy consumption in MMV buildings [84]. The Center for the Built Environment (CBE) at The University of California has developed a report to summarize the operational control strategies [86]. MMV does not have a "standard" implementation approach as buildings continue to be unique. Though MMV can be divided into two different classifications: Complimentary, the classification with overlapping systems of natural and mechanical cooling, and zoned, generally referred to as physical distribution of different conditioning strategies.

Concurrent Mode

Concurrent mode falls under the complimentary classification. This strategy utilizes both natural and mechanical cooling within the same zone at the same time. Figure 3.6 illustrates this concept. There are many ways concurrent mode can be controlled, all dependent on how the setpoint of the system is. Depending on the desired purpose of the system, the setpoint can be, for example, moved such that the building is primarily in passive mode and the mechanical cooling is only utilized to control the peaks. Windows can be manually controlled, giving occupants more control over their comfort, while mechanical ventilation covers the ventilation requirements. [83, 86, 87]

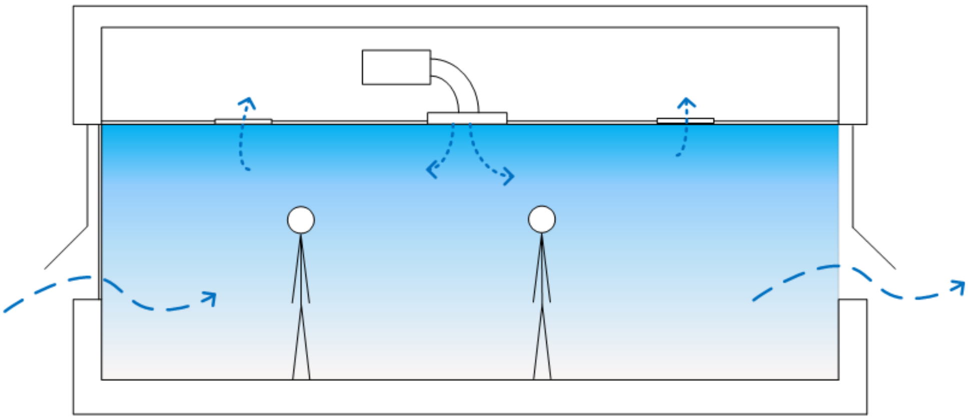


Figure 3.6.: *Illustration of a zone with concurrent mode ventilation*

Change - Over Mode

Change-over mode also falls into the complimentary classification, but natural and mechanical ventilation does not coincide, unlike concurrent mode. See Figure 3.7 for illustration. Buildings with this mode often have a set of different strategies that is switch between. These modes are used over different periods depending on the situation. The building automation system may determine the mode of operating based on outdoor conditions, an occupancy sensor, a window (open or closed) sensor, or based on operator commands. Buildings that can change their strategies over short periods often react to inside or outside conditions like CO_2 levels, interior or exterior temperature, humidity, or occupancy levels. The changes-over medium time period is a reaction to more extensive changes like day to night and long time period, such as a reaction to seasonal changes throughout the year. The parameters that dictate which timescale(s) is used include climate, building characteristics, and site location. [83, 86]

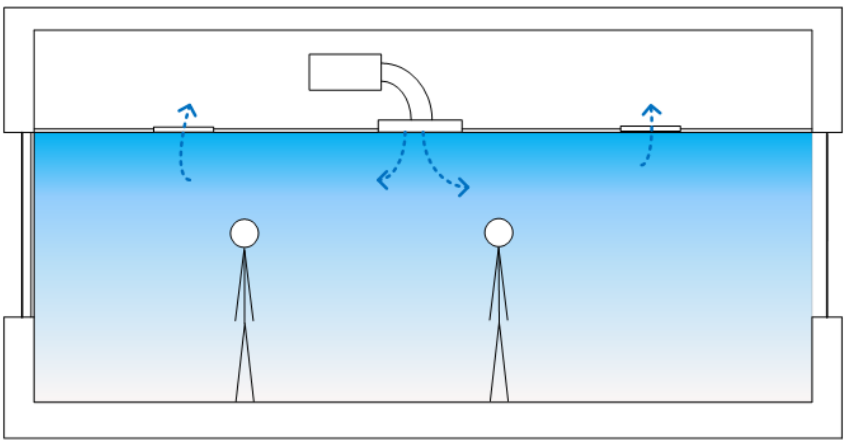


Figure 3.7.: *Illustration of a zone with change-over mode ventilation*

Alternate Mode

Alternate mode is a category describing a system that runs indefinitely on one system but can switch between modes manually. The switch between the modes is often tied to seasonal climatic variations. Alternate mode may consequently be fit into an expanded definition of the change-over mode. It is important to note the difference between manual and automatic operation control that differentiates the two modes. [83, 86]

Contingency Mode

Contingency mode is more about the structure of the building than the ventilation strategy itself. Contingency is a mode where only mechanical ventilation or natural ventilation is used for the entire building. Though this description does not fall under hybrid ventilation, the building is structurally built considering possible operation changes. For example, if there is a power outage or other disruption, buildings that are "natural ventilation ready" may still operate to some degree. We can therefore re-classify the building as "retrofittable" or even "Adaptable" to mixed-mode operation. [83, 86]

Zonal Mode

Zonal mode falls under the zoned classification, meaning natural and mechanical ventilation occurs simultaneously but in different building zones. Figure 3.8 illustrates this concept. Zonal mode is often used in buildings where the forces of natural ventilation cannot penetrate throughout the entirety of the building; thus, mechanical ventilation must be used. A building with zoned mixed-mode ventilation might have zones that are either exclusively natural or mechanical ventilated. However, the building may still have zones conditioned by the above strategies of the complimentary classification. It is important to note that both classifications could and should be used for different building parts to optimize ventilation. Zones, rather than buildings, should be the unit of mixed-mode classification [87]. [83, 86]

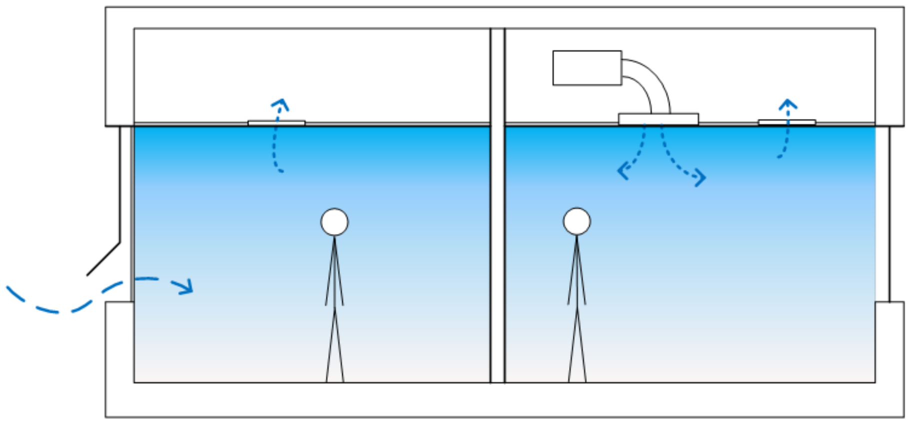


Figure 3.8.: *Illustration of a building with zonal mode ventilation*

3.5. Building Automation and Window Control

The control strategy of window operation is an essential factor for both natural and hybrid ventilation. As mention in the introduction, the building sector stands for about $40\,\%$ of the total energy usage in Norway. While most of the energy is used by primary building systems and equipment, a significant amount is wasted due to wrongly configured controls, sensors, and management systems [88]. This section presents window operation strategies used in building today and covers some relevant studies and reviews on the topic control strategy of window operation.

3.5.1. Manual and Automatic Control

Operable windows can either be controlled automatically, using an algorithm or designed models for optimizing ventilation, manually, by occupants adjusting to their own needs, or combining both. The criteria for an adequate control system are accuracy, speed/reaction time, and stability. A good window control system should give satisfactory indoor temperatures and IAQ without long delay or extensive temperature fluctuations.

Manual Occupancy Control

Manually controlled window operation gives occupants a sense of control over their thermal environment. As mentioned in Section 2.1.4, an occupant's ability to interact with their environment significantly affects their response to the thermal environment. Brager et al. have investigated the connection between operable windows and personal control in correlation to occupancy comfort [89]. Occupants with different degrees of personal control over window opening had significantly diverse thermal responses, even when they experienced the same thermal environments, clothing, and activity level. Occupants with a higher degree of control over the operable windows expressed comfort at higher temperatures than those with less control. Therefore, manual window control is an excellent tool to increase the range of temperatures within a zone occupants find acceptable. That being said, results from both Heidelberg [84] and Khatami et al. [90] studies show that occupants often react to slow changes in their thermal environment which may result in increased thermal discomfort. In addition, results from studies by Griffiths and Eftekhari [91]. Khatami et al. [90] suggest that occupants are often unaware of CO_2 levels and other parameters that indicate the state of IAQ. Automatic control may therefore be an essential tool to maintain a good IAQ through natural ventilation. Although this may be the case, Ackerly et al. [92] showed that when introducing automatic control in naturally ventilated buildings, the advantages of manual occupancy control may be eliminated. In addition, according to Frontczak et al. [93] occupants in most cases prefer manual controls in naturally ventilated buildings suggesting a combination of both automatic and manual control may be the best solution. More research appears necessary on this topic.

Automatic Window Control

Automatically controlled window operations use a mechanical motor that controls the opening mechanism in reaction to the state of selected setpoints of parameters. Occupancy control is traded off for a more optimized natural ventilation solution. Automatic control uses models and algorithms to evaluate the state of the environment and take appropriate actions to maintain setpoints for ventilation, air quality, and thermal comfort. The complexity of the controller allows for precise control and, in most modern buildings, energy savings and increased comfort for occupants. In a study on the effects of manual and automatic natural ventilation control strategies on thermal comfort, IAQ, and energy consumption by Khatami et al. [94], the introduction of automatic control increased the thermal comfort conditions for occupants, increased IAQ compared to the use of only manual control and helped to reduce energy consumption by $8\,\%$. The study revealed that occupants were much more aware of their thermal comfort than the IAQ state. Therefore, manual window control increased the risk of poor IAQ.

One group of automatic controls is the feedback control, which uses measured values from internal or external parameters to determine appropriate action from the system. Feedback controllers come in varying degree of complexity, the most common being the on-off controller. In connection with window control, an on-off controller will open the window when a control parameter reaches/passes a preset value and close when the control parameter reaches another. Feedback controls can also follow other principles like proportional (P), proportional and integral (PI), and proportional, integral, and derivative (PID) control. P-control increases the system's reaction depending on the deviation between the measured value and the desired setpoint value. The PI- and PID-controls have a high degree of complexity, as they account for the rate of the expected change in the system and the deviation from the setpoint value. The higher the complexity of the control principle, the higher the accuracy, but computation time will increase. High complexity is not always necessary, nor wanted. An assessment of the desired results must be made before choosing between the principals. Due to people's relatively wide range of comfort, on-off controllers are often found sufficient in keeping indoor environment within acceptable levels [95]. A study by Schultze and Eicher in 2013 showed that simple control strategy from window operations performed just as well as complex ones when controlling natural ventilation in energy-efficient buildings located in moderate climates. The choice of setpoint proved more important than the choice of control strategy [96]. [95]

Feed-forward control is another group of automatic controls. These types use predictions and forecasts to determine how the reaction of the system should be. Feed-forward controller could anticipate and prevent overheating a building using weather forecasts and knowledge of the building's thermal response. Feed-forward control in connections with ventilation controlled perform in most cases better than feedback controls. However, an accurate thermal model of the building has to be made at thoroughly tested before accurate predictions of the building's reactions to changing ambient parameters [97]. [95]

3.5.2. Hysteresis for Control Systems

Hysteresis is the lag in response exhibited by a body in reaction to changes in forces affecting it [98]. In control systems, hysteresis can be used to filter signals and avoid rapid operations changes. Rapid operations on windows may be perceived as distracting due to large fluctuations in outdoor temperatures. By adjusting the setpoints for the operation to a more considerable deviation in, for example, upper and lower indoor temperature limits, the output would react less rapidly than it otherwise would. For instance, if one wishes to maintain an indoor temperature of 22 degrees, then the setpoints for opening and closing could be set $20\,°C$ - $24\,°C$. When temperatures exceed $24\,°C$ windows would open and stay open until temperatures in the zone reach $20\,^{\circ}\mathrm{C}$ again. It was proven hard to find relevant studies on hysteresis control in context with window control. However, this method is an essential tool to avoid potential high frequency in window operation and will be used in the created window control algorithm of the thesis.

3.5.3. Control Schemes and Level of Automation

In a study on control schemes and level of automating window control by Chen et al. [99], three control schemes with varying levels of automation were examined. Spontaneous occupancy control informed occupancy control following instructional signals and fully automatic window control. The simulation was conducted to investigate the different scheme's effects on energy-saving and thermal comfort of occupants. A three-story building model was created to simulate energy consumption and indoor thermal comfort for the different cases. The climates from the following cities in China were used: Harbin (Severe cold), Beijing (cold), Shanghai (hot summer, cold winter), Guangzhou (hot summer, hot winter), and Kunming (Temperate). Only the results from a south-facing room on the second floor were used. A case using only the HVAC system was conducted to get a baseline for the energy use of cooling.

The fully automatic window control of this study was tested with two different schemes, *Heuristic Control* and *Model Predictive Control* (MPC). The heuristic control strategy analyses indoor and outdoor conditions and chooses the optimal ventilation strategy to ensure high IAQ. When specific indoor/outdoor requirements are met, the room ventilation switches to natural ventilation by opening the windows and turning the HVAC. MPC optimizes the operation of natural ventilation by evaluating and finding the best immediate action by evaluating a series of specific testing scenarios. Calculating multiple time steps forward from the current state controller finds the action that gives the best outcome of thermal comfort and energy saving. Thermal comfort was given higher priority than energy consumption in this case study.

The concept of occupancy comfort drives Spontaneous control. The zone occupants were given complete control over the window operations and were encouraged to use it as they see fit. Following on the concept that a person in discomfort will act to achieve comfort, occupants were expected to open windows when conditions in the zone became uncomfortable.

Informed occupancy control uses signals to notify occupants when and how to execute an action since occupants cannot predict the optimal control schedules to achieve maximum energy savings. For this study both *Fixed operation schedule* and *Stochastic occupant response* where simulated. Fixed operation schedule signals occupants on fixed times every day. Occupants take action suggested by the signal during a given hour and ignore the signals for the rest of the day. The signals are informed decisions based on information calculated from the heuristic control or MPC of the automatic control strategies. Stochastic occupant response signals occupants continually throughout the "wake time" of the day, $8\,\mathrm{am}$ - $11\,\mathrm{pm}$. The simulated cases operate with varying probability for occupants to take appropriate action. Three cases with probabilities $80\,\%$, $50\,\%$ and $20\,\%$ were simulated. Table 3.1 below shows the resulting percentage reduction in energy use of cooling for all the different cases.

Table 3.1.: *Energy reduction of cooling in the given cities compared to the baseline case for Automatic Control (AC), Spontaneous Occupancy Control (SOC), Fixed Occupancy Operation (FOO) and Stochastic Occupancy Reduction (SOR). Acronyms for the different strategies are used in the table for visual reasons.*

Energy Reduction of Cooling					
	Harbin	**Beijing**	**Shanghai**	**Guangzhou**	**Kunming**
AC (MPC):	50 %	27 %	31 %	17 %	80 %
AC (Heuristic):	38 %	22 %	13 %	10 %	66 %
SOC:	32 %	10 %	12 %	0 %	65 %
FOO (MPC):	35 %	8 %	18 %	2 %	69 %
FOO (Heuristic):	28 %	10 %	0 %	−2 %	66 %
SOR:	-	-	-	-	-

The fully automatic control shows excellent results. Both heuristic control and MPC reduces the energy demand substantially from the baseline case, and no thermal discomfort degree hours were registered in any of the studied cases. The number of window operations was significantly higher for MPC than heuristic control. The most common daily frequency of operations was 20 for MCP and two for heuristic control. The operation time mainly was during the day-time for MPC and morning, evening, and late-night for heuristic control. Consequently, both schemes are deemed hard to realizes without automatic control due to high frequency and inconvenient time for necessary operations.

The spontaneous control scheme could not maintain acceptable indoor temperatures during the simulated year; thus, both hot and cold occurrences were seen for all five simulated climates. Compared to the fully automatic control, spontaneous occupant control showed 1 % - 12 % and 16 % - 18 % lower energy savings than the heuristic control and MPC, respectively. The fixed schedule control also gave insufficient indoor air temperatures for both cases that followed heuristic control and MPC signals. The energy-saving were 10 % - 19 % lower compared to the fully automatic cases. In Shanghai and Guangzhou, the energy savings were near zero, and in one case, higher compared to the baseline.

Table 3.1 does not show the percentage energy savings of the stochastic occupant response cases as these values were not presented as exact numbers in the study. The performance of the stochastic occupant response diminished with the probability of the occupant's actions. The results showed a considerable drop in energy savings when the chance of occupancy compliance dropped from 50 % to 20 %. The cases following the MPC showed better performance in both thermal comfort and energy savings than the cases following the heuristic signals. Higher energy consumption was seen in all cases in Guangzhou following the heuristic signals, and the case of 20 % chance following the heuristic control signals in Shanghai. Though Stochastic occupant control showed better performance than that of the fixed schedule control, 80 % and 50 % showed a high frequency of operation for the MPC and signals at inconvenient timing for the heuristic control, much like that of the fully automatic cases. In theory, stochastic occupant control gives better results. However, in real-world scenarios, the operation frequency and timing could lead to a growing reluctance to respond to the signals, especially in the MPC signals. Eventually, the signals might get ignored altogether, resulting in higher energy use for cooling as the HVAC system is used.

This study has shown that natural ventilation has excellent potential to reduce energy usage while maintaining comfort for occupants. To achieve this, a proper controller for window operation that considers both ambient surroundings and occupancy behavior is crucial in buildings with mixed-mode ventilation. A more exact controller will optimize ventilation but increase complexity and the number of operations on the window, which should be considered when designing a controller. Brager et al. [89] also state, in the selection of natural ventilation control system, our analysis suggests that developers and building owners should not only consider the initial system investment and maintenance cost but also take into account the annual energy savings and occupant satisfaction to realize natural ventilation potential fully.

3.5.4. Machine Learning

The concept of machine learning is based on the automatic and continuous improvement of computers through experience. It is a form of Artificial Intelligence (AI) that allows computer programs to become more accurate at predicting desired outcomes without being explicitly programmed to do so. By using stored data as input, machine learning algorithms predict new output values. For many applications, it is now, with machine leanings technological progress, far easier to train a system by showing it examples of desired input-output behavior than to program it manually, a process that may take a large amount of time. It is an efficient tool used in an increasing number of areas, such as health care, manufacturing, education, financial modeling, policing, and marketing. Machine learning can be divided into four methods: Supervised, unsupervised, semi-supervised, and reinforcement learning. [100]

Machine Learning Methods

Supervised learning is task-driven and requires a data scientist to provide input, pair it with desired output and specify variables the model should analyze. Supervised learning is a precise method but the most complex and time-consuming. [101]

Unsupervised learning is data-driven and does not need a data scientist to specify any information. The model itself uses a deep learning method to come to conclusions on its own. It analyzes a large amount of unlabeled data and identifies patterns to group the data. [101]

Semi-supervised learning is a combination of both supervised and unsupervised learning. The algorithm is firstly provided with a small input and output from a data scientist. The algorithm learns the dimensions of the provided data and applies it to new unlabeled data. Supervised learning is precise but time-consuming, and unsupervised is the opposite. Semi-supervised learning strikes a middle ground between the two methods. [101]

Reinforcement learning works by giving the algorithm a set of distinct goals and a prescribed set of rules the algorithm must operate within. The algorithm seeks positive "rewards" programmed by a data scientist when the program moves towards desired results. The algorithm also receives "punishments" when the program's actions result in an output value further away from the desired result. [101]

Machine Learning Review

Dai et al. [102] carried out a literature review on studies using machine learning models that predicts occupancy and window opening behaviors and their applications in intelligent buildings. The extensive review shows that there has been a large increase in studies on machine learning in recent years. Two thousand one hundred forty-two studies were registered in 2019. On the topic of prediction of occupancy and window opening behaviors, 56 studies were selected for this review. Most studies on occupancy and window-opening behavior are based on supervised learning. The review shows that several different machine learning models for occupancy prediction and window-opening behavior have been successfully applied to buildings and have had a satisfactory performance. The energy-saving after a machine learning method had been implemented were, on average, for the different review studies, 23 %, showing that machine learning has potential for energy savings when implemented for window operations. Machine learning will not be used as a strategy for window operation in this study as the method is quite complex and requires a large amount of knowledge in coding and programming.

3.6. Ventilative Cooling Review

The Energy Performance of Building Directive has required that all buildings that are constructed after 2020 must achieve a level of nZEB (nearly Zero Emission building), leading to the demand for new energy and cost-efficient solutions in the building sector.

The following section will present the findings from a literature review regarding the state-of-the-art ventilation strategies to increase the energy efficiency of ventilated buildings.

3.6.1. Ventilative Cooling

As mentioned in the introduction of this thesis, high-performance buildings are among the most profitable greenhouse gas emission reduction measures in the building sector. Cooling demands are increasing as buildings become more insulated and air-tight, often resulting from underestimation/neglection of cooling requirements based on earlier building's earlier experiences and rules of thumb. This results in high temperatures being frequently reported in modern energy-efficient buildings, even in Nordic climates. A survey of the court of auditors in Vaud Canton in Switzerland showed that in 9 out of 10, sustainable and high energy performance state-buildings presented overheating problems. The survey also showed that buildings with mechanical ventilative cooling presented very high electrical usage [103]. In addition, to avoid thermal comfort issues and draught, limited temperature differences between supply air and room are required, making heat recovery or air pre-heating necessary. The reduced cooling capacity of the supplied air and increased airflow rates become necessary, further increasing the energy consumption of the Air Handling Unit (AHU) fans. Ventilative cooling via natural ventilation is seen as the only means of evacuating heat from a building without increasing energy consumption. [104]

Ventilative cooling is defined as:
"*the application of the cooling capacity of the outdoor airflow by ventilation to reduce or even eliminate the cooling loads and the energy use by mechanical cooling in buildings, while*

guaranteeing a comfortable thermal environment" [104].

Ventilative Cooling Potential of Different Climates

Ambient and indoor air temperature differences primarily determine the potential of ventilative cooling and its application. In colder climates, the cooling potential will naturally be higher than that of hotter climates. In addition, that the potential will vary with cold and hot periods throughout the day and year. In colder climates, there is often a need for evaluating the potential of draught risk due to low ambient air temperatures. In comparison to hotter climates, ventilative cooling may only be applicable during the night when outdoor temperatures are lower. The potential of ventilating cooling utilization must be evaluated early in the design stage of the building for optimized use of the strategy.

The International Energy Agency Energy in Buildings and Communities (IEA EBC) carries out research and development activities known as annexes with the goal to move toward near-zero energy and carbon emissions in the built environment. The Annex 62 research project investigated the implementation of ventilative cooling in 91 case studies of real buildings from several different countries and climate zones [104]. To illustrate the potential expectations of ventilative cooling performance in different climates, the expected thermal comfort and cooling requirement reduction predictions have been calculated for the same building configuration located in different climates. The result of this study and a general conclusion from all the different studies of annex 62 showed that ventilative cooling could have a considerable impact on reducing overheating hours. In cold and temperate climates, the risk of overheating can be eliminated entirely, while supplemented cooling solutions were needed to maintain acceptable comfort in warm and hot climates. In cold climates like Oslo, ventilative cooling via natural ventilation was sufficient to remove all cooling loads.

In contrast, it was essential to apply night-time cooling strategies to remove excess heat from the building in most other climates. Warm and hot climates are essentially dependent on night-time cooling strategies as day-time cooling rarely has any effect due to the high ambient temperatures. A study by Oropeza-Perez investigating the energy performance of natural ventilation as a passive cooling method on buildings located in temperate countries [105]. Using Denmark as a case study shows that a large reduction in mechanical ventilation usage was possible. During June, July, and August, $90\,\%$ of hours where mechanical ventilation was used could be replaced with passive ventilation and still maintain thermal comfort for occupants. Oropeza-Perez also investigated the potential of passive cooling in warmer climates, sing Mexico as the case study [106]. The results suggest that the energy consumption for cooling could be reduced by up to $54.4\,\%$ with the use of passive cooling.

3.6.2. Night-time Cooling

In a study by Artmann et al. on passive cooling of buildings in Europe, the climatic potential of night-time ventilation was investigated [107]. The basic concept of night-time cooling utilizes the relatively cold air during the night to cool the structure/thermal mass .The cooling of the building provides a heat sink available during occupancy hours. This leads to a decrease, abatement, or postponing of temperature peaks and better thermal comfort for occupants during the day. The potential for passive cooling of buildings by night-time ventilation was evaluated by analyzing climatic data without considering any building-specific parameters. The study showed that night-time ventilation has a high potential as a ventilative cooling strategy in Northern Europe and significant potential in Central, Eastern, and some regions of southern Europe. Some series of warmer nights showed that night-time ventilation alone might not be sufficient to guarantee thermal comfort in every scenario.

A state-of-the-art review regarding passive cooling techniques was performed by Santamouris and Kolokotsa in 2013, reviewing the effects and potential of passive cooling on residential and non-residential buildings [108]. The review states that passive cooling technologies can provide comfort in non-air-conditioned buildings and decrease considerably the cooling load of thermostatically controlled buildings. The proposed technologies have been tested in demonstration and real-scale applications. With the implementation of passive cooling, the expected energy saving can reach 70% compared to the conventional air-conditioned building. This result was depended on the rate and duration of the ventilation, the building mass, and the surrounding climatic conditions. A reviewed study regrading implementation of night-time ventilation to lightweight construction in northern China showed that the indoor surface temperature of the building could be decreased by up to $3.9\,°C$ depending on the duration of the operation [109]. The study also showed that more energy could be saved in office buildings cooled by night ventilation systems than those that did not employ the strategy. Night-time ventilation was found as an efficient method to reduce air conditioning demands for office buildings and improve thermal comfort during the day-time.

3.6.3. Annex 62: Ventilative Cooling in Existing Buildings

The state-of-the-art review of ventilative cooling presents 26 studies of the 91 case studies conducted under the annex 62 research project [104]. These cases show a wide variety of different building types like office, educational and residential buildings. Some cases involving both office and educational buildings located in colder climates will be presented further in this section, as the used strategy shows potential for application to the ZEB Laboratory.

Ventilative Cooling in Educational Buildings

Mellomhagen School in Norway was originally built in 1960 and retrofitted with new insulation, new windows, and a hybrid ventilation system in 2010. The ventilation system combines controlled opening of windows to promote air exchange and an extraction fan when natural ventilation is either inadequate or inadvisable due to low outdoor temperatures. Each window is divided into two parts where the lower functions as a normal window, while the motorized system controls the smaller, upper part. During the winter period, window operation is limited to avoid cold draught and large heating demands. Window operations are activated in periods when indoor temperatures exceeds $21\,°C$ or if the CO_2 concentration exceeds $1300\,ppm$, and is limited to a $50\,\%$ maximum opening. The CO_2-setpoint of $1300\,ppm$ operates all year round. A local weather station records wind conditions, temperature, and rainfall, and these values are used to determine, in combination with occupancy schedules, the control and timing of window operations. In periods of inadequate conditions, the exhaust fan will then control the ventilation. Occupants may override the control system. During the summer period, the zone setpoint temperature is $22\,°C$ and window operations only occur during periods with higher indoor temperatures. Windows ventilation will be utilized during night-time when indoor temperatures exceed 23 °C. This study found that the CO_2 concentration need higher priority, as they found that they had to focus more on the IAQ than energy reduction. [104]

Solstad kindergarten in Norway desired to reduce the energy consumption to half of what TEK07 required. The solution had to be cost-effective, and natural ventilation controlled by motorized windows was chosen. During the winter period, both mechanical and natural ventilation control operates after a CO_2-setpoint. Mechanical ventilation operates with a zone setpoint of 900 – 1200 ppm, whereas window operation has a CO_2-setpoint of 950 – 1500 ppm, significantly higher, to avoid the use of natural ventilation as much as possible during the colder periods of the year. Only occurring if the mechanical ventilation system is insufficient. Window operations are only allowed when indoor temperatures exceed 19 °C and are limited to a 50% maximum opening. During the summer period, the zone setpoint for window operation is set to 21 °C, while the mechanical ventilation system is controlled by a CO_2 - setpoint of 900 - 1300 ppm. Night-time cooling occurs if temperatures exceed 23 °C outside of occupancy hours. During the summer season, mechanical ventilation is not utilized very often under these simulation conditions. The airflow rates needed in order to remove surplus of heat are shown to be often sufficient to keep CO_2 concentration below acceptable values.

Ventilative Cooling in Office Buildings

CIT Zero 2020 Building in Cork utilizes single-sided natural ventilation to meet cooling demand and reduce energy consumption. Window operations are available during occupancy hours through a combination of manual and automatically controlled openings for increased ventilative cooling. The controlled window operations operate after an air temperature setpoint of 21 °C under the conditions that the outside temperature exceeds 15 °C. Due to very high ambient temperatures during the period of performance testing in the summer of 2013, many hours were considered too hot. Generally above 25 °C in July and above 23 °C in August. However, the overall occupant feedback was generally positive throughout the summer.

The police office in Schoten, Belgium, uses buoyancy forces to drive natural ventilation to achieve good thermal comfort and indoor air quality through both day and night ventilation. The night ventilation is automatically controlled and operates between 10 pm and 6 am. Indoor temperatures must be above 21 °C, and there must be a temperature difference of 1 °C between indoor and the ambient conditions. Openings are closed if wind speeds exceed 10 m/s or if rainfall is noticed. Day-time ventilation activation requirements are maximum indoor temperatures exceeding 24 °C and the average outdoor temperature exceeding 12 °C. The actual control of the ventilation is dependent on manual occupancy control in the individual offices of the building and automatically controlled after a CO_2-setpoint of 600 - 900 ppm for the landscaped offices. Users can manually open windows as they please as well. Good thermal comfort was achieved during the warm summer period, except for periods where the outside temperature exceeded 30 °C. Some hours of low temperatures in the mornings of the summer period in some of the landscaped offices indicating that a lower degree of night cooling may be applicable. This could potentially be solved with a higher setpoint temperature for night ventilation.

The cases of both educational and office buildings generally confirm what has already been stated in the presentation of the Annex 62 project. Northern countries with colder climates can utilize ventilative cooling during the summer months to eliminate the overheating risk while maintaining a good thermal and atmospheric environment. Night-time cooling is a great strategy to cool the building mass and avoid overheating during the day but was rarely found necessary to reach setpoint temperatures as passive cooling during the day was sufficient.

3.6.4. Nydalen Vy

Futurebuilt is a program launched in 2010 that aims to develop carbon-neutral urban areas and high-quality architecture across 50 pilot projects by 2020 [110]. One of these projects is the mixed-use building Nydalen Vy, consisting of office, residential, and industrial units. It is a 16 story building with the ambition of achieving the status of nZEB [111]. The building will not require externally supplied energy from ventilation, heating, or cooling of the building, so-called *TripleZero*, and the office section of the building is purely naturally ventilated. The purpose of the building is to demonstrate that the environmental buildings can be made more simply and robustly than they are today while achieving satisfactory conditions through architectural and technological solutions that complement one another. Furthermore, geothermal production, geothermal cooling, and PV panels on the building facade are on-site energy production.

The natural ventilative system is supplied by *Windowmaster*. According to Windowmaster, natural ventilation is automatically controlled by opening external openings on the facade. Parameters such as external and internal temperature, CO_2 levels and pressure created from the velocity and direction of the wind determine how much air is supplied through the external openings [112]. During the summer, the air is supplied directly into the occupied zone through openings high up on the facade. During the winter period, the air is supplied in veneer chambers located in the ceiling, from which air seeps into the zone through perforations, eliminating the potential of draught.

Nydalen Vy is a showcase project inspired by the successfully constructed TripleZero office building, located in Lustenau Austria, 2226 [113]. It is named after the required comfortable indoor temperature range of 22 °C to 26 °C. The required heat gain of the building is provided by lighting, computers, and occupants that inhabit the building. At the same time, natural ventilation is used through manually or automatically controlled windows that react to the internal CO_2 levels. The 2226 project was proven successful as a satisfactory indoor environment was achieved, with CO_2 levels mostly staying below 1200 ppm, internal temperatures in the required range 22 °C to 26 °C, and a zero-energy demand from heating, cooling, and ventilation [114]. The Austria climates differ from that of Norway. Therefore, there is some uncertainty regarding the resulting quality of the indoor environment in Nydalen Vy that can only be determined through testing.

3.6.5. Optimal Ventilation in ZEB Laboratory

A master thesis conducted in 2019 by Maren Elise Leinum examined how to combine natural and mechanical ventilation in an optimal way to provide sufficient ventilation for the ZEB Laboratory throughout the year, concerning energy use and indoor climate [11]. The results showed that the most optimal way to control the ventilation varied between seasons. Clean mechanical ventilation was the most energy-efficient solution in the winter season due to the ambient conditions being utilized to utilize passive cooling. During the transition seasons (Spring and Autumn), both lower temperature differences between indoor and outdoor conditions and lower heating demand occur. The use of clean mechanical ventilation was the solution with the lowest heating demand, and a hybrid ventilation solution gave the lowest fan power demand. It was also shown that clean natural ventilation strategy resulted in close to satisfactory indoor environment levels. For the transition seasons, the hybrid ventilation strategy was seen as an acceptable solution. Lastly, during the summer season, a clean natural ventilation solution with morning aeration was found to be sufficient to keep the amount CO_2 within acceptable levels during occupancy hours.

The simulations were conducted in the simulation tool CONTAM, a tool unable to simulate models with implemented internal heat gains, temperature changes in zones, or zone-to-zone airflow. These simplifications mainly affect the energy demand for heating and thermal comfort, which may considerably affect which ventilation strategy is the most optimal for the given season. Simulations with implemented internal gains would result in lower heating demand, higher indoor temperatures, and better possibilities and opportunities to utilize natural ventilation. If the results of this study are to be used for this master thesis, then the results have to be controlled checked with the implementations of internal gains in IDA ICE.

4. Introduction of the ZEB Laboratory

4.1. Zero Emission Building

Zero emission buildings, as defined by the Norwegian Research Centre on Zero Emission Buildings, is a building that produces enough on-site renewable energy to compensate for the building's greenhouse gas emissions over its life span [5]. The ZEB definition is further characterized by the Norwegian Research Centre of Zero Emission Buildings through various ambition levels of renewable energy compensation, ranking from ZEB-O to ZEB-Complete. A ZEB-O building aims to produce renewable energy to compensate for the greenhouse gas equivalent emissions from the operation of the building. ZEB-Complete aims to produce renewable energy to compensate for the greenhouse gas equivalent emissions from construction and operation of the building, production of the materials, and demolition/recycling at the end of its lifetime. [5]

The ZEB Laboratory aims to achieve a level of ZEB-COM, the fourth and second to most ambitious ZEB level [9]. The energy use and greenhouse gas emission that is produced from the construction stage, operation of the building, and the production of the materials should be reduced as much as possible, and on-site production of renewable energy should be sufficient enough to compensate for that amount of greenhouse gas emitted over 60 years of its lifetime. [5, 8]

4.2. ZEB Laboratory

The ZEB Laboratory is a living laboratory that is located at campus Gløshaugen in Trondheim. The construction of the building was completed in August 2020 and was ready for use at the turn of the year 2021.

4.2.1. Purpose and Ambition

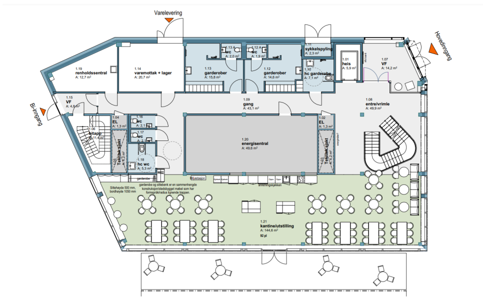
The vision of the ZEB Laboratory is for it to be an arena where new and innovative components and solutions are developed, investigated, tested, and demonstrated in mutual interaction with the occupants of the building [9]. The building will form a living office laboratory that continuously collects experimental data throughout everyday office work or for educational purposes. The building's facades, components, and technologies can be modified and replaced, which gives the building the adaptability to investigate different building configurations, technologies, and usages that can be implemented in other designs and constructions of zero emission buildings. The goal being scientific contribution that reduce the entrepreneurial risk for companies that are willing to invest in passive house technology and zero emission buildings. The ambition of the ZEB Laboratory is, in summary: to act as a role model and testing ground for future projects that may involve architectural attributes, material use, control strategies, and technologies.

4.2.2. Building Structure

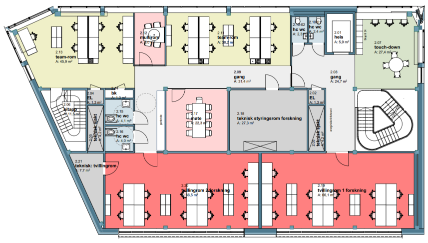
The ZEB Laboratory is a four-story-high living office laboratory of approximately $2000\,\mathrm{m}^2$. Its an experimental facility located in NTNU Gløshaugen campus in Trondheim, close to the existing facilities of SINTEF Community and NTNU Department of Civil and Environmental Engineering. The materials for different parts of the building are mainly made from wood. Different laminated timber for the interior surfaces and columns and wood frames insulated with glass wool as the outer wall. PV panels cover the roof, the whole southern facade, and some parts of the east and west facade to ensure satisfactory on-site renewable energy generation. The entirety of the southern facade or just the window elements are designed to be replaced and rebuilt. The ZEB Laboratory can apply and investigate new components and technologies to optimize the building envelope and building performance. [9]

Building Body

Figure 4.1 presents the floor plans of every floor of the ZEB Laboratory. The floor plan includes the zonal division of each floor. Appendix A presents larger images of each floor plan of the ZEB Laboratory.

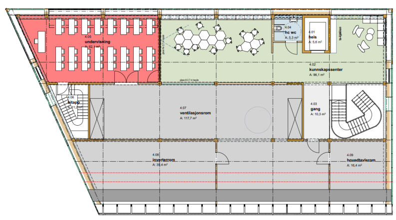


(a) *Ground floor of the ZEB Laboratory*



(b) *First floor of the ZEB Laboratory*

(c) *Second floor of the ZEB Laboratory*



(d) *Third floor of the ZEB Laboratory*

Figure 4.1.: *Floor plan drawing of all floors of the ZEB Laboratory (permission for display given by Cecilie Schei, Civil Architect, Link Arkitektur)*

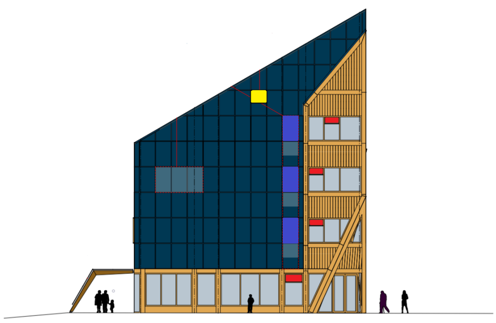
The ground floor consists of the building's different entrances, a cafeteria, receipt of goods, bathrooms, wardrobes, and an energy plant. The first and second floors consist of different workspaces like team rooms, touch-down areas, meeting rooms, and bathrooms. What differentiates the two floors are the twin rooms on the first floor, two identical research areas. The second floor also has a more open workplace design, though both the first and second floors consist of flexible areas to vary the workplace design. On the third floor, there will be an auditorium, toilets, a touch-down area, and a technical room. The technical room will act as a "showroom" where the technical solutions of the building may be presented.

Twin Rooms Test Facility

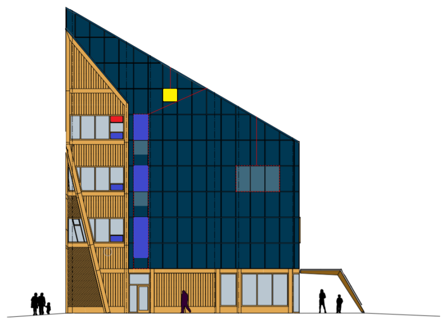
The ZEB Laboratory has two identical office rooms of $66\,\mathrm{m}^2$ on the first floor. They are each equipped with independent HVAC systems, a dedicated control room, and an increased number of sensors that monitor the influential parameters of occupant comfort. The twin rooms allow for comparative and close to calorimetric studies. NTNU and SINTEF have expertise in living laboratories with their earlier projects ZEB Test Cell Laboratory [115], and the ZEB Living Lab [116]. Both were built for environmental research on, respectively, office rooms and residential buildings. Most attractive solutions for passive buildings require a large-scale facility for implementation and testing to achieve realistic conditions. The ZEB Laboratory provides this in contrast to the other facilities. [9]

Openings on Facades

About 28% of the total facade area is made up of windows. A large share of these windows will be openable, either controlled automatically or manually by occupants. Figure 4.2 shows the facades of the ZEB Laboratory. Appendix B presents larger images of each facade of the ZEB Laboratory.

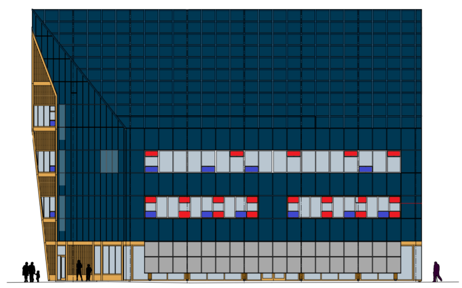


(a) *East facade of the ZEB Laboratory*



(b) *West facade of the ZEB Laboratory*

(c) *North facade of the ZEB Laboratory*



(d) *South facade of the ZEB Laboratory*

Figure 4.2.: *Architectural drawing of the facades of the ZEB Laboratory (permission for display given by Cecilie Schei, Civil Architect, Link Arkitektur)*

The red squares represent the windows that are automatically controlled, and these are placed on the upper part of the floor facade though there are exceptions on the south facade facing into the twin zones. The blue squares represent the manually operable windows exclusively placed on the lower part of the floor facade. The yellow squares represent the motor-controlled fire hatches.

4.2.3. Occupancy Hours

Occupancy hours follow standard opening hours of NTNU from $7\,\mathrm{am}$ to $18\,\mathrm{pm}$, with core time between $8\,\mathrm{am}$ and $16\,\mathrm{pm}$. The number of occupants present throughout the day will vary as the building is used for different purposes, office work, lectures, studying. Different user profiles are presented in documents from the project description and have been used to create an expected occupancy profile for the different parts of the building. This will be presented in the next chapter of this report. [117]

4.3. Building Service - HVAC

The ZEB Laboratory is facilitated with the ability to explore different ventilation strategies, natural, mechanical, or hybrid/mixed. The following section will present the building's properties and equipment for ventilation.

4.3.1. Natural Ventilation of the ZEB Laboratory

Natural ventilation is implemented in the ZEB Laboratory through the opening of doors and both manually and automatically controlled doors. Window placements are designed to ensure cross ventilation in zones and between floors. The main staircase in the building works as an extract duct for both natural and mechanical ventilation, with an outlet in the form of a large extract grate on the third floor, placed next to the staircase opening. Thermal buoyancy and a fan on the extract duct drive flow. The twin rooms, as mention, are equipped with independent HVAC systems and can extract air via ducts in different configurations. The rooms are also equipped with operable windows, both manual and automatic. In contrast to the rest of the building, automatically controlled windows are placed both at the lower and upper part of the wall, making it possible to implement different natural ventilation strategies like single-sided and cross ventilation. For the purpose of this report, the twin zones are simulated as a part of the First floor zone. [9]

4.3.2. Mechanical Ventilation of the ZEB Laboratory

The ZEB Laboratory is equipped with a central mechanical ventilation system. Though different air distribution systems were designed for the four floors in the building, they all rely on the displacement ventilation principle. On the ground floor, air is supplied at floor level through inlet devices. The First floor supplies air with porous ceiling boards from the suspended ceiling, in the second, through slots, and in the third floor through wall air terminals placed at floor level.

The exhaust air will be removed from the building envelope through exhaust ducts placed in wardrobes, toilets and through the exhaust duct placed at the top of the staircase on the third floor. A heat recovery unit with an annual average efficiency of $> 80\%$ is installed in the ventilation system. No cooling system is installed on the central mechanical ventilation system. However, the ventilation system connected to the twin zones is an exception, as they do have the ability to apply both heating and cooling to their respective zones. The mechanical ventilation system must satisfy regulatory requirements for material and personal load in regards to supplied air.

5. Method

This chapter presents the method used for this thesis. The chosen simulation tool IDA ICE, the modeled ZEB Laboratory and a description of how the results will be evaluated is presented in this chapter.

5.1. IDA ICE as a Simulation Tool

IDA ICE is a dynamic simulation tool used to analyze a building's performance concerning energy consumption, indoor climate, air quality, and thermal comfort. It is a three-dimensional tool that allows for multi-zonal simulation on a model with arbitrary geometry. The program is based on modules that describe the behavior of different parameters and building components. This implies that parameters like temperature points, the intensity of different internal loads, and user patterns can be specified for each zone, in addition to technical specifications like HVAC, and energy systems for the entire model or individual parts. This provides the opportunity to run detailed simulations and control and log desired parameters and results. IDA ICE uses equations-based technology, which allows the user to inspect how the model was created, meaning that there is no "black box" that needs to be trusted. This gives the user extensive control over the model and a large responsibility for the results. Most other simulation tools on the market focus either on the setup of the building model, the different systems placed in the model, or controllers for the different systems. IDA ICE accurately models all these simultaneously to ensure the lowest possible energy consumption and the best possible comfort for occupants. [118, 119]

There are some disadvantages to IDA ICE. Firstly, it is a complex simulation tool that takes time to understand and master. Running simulations takes much computational power and may take a long period to simulate, especially when modeling large and complex buildings. Further, there is also a slight drawback to the geometry builder in IDA ICE. It is unable to create slopped facades on the building geometry. The tool can run simulations of more complex geometry, but the model must then have been imported as a CAD file for another program. The tool is able to simulate more complex air movement with the right add-on but this is outside of the scope of this thesis. Without this add-on the tool can only simulate air movement in and out of defined zones, and therefore, draught risk may be challenging to determine.

Earlier reports researching and simulating the ZEB Laboratory have used both the simulation tool CONTAM, and Design-Builder [11, 120]. Though both reports express satisfaction with the simulation tool for their respected studies, disadvantages are also mentioned. The main drawback of Design-Builder is the lack of zone-to-zone airflow analysis, which is an essential trait when studying natural and hybrid ventilation and simulating a multi-zonal building. It is crucial to consider zone-to-zone airflow to achieve realistic air quality, thermal comfort, and energy usage. The simulation tool CONTAM can simulate airflow between zones but cannot account for the heat transfer between them. Air temperatures will remain equal to the pre-set value through the simulation. The zone is not affected by internal gains, solar heat gain, air supplied, among other parameters that affect temperature variation. This results in an airflow pattern that does not comply with real-world scenarios. IDA ICE is therefore seen as an overall better simulation tool for the purpose of this thesis.

5.2. Presentation of the modeled ZEB Laboratory

This section presents the modeled ZEB Laboratory. The model is a simplified version of the ZEB Laboratory constructed to determine temperatures, airflow rates, and airflow patterns that can occur in the building throughout different periods of the year. The primary purpose of the simulated cases is to investigate potential of ventilative cooling in the ZEB Laboratory. The model has been created following an *as-built* document obtained through the project description. Values and parameters used in the model are selected following this document unless specified otherwise [121].

5.2.1. Building Structure

The model of the ZEB Laboratory is designed with a floor area on each story of $440\,\mathrm{m}^3$ and a height of $22.77\,\mathrm{m}$. The geometry of the building is constructed following the plan drawings presented in Section 4.2. Due to the limitation of IDA ICE, some alterations have been made to the building geometry, mainly the inward slops on the facades. The effects these alterations may have had on the simulated results are considered to be negligible due to the minimal increase in floor area. The model is divided into four main zones, which each respectfully takes up close to the entire space of the four floors. The zones are: "Ground Floor", placed at $0\,\mathrm{m}$ above ground level, "First floor", placed $4.45\,\mathrm{m}$ above ground level, "Second Floor", placed $8.3\,\mathrm{m}$ above ground level, and "Third Floor", placed $12.15\,\mathrm{m}$ above ground level. From this point on, these zones, when presented together, will be presented as the "Floor zones". The fifth zone is dubbed "Staircase" and is a smaller zone that moves from ground level up to the top floor. It is placed on the East side of the model and represents the open staircase of the building. The zone passes through every floor and is equipped with openings placed on every wall connecting to the floor zones to allow free airflow throughout the model. Every zone except for the "staircase" is equipped with an ideal heater with an installed effect of $160\,\mathrm{W/m}^2$. The setpoint temperature for heating is $21\,^{\circ}\mathrm{C}$. The specific heat emitted from lighting and equipment is $2.4\,\mathrm{W/m}^2$ and $3.2\,\mathrm{W/m}^2$ in the floor zones. A clipping of the modeled building is presented in Figure 5.1 below.

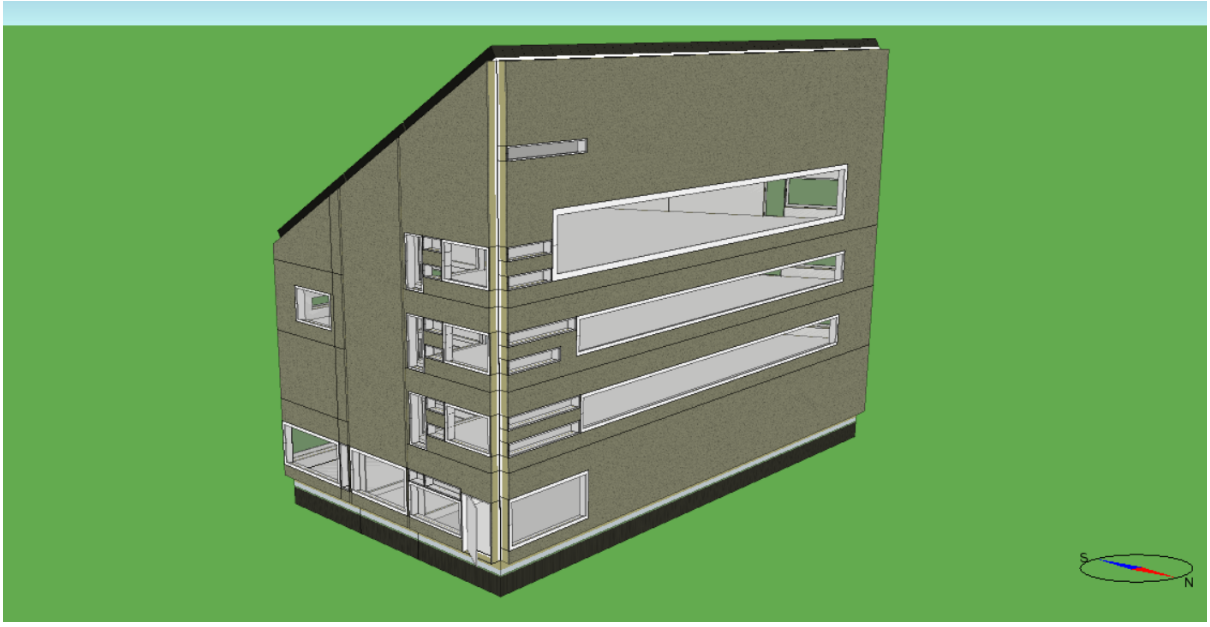


Figure 5.1.: *The modeled ZEB Laboratory shown from the North-East*

To achieve a level of ZEB-COM, a sufficient thermal transmittance level must be met. The *U-value* for the different parts of the buildings construction is presented in Table 5.1 below.

Table 5.1.: *The thermal transmittance of the models structure*

Construction Part	U-Value
External walls	$0.15\,\mathrm{W/m^2K}$
Roof	$0.9\,\mathrm{W/m^2K}$
Floor towards ground	$0.1\,\mathrm{W/m^2K}$
Window	$0.77\,\mathrm{W/m^2K}$
Thermal bridges	$0.04\,\mathrm{W/m^2K}$

5.2.2. External Openings

The external openings of the modeled ZEB Laboratory consist of doors and windows. The following section presents the properties and placement of these openings. All doors are manually operated, and the windows are either automatically or manually controlled or inoperable. The discharge coefficient inflow for large opening C_d has been set 0.65 for all openings. In reality, the discharge coefficient is, as explained in Section 3.3.1, a function of the temperature difference, wind velocity, and opening geometry and varies throughout the simulation, but IDA ICE is not capable of changing the discharge coefficient as it is conducting a simulation. The default value set by IDA ICE is therefore used.

Doors

There is one door in the IDA ICE model implemented on the ground floor, the main entrance. The door has a height of $2.99\,\mathrm{m}$, a width of $2.00\,\mathrm{m}$ and allows for two-way airflow. The opening schedule is based on assumptions and expected usage. The main entrance is expected to have a $20\,\%$ usage pattern between $7\,\mathrm{am}$ to 09:30 am and $15\,\mathrm{pm}$ to $17\,\mathrm{pm}$, the period where most occupants are expected to arrive and leave the building.

Windows

All windows have a U-value of $0.77\,\mathrm{W/m^2K}$, as presented in Table 5.1, frame factor of 0,2 and all windows have a solar heat gain $\mathrm{g_{tot}}$ of 0,45. The South facade is equipped with external shading that reduces the solar heat gain to 0,1 when in use. The East and West sides have internal shading that reduces the solar heat gain to 0,35 when in use.

Initially, the window placement was intended to mimic the facade drawings, presented in Section 4.2, but due to extensive computational time for simulations, every window on the same facade, with identical construction and operation controller, were merged into one window. IDA ICE allows for this simplification as ambient conditions affect the facade as a whole. Automatically and manual controlled windows were only merged if windows were placed at the same height on the facade. Table 5.2 presents the specified area of window with automatic, manual, and no controller on all facades and all floors.

Table 5.2.: *Specified area of windows on all facades and on all floors, separated by the controller used on the window*

	Controller	**Window area**			
		North facade	**East facade**	**South facade**	**West facade**
Ground floor	**Automatic**	$0\,\mathrm{m^2}$	$0.78\,\mathrm{m^2}$	$0\,\mathrm{m^2}$	$0\,\mathrm{m^2}$
	Manual	$11.45\,\mathrm{m^2}$	$0\,\mathrm{m^2}$	$0\,\mathrm{m^2}$	$0\,\mathrm{m^2}$
	None	$13.9\,\mathrm{m^2}$	$35.27\,\mathrm{m^2}$	$55.18\,\mathrm{m^2}$	$10.32\,\mathrm{m^2}$
First floor	**Automatic**	$3.09\,\mathrm{m^2}$	$0.85\,\mathrm{m^2}$	$4.72\,\mathrm{m^2}$	$0\,\mathrm{m^2}$
	Manual	$3.28\,\mathrm{m^2}$	$4.24\,\mathrm{m^2}$	$3.54\,\mathrm{m^2}$	$4.37\,\mathrm{m^2}$
	None	$45.11\,\mathrm{m^2}$	$5.93\,\mathrm{m^2}$	$32.94\,\mathrm{m^2}$	$7.92\,\mathrm{m^2}$
Second floor	**Automatic**	$2.90\,\mathrm{m^2}$	$0.85\,\mathrm{m^2}$	$3.88\,\mathrm{m^2}$	$0\,\mathrm{m^2}$
	Manual	$2.34\,\mathrm{m^2}$	$4.24\,\mathrm{m^2}$	$3.09\,\mathrm{m^2}$	$4.37\,\mathrm{m^2}$
	None	$46.23\,\mathrm{m^2}$	$11.96\,\mathrm{m^2}$	$32.21\,\mathrm{m^2}$	$15.84\,\mathrm{m^2}$
Third floor	**Automatic**	$5.1\,\mathrm{m^2}$	$3.09\,\mathrm{m^2}$	$0\,\mathrm{m^2}$	$0.72\,\mathrm{m^2}$
	Manual	$1.82\,\mathrm{m^2}$	$4.24\,\mathrm{m^2}$	$0\,\mathrm{m^2}$	$4.37\,\mathrm{m^2}$
	None	$83.78\,\mathrm{m^2}$	$5.93\,\mathrm{m^2}$	$0\,\mathrm{m^2}$	$7.94\,\mathrm{m^2}$

The automatically control windows are all placed $2\,\mathrm{m}$ above floor level on all zones. There are two exceptions, a $3.28\,\mathrm{m^2}$ window on the North facade on the third floor, which is placed $6.4\,\mathrm{m}$

above floor level and a $3.54\,\mathrm{m}^2$ window on the south facade on the first floor, which is placed $3.54\,\mathrm{m}$. The manually controlled windows are all placed $0.7\,\mathrm{m}$ above floor height.

Cracks and Small Openings on Facade

Some inward or outward air leakage will occur through the building body other than through the opening of windows and doors. These airflows are assumed to occur through small openings in the facade. These air leakages are driven by pressure differences across the building envelope due to a combination of pressure created by buoyancy, wind, and mechanical ventilation. Ex- and infiltration is implemented in IDA ICE as the airtightness of the modeled building. The value $0.3\,\mathrm{h}^{-1}$ at pressure difference $50\,\mathrm{Pa}$ is used.

5.2.3. Mechanical Ventilation

The mechanical ventilation of the building is implemented as a single AHU. In reality, two identical AHUs cover the main areas of the ZEB Laboratory, and one AHU for each of the twin zones, but to reduce computational time, one will be used for the entire model. The staircase zone is the only zone in the model that is not connected to the AHU. The ventilation is a balanced CAV system that supplies and extracts air from the different zones with an airflow rate of $6\,\mathrm{m}^3/\mathrm{hm}^2$, with constant $19\,°\mathrm{C}$, and with the air distribution strategy, displacement ventilation. The airflow rate value is taken from the project description and controlled to check against the standard required airflow rate specified by TEK17. An illustration of the AHU is presented in Figure 5.2 below.

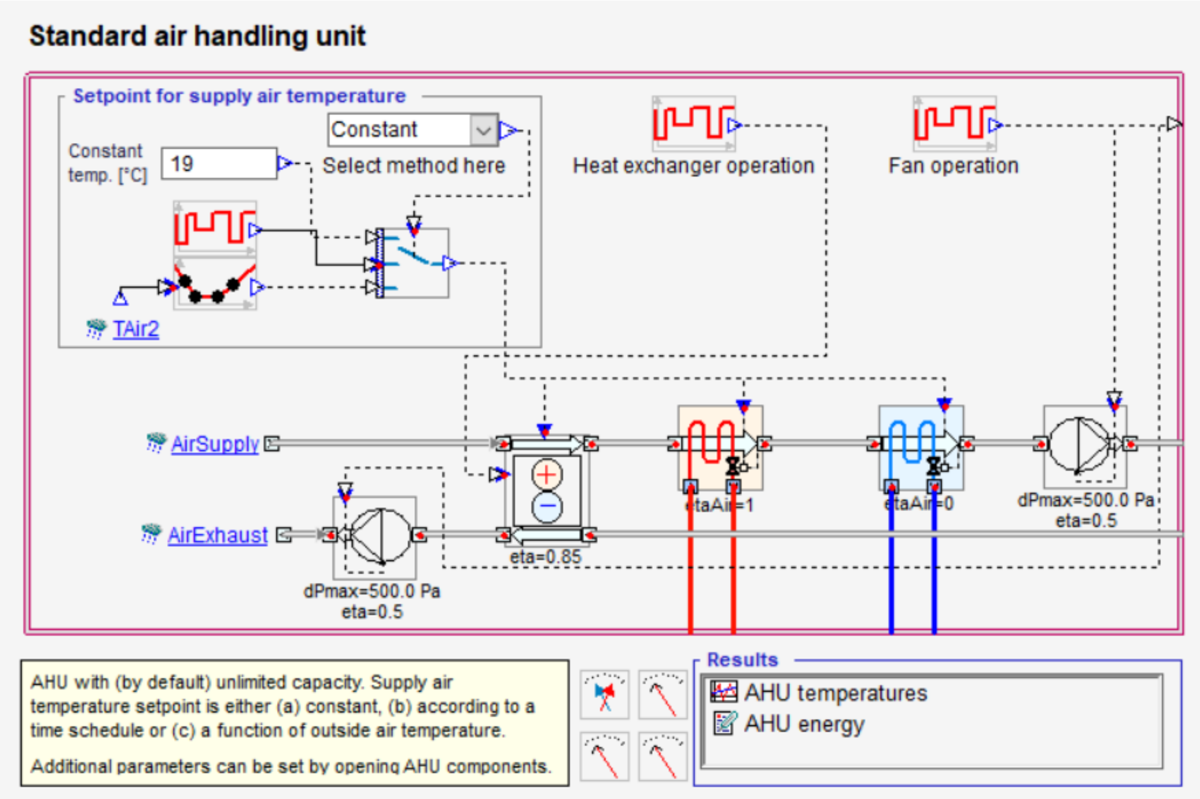


Figure 5.2.: *Illustration of the AHU of the modeled ZEB Laboratory, clipped from the IDA ICE model*

It is equipped with a heat exchanger with an efficiency of $85\,\%$. This heating coil covers the remaining energy demand to achieve a supplied-air temperature of $19\,°\mathrm{C}$ and a fan on both the supply and exhaust side of the system. The fans have a Specific Fan Power (SFP) of $1.0\,\mathrm{kW}/(\mathrm{m}^3/\mathrm{s})$ and operates at $10\,\%$ capacity outside of work hours, $100\,\%$ otherwise. The coiling coil is given efficiency of $0\,\%$ due to there not being any cooling options in the building. The cooling coil is not removed from the AHU due to the interest in simulating with the cooling coil activated. To determine the total cooling capacity of the chosen ventilative cooling strategy.

5.2.4. Expected Occupancy of the ZEB Laboratory

As the ZEB Laboratory is, as of now, not in complete operation, a credible occupancy model has not been determined. Schedule of assumed occupancy are therefore used in the simplified model of the ZEB Laboratory. The floor plans, presented in Section 4.2.2 are used to get an estimate of the maximum occupancy capacity, and user profiles, taken from the project description are used to get an idea of the percentage expected capacity for the different zones [117]. The user profiles are presented in Appendix C. The document specifies assumed occupancy levels for meeting rooms, auditorium, offices, team rooms, cafeteria, and touch-down areas. As the model operates with one zone per floor, some simplifications are therefore necessary. The "as-built" document specifies an estimated heat supplement from occupancy of $4.0\,\mathrm{W}/\mathrm{m}^2$. The occupancy schedules used for the model are made taking these factors into account. To simplify the simulations, the same user profiles for weekdays are used for weekends and holidays.

The ground floor zone is simulated with an assumed maximum occupancy capacity of 74 people, in correlation with the number of seats from the floor plan. The cafeteria is the only usable seating area on this floor, and the user profile for cafeteria/touch-down rooms is selected. The maximum capacity has been reduced to 75 %, as 100 % user capacity is rarely expected and is assumed for every occupancy user profile of this model. This correction has been made for all occupancy schedules of the model. See Figure 5.3 for illustration of profile.

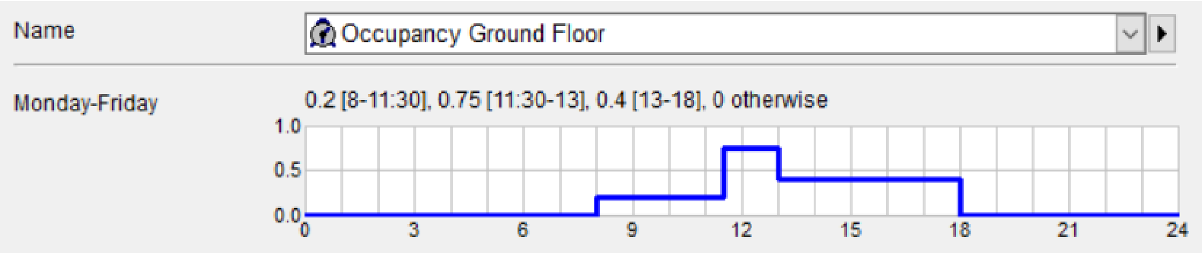


Figure 5.3.: *Occupancy user profile used for the ground floor*

The first and second-floor zone are simulated with an assumed maximum occupancy capacity of 42 and 48 people, respectfully. When determining these values, seated office spaces were counted as one, and the remaining seats were 0,2 occupancy as these seating areas had lower expected usage than the office spaces. These two floors are more complicated to specify a user profile for as the seated areas on the floors are defined with more than one type of zones, thus different user profiles. All profiles have been assessed, and the user profile for office zones has been chosen, as both floors will mainly be used for office work. The floors have a similar usage purpose, and Figure 5.4 is the chosen profile for both zones.

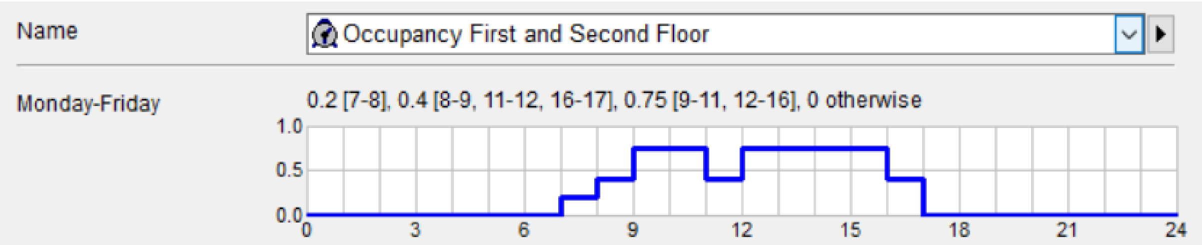


Figure 5.4.: *Occupancy user profile used for the first and second floor*

The third-floor zone is simulated with an assumed maximum occupancy capacity of 45 people. All seats in the auditorium have been accounted for, while the touch-down area outside the auditorium is accounted for as 0,2. Because the auditorium houses a larger part of the assumed occupancy, the user profile for the auditorium is used for the third-floor zone. Figure 5.5 illustrates this profile.

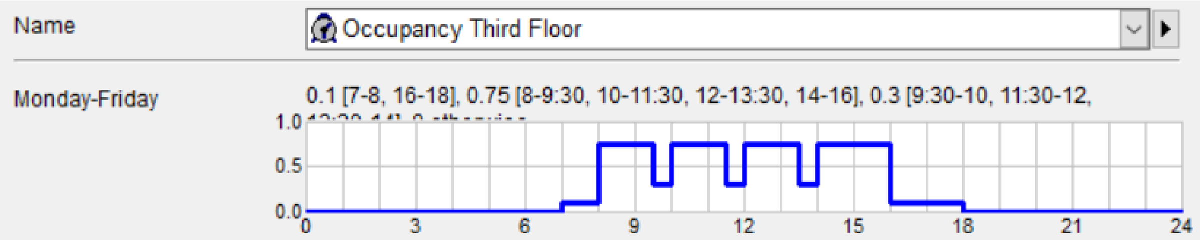


Figure 5.5.: *Occupancy user profile used for the third floor*

The schedules for occupancy presented above are not expected to portray the exact user behavior the occupants of the ZEB Laboratory may provide. Nevertheless, it is seen as a reasonable and adequate schedule with a realistic and varying heating load that is sufficient for this thesis.

5.2.5. Controller for Window Operations

The control algorithm for window operations has been created to utilize natural ventilation for cooling purposes while maintaining a comfortable and healthy indoor environment for all periods of the year. A simple on-off control has been chosen, with the ability to cool the building between chosen temperature setpoints. The setpoints used during the summer season are selected to achieve optimal indoor temperatures in correlation with NS-EN 15251:2007, shown in Figure 2.1, the acceptable indoor temperatures during cooling season for buildings without mechanical cooling systems. The initially selected setpoints for cooling in the summer/cooling season are $24\,^{\circ}\mathrm{C}$ during the day and $26\,^{\circ}\mathrm{C}$ during the night, with the ability to change it depending on the simulated scenario. To minimize operations during occupancy hours, a hysteresis effect has been implemented for the controller. After the indoor air temperature exceeds the setpoint and windows open, the controller will not allow for operation before the temperature falls below $22\,^{\circ}\mathrm{C}$, $2\,^{\circ}\mathrm{C}$ below the setpoint for cooling. For the remaining parts of the year, the setpoint $26\,^{\circ}\mathrm{C}$ and $24\,^{\circ}\mathrm{C}$ were selected to minimize the usage of natural ventilation, to avoid unnecessary draught and discomfort for occupants. For all cases, the $\mathrm{CO_2}$ concentration setpoint is $900\,\mathrm{ppm}$. If the $\mathrm{CO_2}$ concentration exceeds this value, the windows will open regardless of the indoor temperature. The closing setpoint is set to $700\,\mathrm{ppm}$, though the controller may close the window before if the temperature falls below $22\,^{\circ}\mathrm{C}$ or stay open if the temperature is above the temperature setpoint. The control algorithm also defines the degree of opening of the window. The windows on the ZEB Laboratory has a maximum degree of opening of $60\,\%$, and the starting/base value of the following simulation scenarios will be $30\,\%$

The controller operates throughout the entire day, given that all indoor and ambient parameters are within acceptable values. It alternates between day-time and night-time cooling at the time stamps $7\,\mathrm{am}$ and $18\,\mathrm{pm}$. Both temperature differences between indoor and ambient surroundings and ambient wind speed are evaluated to reduce the possibilities of drought and a lowered thermal sensation for occupants. If the ambient temperature falls below $12\,^{\circ}\mathrm{C}$ or the wind speed exceeds $10\,\mathrm{m/s}$, the controller will not allow the window to open. They are seen as acceptable setpoints as both were used in one of the presented studies in Section 3.6.2 and achieved an acceptable indoor environment. Figure 5.6 presents the window control algorithm. A detailed explanation of the controller is given in Appendix D.

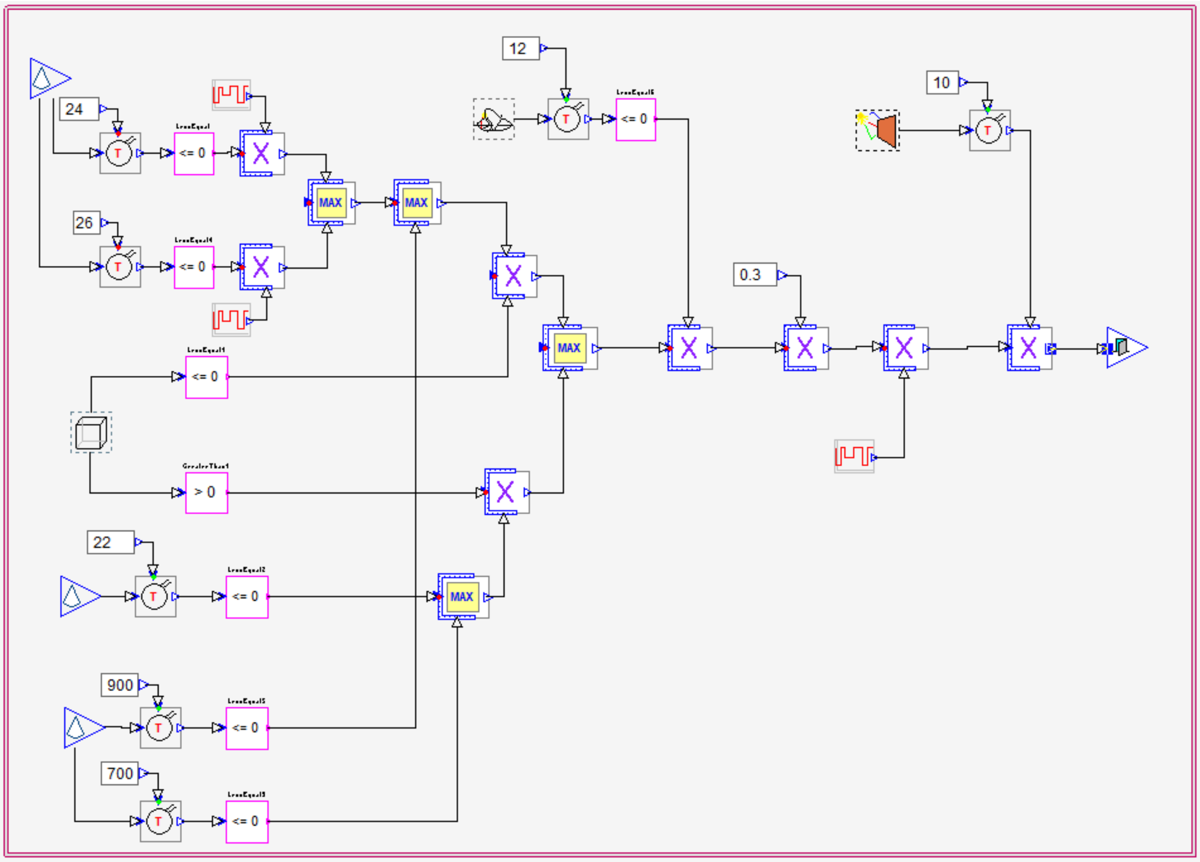


Figure 5.6.: *The created window control algorithm illustrated through visual coding*

5.2.6. Ambient Conditions of the Model

IDA ICE has access to an extensive library of designed data from the ASHRAE International Weather Files for Energy Calculations (IWEC), downloadable from the EQUA Climate Data Download Center. The climate file used for the simulations is made from design data collected at Trondheim Vernes. Of the downloadable files, this was the location closest to the ZEB Laboratory. It is essential that the selected weather file portrays the general climate around the ZEB Laboratory and that the given ambient conditions throughout the year are within realistic proportions. The selected climate files are seen as acceptable for the simulation case.

The pressure coefficient C_p, used for the simulated scenarios is the default values defined as "semi-exposed". The ZEB Laboratory is assumed to be semi-exposed compared to the climate meter at Vernes, where the used weather data is measured.

The as-built document specific a sun-shading factor for all facades of the building. Site shading due to surrounding buildings and other objects can only be implemented through the site object tool in IDA ICE. However, accurate measurements or an imported CAD file are needed. Therefore, the site shading 0,74 / 0,95 / 0,97 / 0,95 (N/E/S/W) are not factored in the simulations. The model will be exposed to a large solar heat gain than in a real-world-scenarios. Although this increase is not expected to be large due to the small site shading values, especially low on the facades exposed directly to the sun.

5.3. Presentation and Evaluation of the Simulated Results

This section explains how the simulation method and results will be presented and evaluated.

The simulated scenarios that will be presented consecutively in Chapter 6 have been conducted to investigate the most optimal way to implement ventilative cooling in the ZEB Laboratory. The results will be evaluated based on thermal comfort, indoor air quality, and energy consumption. For all scenarios, maintaining thermal comfort and a good atmospheric environment within occupancy hours will be prioritized, only looking for energy reduction possibilities while these are achieved. A base case with no implemented ventilative cooling will be used to quantify the decrease in hours of thermal discomfort and energy consumption for the simulated scenarios.

The thermal comfort assessment will be determined by the percentage of total hours where temperatures are outside of the design values for indoor operative temperature for buildings without mechanical cooling, as specified by NS 15251. Results from the floor zones will be evaluated. The presented IDA ICE model specifies no occupancy in the *staircase* zone. In reality, this is not entirely true as occupants stay in the stairwell as they move in between the different floors. Therefore, the results from this zone will be assessed and presented if the resulting values of temperature or thermal comfort become unreasonable as occupants only stay within the staircase zone for short periods of time.

The two main factors that must be assessed to determine the draught risk in the ZEB Laboratory are the general air velocity in the zone and the air velocity of the airflow passing into the building through the openings on the facade. Generally, to avoid thermal discomfort entirely, air velocity surrounding occupants should not exceed $0.3\,\mathrm{m/s}$ during the cooling season. As explained in Section 2.1.4, air velocity's effect on thermal comfort is highly dependent on the occupant's thermal sensation and indoor temperature of the zone. Air velocities may, therefore, if the circumstances allow for it, exceed this value. IDA ICE cannot measure the air velocity at a given point in a zone, making it difficult to evaluate the draught risk an occupant stationed at a given point faces. The draught risk from the supplied air is evaluated by assessing the separation distance of the wall air jet. Most automatically controlled openings on the facade are placed at the height of the suspended panels from the roof of the zone, at a height outside the occupied area. Due to the Coanda effect, the airflow velocity decreases slower and reaches further into the zone than it otherwise would. The Coanda effect is described in the report *Ventilation of Large Spaces in Buildings*, a summary of the work performed in subtask 2 of IEA Annex 26 Energy Efficient Ventilation of Large Enclosures by Heidelberg et al. [77] When air is supplied along a restriction surface, such as a ceiling, the jet will not receive feed air from the side, creating negative pressure between the airflow and the surface. As a result, the air jet will adhere to the restriction surface. At the distance where the air jet separates from the ceiling, the air jet will rapidly descend. The separation distance for a wall air jet from a circular opening can be calculated using Equation (5.1). [63, 77]

$$x_k = K_{sa} \cdot K_a \cdot \sqrt{\frac{v_c^2 \cdot \sqrt{A_c}}{\Delta T_s}} \tag{5.1}$$

The values of $\mathrm{K_{sa}}$ is constant and depends on parameters outside the jet, such as the dimensions of the room and location of thermal load. The $\mathrm{K_a}$ is determined by a ratio between the height and width of the supply opening. Values for K_a used in calculations of this thesis are taken from table 8.14 in the Compendium of TEP4315 and TEP4245 (Subjects at NTNU). Further, v_c is the air velocity at which the airflow is passing through the opening, A_c is the area of the opening, and ΔT_s is the temperature difference between the indoor air temperature and the temperature of the supplied air. From experience, the resulting separation distance should correspond to 50 to $60\,\%$ of the room length to avoid problems with a cold draught from the descending air. The separation distance is highly dependent on Archimedes number. Experience shows that the separation distance corresponds to $Ar_x \approx 0{,}15$.

The atmospheric environment will be evaluated based on the resulting CO_2-concentration of the floor zones. As specified in the window control algorithm and by building category II, CO_2 concentration should be less the $500\,ppm$ above the CO_2 concentration of the ambient surroundings. The age of air will also be evaluated if the resulting values are concerningly high due to the potential of accumulation of other pollutants than CO_2.

More details about the simulated scenarios will be presented consecutively in Chapter 6, in parallel to the results of the conducted simulation.

6. Simulations

This chapter presents the description of the conducted IDA ICE simulation scenarios and their respected results. The purpose of the simulations is to investigate the potential of ventilative cooling in the ZEB Laboratory and determine the most optimal way to implement the strategy. Firstly, a base case scenario with no implemented form of cooling was determined. Second, the ventilative cooling potential was investigated by implementing the presented window control algorithm to the base case model, quantified by decreasing total hours of discomfort and energy-saving for cooling. Further, the possibility of clean natural ventilation of the ZEB Laboratory was investigated. Lastly, based on the presented results of the conducted simulations, new window design suggestions for the ZEB Laboratory were explored. Maintaining thermal comfort and a healthy atmospheric environment for occupants was prioritised.

6.1. The Base Case of the ZEB Laboratory

As mention in Section 5.3, a base case scenario must firstly be determined to quantify the decrease in hours of thermal discomfort and energy consumption for the remaining simulated scenarios. A whole year simulation was conducted on the presented model of the ZEB Laboratory, with no implemented cooling capabilities.

6.1.1. Year Simulation Results

The resulting temperatures of the base case year simulation is illustrated as a carpet plot in Figure 6.1 below. The temperatures from the "Second floor" zone was chosen as a general representation of all zones as temperatures vary minimally between the zones in this scenario.

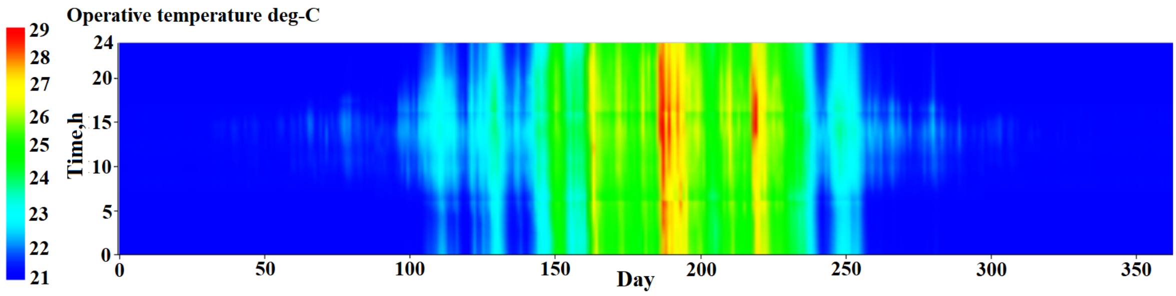


Figure 6.1.: *The recorded temperatures throughout the year of the simulated base case, illustrated as a carpet plot*

Throughout the period between October to early April, the temperatures in all zones never fall below 21 °C and rarely deviates largely from this value. The heating setpoint of the ZEB Laboratory is 21 °C, an indication that there is a constant heating demand. Some periods in April and May show an increase in indoor air temperature occurring during periods where ambient temperatures exceed 10 °C, achieving up to 23 °C in April and 24 °C in May. The period of September shows similar results as of May. During the summer period of June to August, temperatures are generally measured between the values of $23\,^{\circ}\mathrm{C}$ and $26\,^{\circ}\mathrm{C}$, and during periods of extra hot days, up to $28\,^{\circ}\mathrm{C}$ and $29\,^{\circ}\mathrm{C}$. The cooling demand occurs exclusively between June and August. As explained in Section 2.1.3, an indoor temperature of $26\,^{\circ}\mathrm{C}$ or higher is only acceptable for a total of 50 hours during the summer period, not counting periods where the ambient temperature exceeds $24\,^{\circ}\mathrm{C}$, or hours outside of the specified occupancy hours of the zone. The percentage hours of dissatisfaction was not calculated for the staircase zone as there is no specified occupancy. Table 6.1 present the percentage hours where temperatures within occupancy hours were deemed discomforting. The maximum percentage hours of accepted discomfort is $2.26\,\%$, 50 out of the total 2208 hours of the summer months.

Table 6.1.: *Percentage hours of assumed dissatisfaction of all zones in the model*

Zone	Percentage hours of dissatisfaction
Third Floor	$12.70\,\%$
Second Floor	$16.21\,\%$
First Floor	$12.47\,\%$
Ground Floor	$14.99\,\%$

The presented results show that every zone of the model ZEB Laboratory experiences an unacceptable number of hours that induce thermal discomfort for occupants. A form of cooling must be implemented in the building to combat the overheating and achieve acceptable conditions.

The atmospheric environment results of the simulated scenario show a healthy and stable indoor air quality throughout the year. $\mathrm{CO_2}$-levels never exceed $650\,\mathrm{ppm}$, not surprisingly since the mechanical ventilation system was designed to cover the entire air pollution load without assistance from the natural ventilation.

6.1.2. Implementation of Ideal Cooler

An ideal cooler was implemented in the floor zones to quantify the cooling load needed to achieve an acceptable thermal environment under the presented conditions. Three scenarios with different cooling setpoints were simulated to investigate the change in cooling demand depending on the desired indoor temperature. $26\,^{\circ}\mathrm{C}$, $25\,^{\circ}\mathrm{C}$ and $24\,^{\circ}\mathrm{C}$ was used.

Cooling Setpoint - 26 Degrees

Examining the results of this scenario, it was shown that the periods of high temperatures had been reduced greatly. Temperatures do still exceed $26\,^{\circ}\mathrm{C}$ but never $27\,^{\circ}\mathrm{C}$. Figure 6.2 presents the percentage hours of dissatisfaction occurring in the floor zones.

Table 6.2.: *Percentage hours of assumed dissatisfaction in the modelled ZEB Laboratory with the cooling setpoint* 26 °C

Zone	Percentage hours of dissatisfaction
Third Floor	5.89 %
Second Floor	8.42 %
First Floor	5.77 %
Ground Floor	9.33 %

Using 26 °C as the setpoint for cooling gives an unacceptable number of hours of discomfort. However, the periods where temperatures achieved 28 °C and 29 °C have been eliminated, indicating that the implemented cooling has increased the quality of the indoor environment. The total energy used for cooling in the simulated scenario was 503 kWh with a peak demand of 13.05 kW.

Cooling Setpoint - 25 Degrees

When using 25 °C as the setpoint for cooling, no hours of discomfort were recorded, achieving an acceptable thermal environment. The total energy used for cooling in the simulated scenario was 1141 kWh with a peak demand of 14.45 kW. The total cooling demand has more than doubled while the peak demand increased by 1.4 kW.

Cooling Setpoint - 24 Degrees

Even though the setpoint of 25 °C was proven as a sufficient cooling setpoint for achieving an acceptable thermal environment, there was an interest in investigating the cooling demand with the setpoint 24 °C, as indoor temperatures below 24 °C are generally considered a more acceptable indoor temperature during the cooling season. The total energy used for cooling in the simulated scenario was 2258 kWh with a peak demand of 15.49 kW. Showing the same pattern of doubling the cooling demand and a slight increase in the peak power demand of 1.09 kW. Table 6.3 presents a summary of the cooling demand results.

Table 6.3.: *The resulting cooling and peak power demand of the simulated summer period with the cooling setpoints* 26 °C*,* 25 °C *and* 24 °C

Cooling setpoint	Cooling Demand	Peak power demand	Power demand per square meter
26 °C	503 kWh	13.05 kW	0.741 W/m^2
25 °C	1141 kWh	14.45 kW	0.821 W/m^2
24 °C	2258 kWh	15.49 kW	0.880 W/m^2

6.2. Ventilative Cooling of the Modelled ZEB Laboratory

The following section presents and evaluates ways ventilative cooling may be applied to the ZEB Laboratory. Ventilative cooling was implemented as a cooling supplement to the existing mechanical ventilation system in the form of the window control algorithm presented in Section 5.2.5. The cooling potential was evaluated through simulation of the summer period. In addition to the potential thermal discomfort caused by high temperatures, the draught risk from the air jets passing into the building through the automatically controlled windows must be evaluated. Lastly, the model was simulated with clean natural ventilation to investigate the potential fan power reduction.

6.2.1. Implemented Window Control Algorithm

The results from the base case simulation showed that the cooling demand occurs exclusively during the summer months. The modeled ZEB Laboratory was simulated from June through August as presented in the base case scenario, with ventilative cooling implemented.

Figure 6.2 illustrates the recorded temperature in zone "Second Floor" in the form of a carpet plot. The recorded temperatures in all the other zones gave a close to identical profile but had fewer recorded hours with temperatures above 26 °C and therefore, the results illustrated are the most interesting to present. The figure shows that the indoor temperatures throughout the summer period mainly vary between 23 °C and 25 °C, occasionally, exceeding 26 °C. This occurs primarily during periods where outdoor temperatures also exceed 26 °C. The percentage hours of dissatisfaction that occurs in the "Second Floor" zones was 0.22 %, and lower in the remaining zones. The results show that the implementation of ventilative cooling can substantially reduce the temperature peaks that occurred during the hotter days of the summer period.

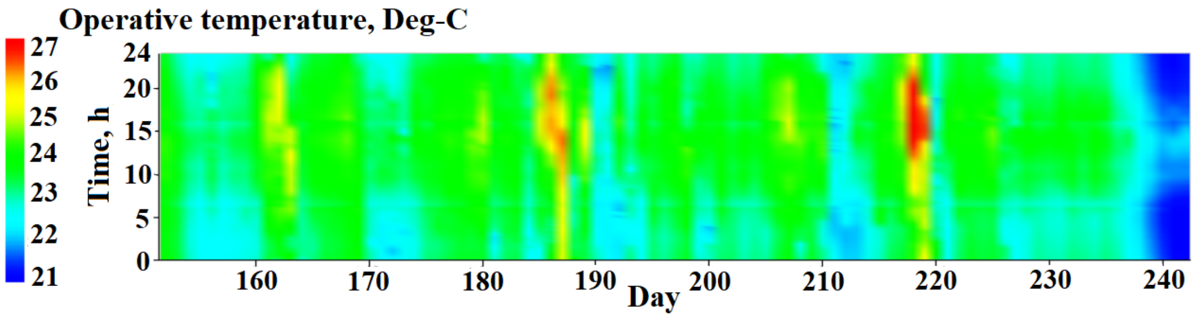


Figure 6.2.: *Carpet plot of the recorded temperatures in zone "Second Floor"*

The resulting CO_2-concentration curve for every zone throughout the simulation period portrays a healthy indoor environment, as CO_2-concentration rarely exceeds 600 ppm and never exceeds 650 ppm, far below 900 ppm.

6.2.2. Draught Risk of Ventilative Cooling

Draught risk is an essential factor in assessing the quality of an indoor environment. Large air velocities or cold descending air from air jets originating from inflow through the automatically controlled windows may considerably affect the occupant's thermal comfort. 7^{th} of August, a day with high ambient temperatures was investigated further. High airflow rates do not necessarily result in an uncomfortable indoor environment, but large air velocities will occur when passing through a small opening. The airflow rate passing through all automatically controlled windows was investigated at $2\,\mathrm{pm}$, at a period where the ambient temperatures were $20\,°\mathrm{C}$ and indoor temperatures were increasing. This results in a temperature difference of $6\,°\mathrm{C}$ between the indoor air and the temperature of the air jet. The Entrance door is not open during this time, and values from $10\,\mathrm{am}$ were therefore evaluated for the Entrance door. The resulting air velocities are expected to be higher as the temperatures in the zones increase further into this day. However, due to the low temperature difference and high indoor temperatures at these times, no draught risk was assumed. Table 6.4 below presents the calculated air velocity passing through all openings on the facades at $2\,\mathrm{pm}$. Positive values equal airflow passing into the building, and negative values equal airflow passing out of the building.

Table 6.4.: *Resulting air velocity through all openings on the building. Positive values equals inflow, negative values equals outflows*

Ground floor	**Airflow rate**$(\mathrm{m^3/s})$	**Opening area**$(\mathrm{m^2})$	**Air velocity**$(\mathrm{m/s})$
East-side window	0,120	0,234	0,513
Entrance door	5,98	0,546	0,091
First floor			
North-side window	0,330	0,927	0,356
East-side window	0,086	0,255	0,337
South-side upper window	0,421	1,416	0,344
South-side lower window	0,365	1,062	0,643
Second floor			
North-side window	- 0,105	0,87	- 0,121
East-side window	- 0,100	0,252	- 0,397
South-side window	- 0,210	1,164	- 0,180
Third floor			
North-side upper window	- 0,471	0,984	- 0,48
North-side lower window	- 0,217	0,546	- 0,397
West-side window	- 0,107	0,216	- 0,495
East-side window	0,100	0,255	0,392
Staircase			
Ground floor opening	3,348	43,95	0,076
First floor opening	4,427	37,05	0,120
Second floor opening	3,686	44,06	0,084
Third floor opening	5,130	50,28	0,102

The resulting air velocities can then be used to calculate the separation distance and, further, the percentage penetration length of the flow. Equation (5.1) presented in Section 5.3 is used. The heat sources are assumed to be evenly distributed over the area, and the room is assumed large enough to set K_{sa} = 1,5. The K_a coefficient is chosen from a table presented in the Compendium for TEP4315 and TEP4245, determined by the ratio between the width and height of the opening [63]. The room length is specified by the plan drawings from the project description as the length from the opening to the closest parallel wall. As explained in Chapter 5.2.2, all windows of the same type have been merged for simplification and reduction of computational time. In IDA ICE, all windows on the same facade connected to the same zone are affected by the same forces. Therefore, the calculated air velocity through the merged version of the window is assumed to equal the air velocity achieved through every individual window.

The airflow moving out of the building poses no risk of causing discomfort for the occupants, as no air jets occur in or above the occupied area. Therefore, openings where this occurred, was not investigated further. The "Entrance door" and the "South-side lower window" supply air at occupancy height, meaning the presented calculation method is not applicable. Because the air velocity through the "Entrance door" is low and occupants do not spend considerable time in this area, the draught risk is assumed negligible. The "South-side lower window", on the other hand, will be closed in further simulations. Supplying air at such a large air velocity directly in an occupied area is unacceptable and may result in a large amount of discomfort for occupants. Closing this window results in a new pressure balance on the building and a higher air velocity through the "south-side upper window". The new air velocity through this window is $0.40\,\mathrm{m/s}$. No considerable change on the other windows occurred. The airflow passing between the staircase zone and the floor zones was investigated, and the resulting air velocities fall well within acceptable values. In addition, occupants do not stay in this zone for a considerable amount of time. The resulting percentage penetration length of the supplied airflow for the remaining windows is presented in Table 6.5.

Table 6.5.: *Separation distance and the resulting percentage penetration length for all air jets passing into the building through the openings on the facades*

Ground floor	K_a	**Separation Distance**	**Room length**	**Penetration**
East-side window	8,5	1.86 m	4.873 m	38.2 %
First floor				
North-side window	7,8	1.67 m	4.073 m	41.0 %
East-side window	8,7	1.28 m	4.873 m	26.3 %
South-side window	8,5	1.49 m	4.073 m	36.7 %
Third floor				
West-side window	8,5	0.377 m	3 m	12.6 %

All calculated separation lengths fall below the desired value of 50 to 60 % penetration. The window on the east side of the ground floor supplies air to an unoccupied area. Therefore, unless the separation distance is unreasonably short and the temperature difference is large, there is no concern of drought risk. The North and South-side windows supply air directly into the occupied office spaces. These areas are seen as the most critical as they are occupied for the longest period of all the different rooms in the building. According to the presented separation length, there is a risk of drought in these areas. The same goes for the East-side window on the first floor and the west-side window on the third floor. The "touch-down" area that the east-side window supplies air directly into is substantially less occupied and is seen as a less critical area. However, the penetration length is low, and a high-temperature difference between the ambient and indoor conditions may result in discomfort. The penetration length of the west-side window is low as well. However, it can be argued that the placement and the resulting direction of the air jet and the seating plan of the auditorium result in the air jet descending away from the occupants and therefore not resulting in any draught risk. However, this is just speculation.

Though the separation distance indicates a draught risk in all of the presented zone, the temperature difference between the indoor and air-jet flow is low. In addition to the indoor temperature being generally high, it can be argued that the draught would be, in most cases, experience as a pleasant breeze rather than the draught.

6.2.3. Night-time Ventilative Cooling of Mechanically Ventilated ZEB Laboratory

In the literature review on ventilative cooling, night-time cooling in Northern Europe was established as a strategy with high cooling penitential. Night-time ventilative cooling of the ZEB Laboratory was investigated by simulating an "average" and a "hot day". The "average day" was a chosen day of the summer period that provided temperatures throughout the day did not deviate largely from normal summer temperatures, and a "hot day" was a day with higher than average temperatures. Temperatures reach up to 17 °C during the "average day" and up to 27 °C during the "hot day". Both scenarios were simulated with and without night-time cooling to showcase the potential of night-time ventilative cooling potential. The Figures 6.3 to 6.6 presents the results for the simulation scenarios. Night-time cooling period is specified as the period outside of occupancy, 6 pm to 7 am. The setpoint of cooling used for both day and night-time ventilative cooling was 24 °C.

Mechanical Ventilation without Night Cooling, Average Day

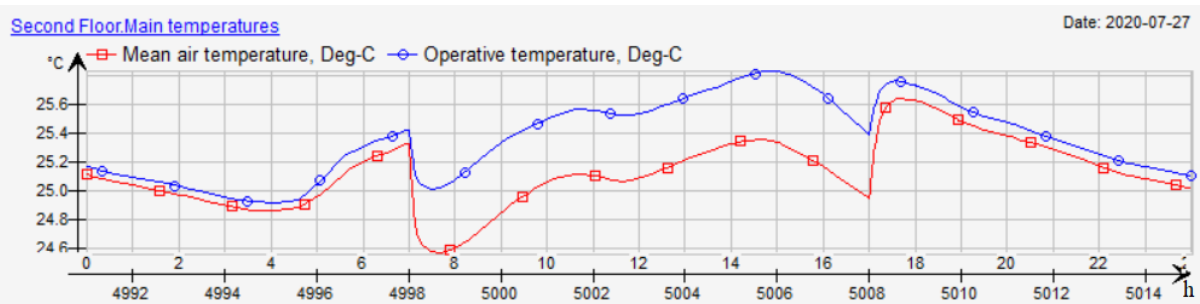


Figure 6.3.: *Resulting indoor temperatures in the model simulated with mechanical ventilation and no night-time cooling, on the 27^{th} of June , a "average day"*

Mechanical Ventilation with Night Cooling, Average Day

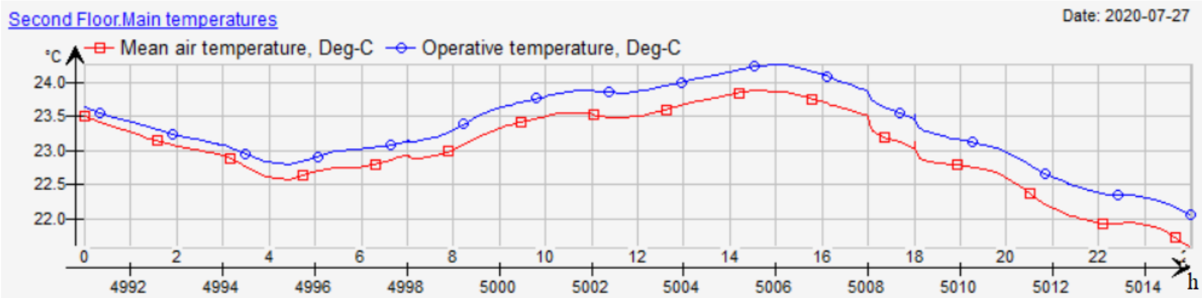


Figure 6.4.: *Resulting indoor temperatures in the model simulated with mechanical ventilation and night-time cooling, on the 27 th of June , a "average day"*

Mechanical Ventilation without Night Cooling, Hot Day

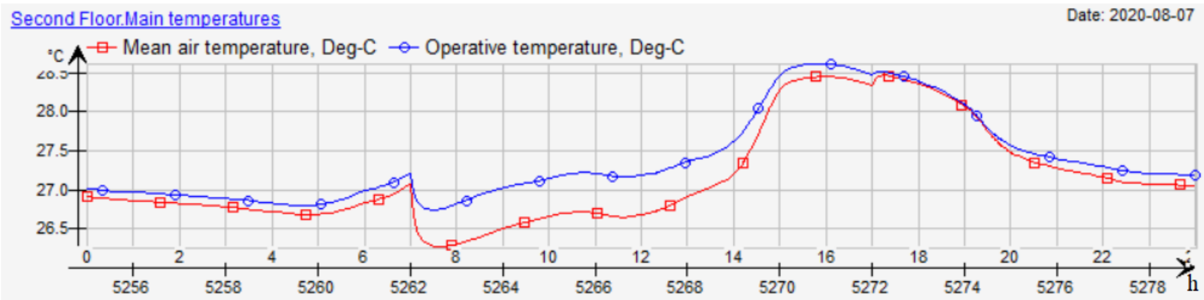


Figure 6.5.: *Resulting indoor temperatures in the model simulated with mechanical ventilation and no night-time cooling, on the 27 th of June , a "hot day"*

Mechanical Ventilation with Night Cooling, Hot Day

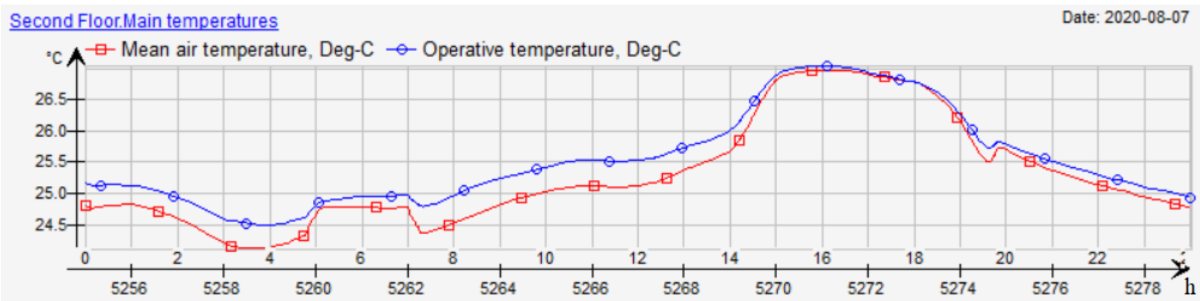


Figure 6.6.: *Resulting indoor temperatures in the model simulated with mechanical ventilation and night-time cooling, on the 27 th of June , a "hot day"*

Figure 6.3 and 6.4, 6.5 and 6.6 show that night-time ventilative cooling helps reduce the general and peak temperatures of both simulated days. All simulated scenarios achieve temperatures that are within the standards of buildings without implemented cooling. However, the resulting temperatures when night-time cooling was utilize were lower and generally considered more satisfactory for occupants. The night-time cooling strategy is therefore an encouraged strategy but not a necessity.

6.2.4. Natural Ventilation of the ZEB Laboratory

To further investigate the energy reduction potential of the ZEB Laboratory, the model was simulated without mechanical ventilation. How much can the fan power requirement be reduced without affecting the thermal or atmospheric environment of the building?

Figure 6.7 below presents a carpet plot of the recorded temperatures in the "Second Floor" zone when the model was simulated with only natural ventilation. Much like the temperature results presented in Section 6.2, all resulting temperatures of the other zones portray a similar plot. The results from the second floor zone are considered an acceptable representation of all the other zones and will be discussed further.

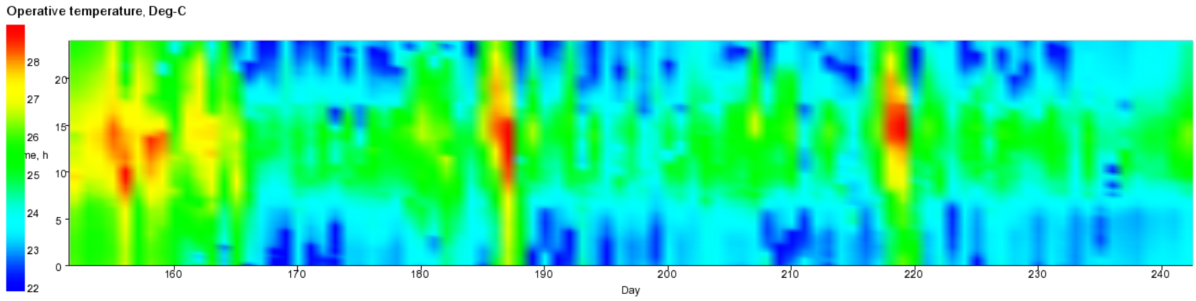


Figure 6.7.: *Carpet plot of the recorded temperatures of Second Floor zone. Simulation conducted with clean natural ventilation*

Resulting temperatures generally fluctuate between $23\,°C$ and $25\,°C$ throughout the simulation. They rarely exceed $26\,°C$ but a considerable period of high temperatures occurs during most of June. The percentage hours of dissatisfaction achieved was $13.52\,\%$ on the second floor zone. The remaining zone also achieved values over the maximum dissatisfaction level $2.26\,\%$.

The general CO_2 concentration recorded throughout the simulated period depicts a healthy indoor environment. The CO_2 levels rarely exceed $900\,ppm$ for most of the simulation time, but due to a predetermined setpoint in the window control algorithm, there are periods without ventilation. This occurs when ambient temperatures fall below $12\,°C$ or wind velocities exceed $10\,m/s$. CO_2 levels and temperatures rapidly increase and may reach levels as high as $2000\,ppm$ and $29\,°C$. The model is especially vulnerable to high CO_2 concentrations in early June and late August, where ambient temperatures are average below $10\,°C$. This shows that ventilating exclusively with natural ventilation under the presented conditions is an unacceptable solution.

To avoid an accumulation of CO_2 during periods where ambient temperatures fall below $12\,°C$. A mechanical ventilation control algorithm was created, presented in Figure 6.8 below.

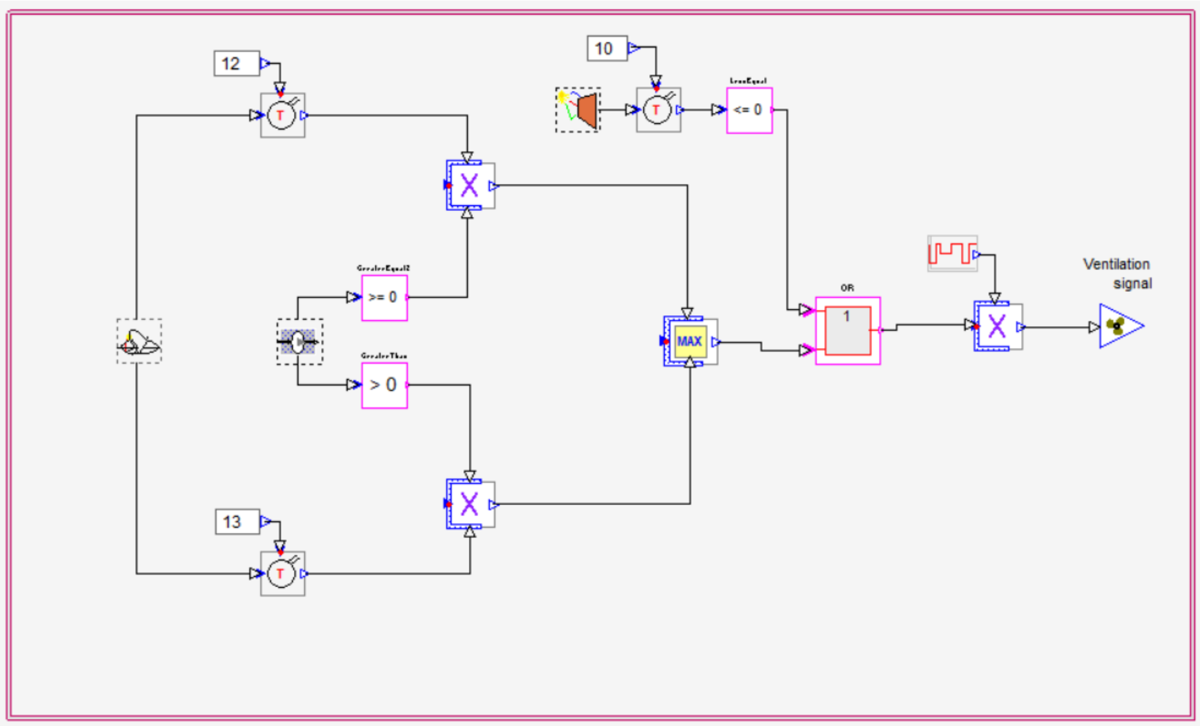
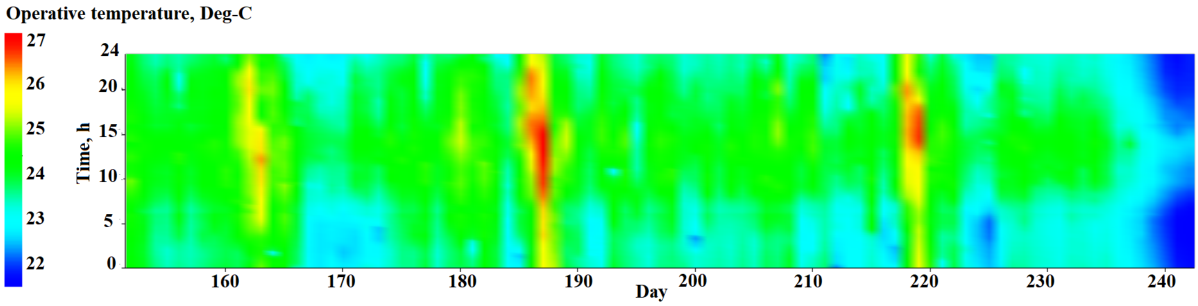


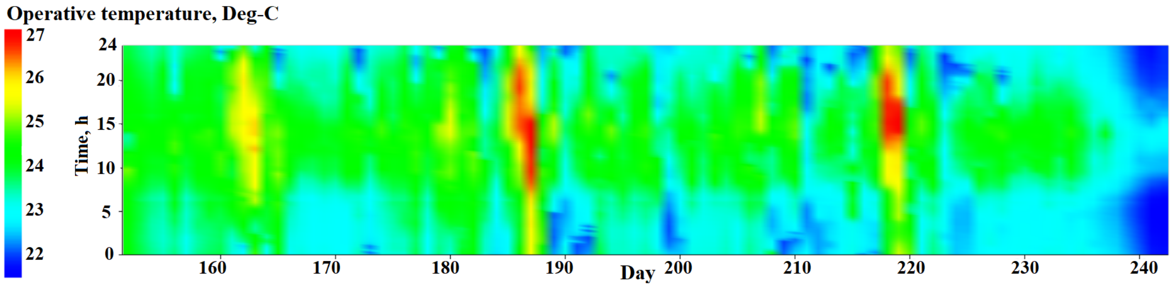
Figure 6.8.: *The mechanical ventilation control algorithm for avoidance of CO_2 accumulation and temperature increase in periods with unacceptable conditions for ventilative cooling*

When ambient temperatures fall below $12\,°C$, the mechanical ventilation turns on. A hysteresis effect was implemented to avoid the potential of a large number of operations over a short period. Once the mechanical ventilation turns on, it will not turn off before the ambient temperatures exceed $13\,°C$. A detailed explanation of the presented mechanical ventilation control algorithm is given in Appendix E.

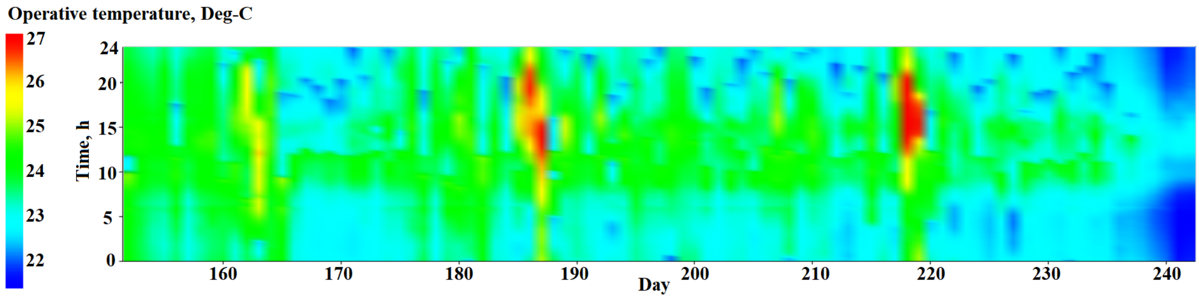
After implementing the mechanical ventilation control algorithm, the periods with high accumulation of CO_2 were removed. The remaining parts of the simulated period were unaffected, and the CO_2 concentrations now vary between 600 to $900\,\text{ppm}$ for the entire simulation. The occurrence of high indoor temperatures were removed and the percentage hours of dissatisfaction was lowered below $1\,\%$ for every zone. Figure 6.9 presents carpet plots of the recorded temperatures in all of the building zones through the summer season. All zones are presented to showcase the different temperature variations in the zones.



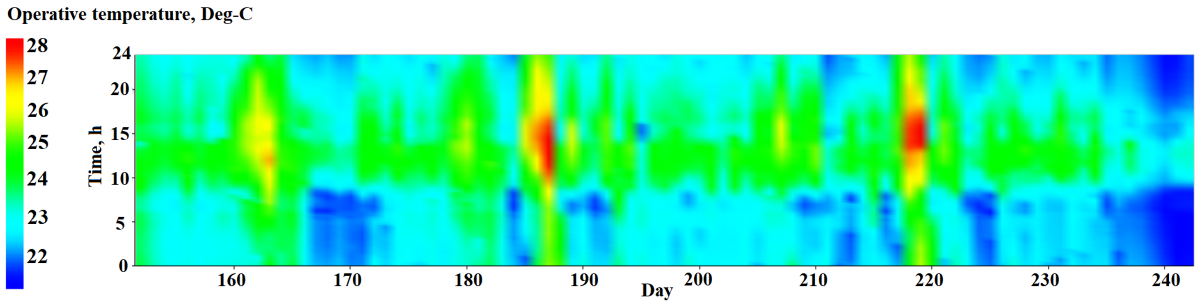
(a) *Carpet plot of the recorded temperatures in the Third floor zone*



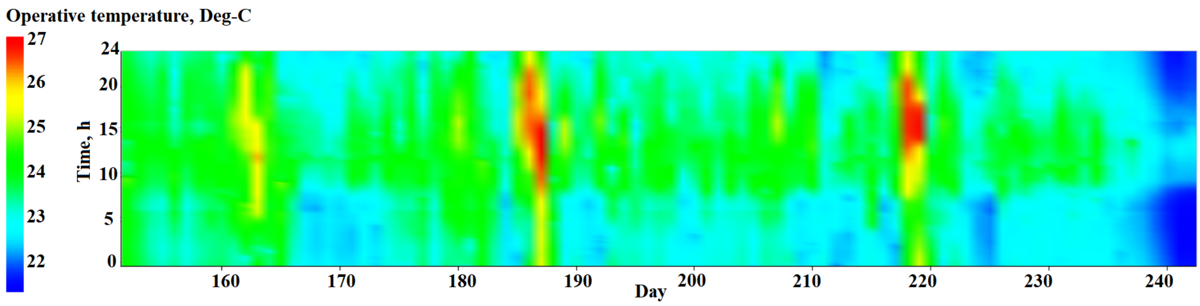
(b) *Carpet plot of the recorded temperatures in the Second floor zone*



(c) *Carpet plot of the recorded temperatures in the First floor zone*



(d) *Carpet plot of the recorded temperatures in the Ground floor zone*



(e) *Carpet plot of the recorded temperatures in the Staircase zone*

Figure 6.9.: *Carpet plot of the recorded temperatures in the modeled ZEB Laboratory of the simulated scenario. Simulation conducted with ventilative cooling and the presented mechanical ventilation control algorithm*

The carpet plots presented in Figure 6.9 illustrate that the achieved temperatures in the different zones increase with the floor level. The peak temperatures do not vary much, but the total period with temperatures above $24\,°\mathrm{C}$ increase with the elevation height of the floors, shown by the total area of green on the carpet plots. This is explained by the buoyancy effect as presented in Section 3.2.2. Hot air rises above cooler air, which will naturally cause an accumulation of heat at higher levels. Peak temperatures do not vary much between the different zones other than the ground floor, where the maximum temperatures is a degree higher than the rest. The air exchange rate of the ground floor may not be sufficient in removing heat during extra hot periods.

With the implemented mechanical ventilation control algorithm, $68\,\%$ of hours where mechanical ventilation was used was replaced with passive cooling. The base case scenario shows that $5.51\,\mathrm{kWh}$ fan power use is required during the summer months. With the implemented solution, the energy demand was reduced to $1.93\,\mathrm{kWh}$, a reduction of $3.58\,\mathrm{kWh}$.

6.2.5. Night-time Ventilative Cooling of Naturally Ventilated ZEB Laboratory

The potential of night-time ventilative cooling was also investigated for the naturally ventilated scenario. The same "average day" and "hot day" were investigated. Both scenarios were simulated with and without night-time cooling. The Figures 6.10 to 6.13 presents the results of the simulated scenario.

Natural Ventilation without Night Cooling, Average Day

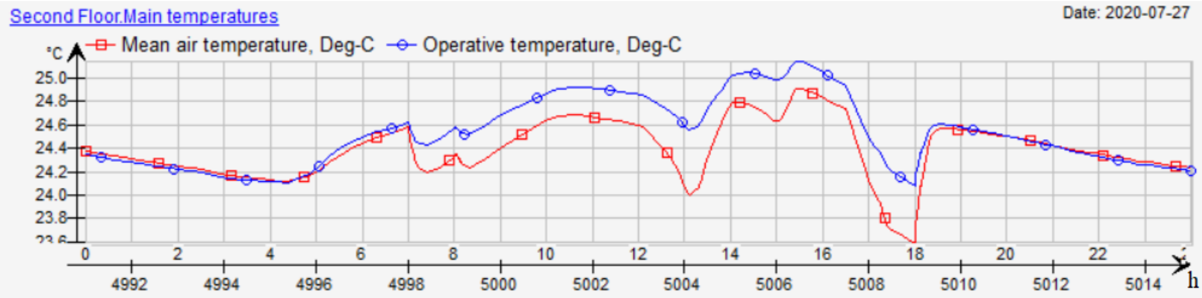


Figure 6.10.: *Resulting indoor temperatures in the model simulated with natural ventilation and no night-time cooling, on the 27 th of June , a "average day"*

Natural Ventilation with Night Cooling, Average Day

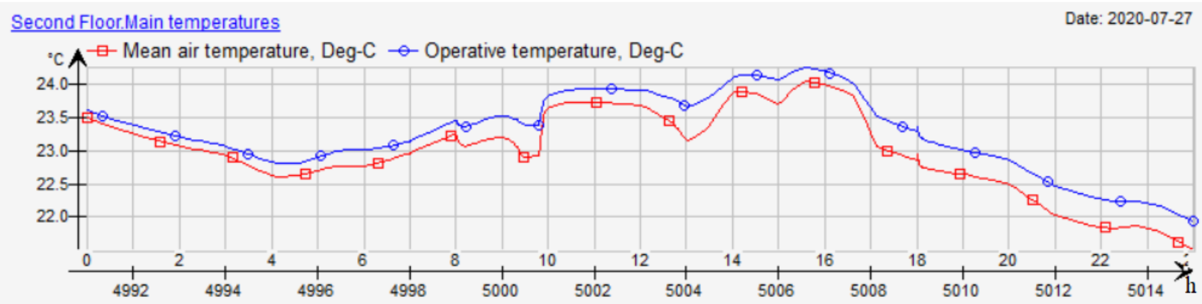


Figure 6.11.: *Resulting indoor temperatures in the model simulated with natural ventilation and night-time cooling, on the 27 th of June , a "average day"*

Natural Ventilation without Night Cooling, Hot Day

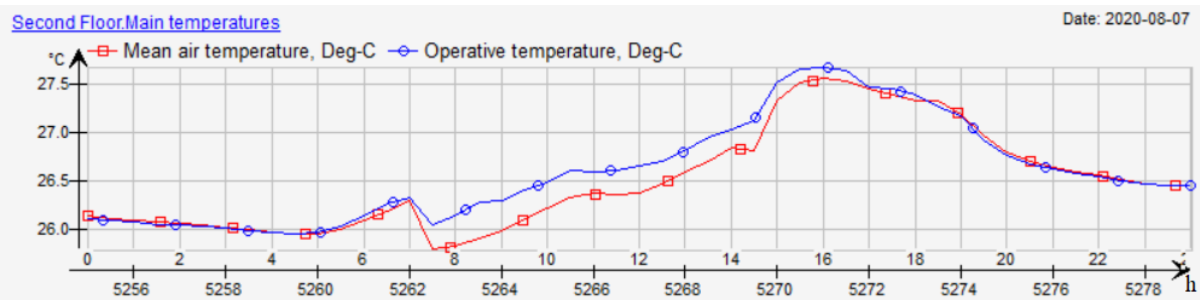


Figure 6.12.: *Resulting indoor temperatures in the model simulated with natural ventilation and no night-time cooling, on the 27 th of June , a "hot day"*

Natural Ventilation with Night Cooling, Hot Day

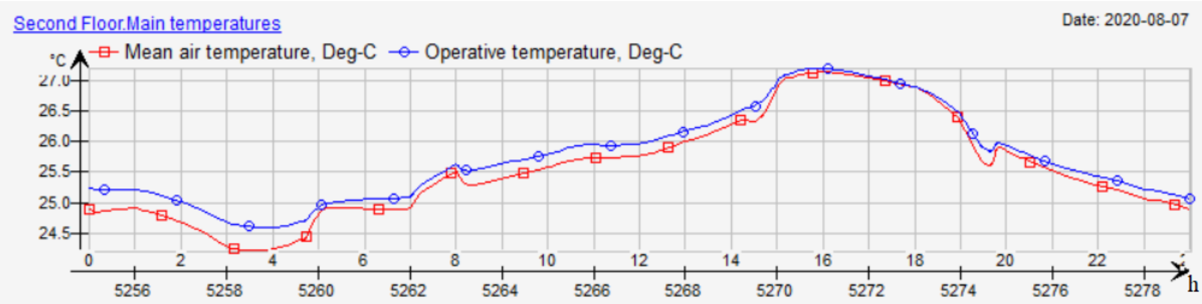


Figure 6.13.: *Resulting indoor temperatures in the model simulated with natural ventilation and night-time cooling, on the* 27 th *of June , a "hot day"*

The temperatures of the simulated scenarios, presented in Figure 6.10, 6.11, 6.12 and 6.13 are all within the standards for buildings without cooling. However, scenarios with night-time cooling achieve lower and generally more acceptable indoor temperatures. "Average day" scenario rarely exceed 24 °C and the temperature during the "hot day" exceeds 26 °C for a considerably smaller period than the scenario without night-time cooling. Night-time cooling is also recommended when naturally ventilating the ZEB Laboratory but the strategy is not necessary to achieve an acceptable indoor environment.

6.3. Optimization of the Natural Ventilation Solution

The presented results show that a healthy atmospheric and thermal environment could be achieved with both the mechanical ventilation system with ventilative cooling through window operation and natural ventilation with mechanical supplement. This section investigates the optimization potential of natural ventilation of the ZEB Laboratory.

6.3.1. Temperature Setpoint for Ventilative Cooling

Three temperature setpoints for ventilative cooling were simulated, in addition to the setpoint of 24 °C that have been used as the setpoint for earlier simulations. The setpoints of 26 °C, 24 °C, 23 °C and 22 °C were investigated further. Two different days were simulated. The 27th of July and the 7th of August. The results are presented in Table 6.6 as the percentage hours recorded above 24 °C as the temperature below is generally seen as a comfortable indoor temperature during the cooling season. The model was simulated with the natural ventilation strategy with mechanical support but the results are seen as relevant and applicable to the mechanical ventilated strategy as well.

Table 6.6.: *The percentage hours of operative temperature recorded above* 24 °C *with the different presented ventilative cooling setpoints for a average and hot summer day*

Setpoint Temperature	27^{th} **of July** **Percentage hours above** 24 °C	7^{th} **of August** **Percentage hours above** 24 °C
Third floor		
26 °C	100%	100%
24 °C	35,6%	100%
23 °C	0%	94,5%
22 °C	0%	95,4%
Second floor		
26 °C	100%	100%
24 °C	19,2%	100%
23 °C	7,7%	94,9%
22 °C	0%	95,9%
First floor		
26 °C	100%	100%
24 °C	22,6%	84,1%
23 °C	10,0%	70,5%
22 °C	0%	70,2%
Ground floor		
26 °C	86,1%	100%
24 °C	32,5%	100%
23 °C	21,8%	94,5%
22 °C	0%	94,5%

The results from the 27^{th} of July simulation scenario show that the setpoint temperature for ventilative cooling largely affects the resulting indoor temperature of the zones, which was also discussed in Section 3.5.1 The use of the setpoint 26 °C was unable to keep temperatures below 24 °C. Temperatures varies between 24.5 °C and up to 26 °C. Only slightly exceeding 26 °C for two hours on the second floor. The natural ventilation is often utilized due to the CO_2 concentration exceeding 900 ppm. Reducing the setpoint to 24 °C or 23 °C drastically reduces the percentage hours recorded above 24 °C. Going from 100% down to 7,7% hours recorded above 24 °C in the Second floor zone. Using the setpoint 22 °C resulted in no registered temperatures above 24 °C.

The quality of the indoor environment of all the scenarios simulated for the 27^{th} of July is considered acceptable, as the CO_2 concentration and the percentage hours of discomfort are within an acceptable level. Though this is the case, keeping the temperature below 24 °C will generally provide more comfortable conditions for the occupants during the cooling season. Using a setpoint lower than 23 °C provides the best results.

The results from the 7^{th} of August scenarios show that the chosen setpoint of ventilative cooling does not affect the resulting temperatures for hot days a great deal. Reducing the temperature setpoint for cooling from 26 °C to 23 °C and 22 °C only reduces the percentage hours a small percentage. This is explained by the high ambient temperature that exceeds 27 °C on this day. However, an increase in the total period the windows are open may increase the air movement within the zone, potentially providing a chilling effect and comfort through air movement.

6.3.2. Optimization of Mechanical Ventilation

By examining the previously presented results, it was discovered that short periods of mechanical ventilation occur during some periods of the day. Figure 6.14 presents the supplied and extracted airflow rate from both natural and mechanical ventilation of the second floor zone on the 27th of June, an "average day". The discussed results is marked with a black circle on the figure.

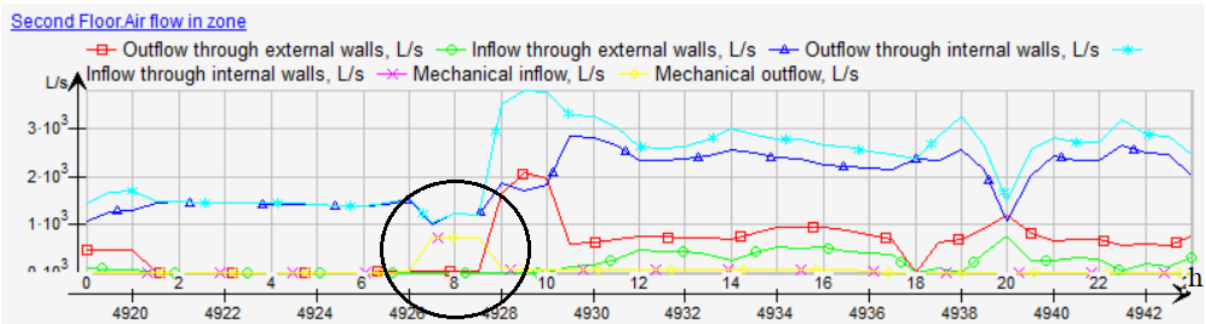


Figure 6.14.: *Supplied and extracted airflow rates from natural and mechanical ventilation of the second floor zone on the 27th of June. Use of mechanical ventilation marked in the black circle*

The mechanical ventilation turns on for a short period due to a slight dip in ambient temperature, and this may also occur in short periods of recorded wind velocity above 10 m/s. The low CO_2 accumulation during the short period where ventilative cooling is inoperable does not justify the need for mechanical ventilation, indicating that the energy use can be reduced further using a more optimal or detailed control algorithm which could evaluate the future conditions and make an informed decision if mechanical ventilation is required.

6.3.3. Window Opening Percentage

The windows used of the ZEB Laboratory have a maximum opening percentage of 60 %. The previously presented simulations have in all scenarios operated with a 30 % opening percentage. It is desired to investigate the effects different opening percentages have on the thermal and atmospheric environment. The building was simulated with natural ventilation and the mechanical ventilation control algorithm with the window opening percentages of 60 %, 30 %, 15 % and 5 %. The 27th of June was the chosen simulation period. The quality of the thermal and atmospheric environment was evaluated after the resulting temperature, CO_2 concentration, and the potential drought risk in the office space area of the Second floor zone. The temperature measurements are presented in Figure 6.15, the CO_2 measurements are presented in Figure 6.16, air velocity is presented in Figure 6.17, and the percentage penetration length of the incoming airflow are presented in Figure 6.18.

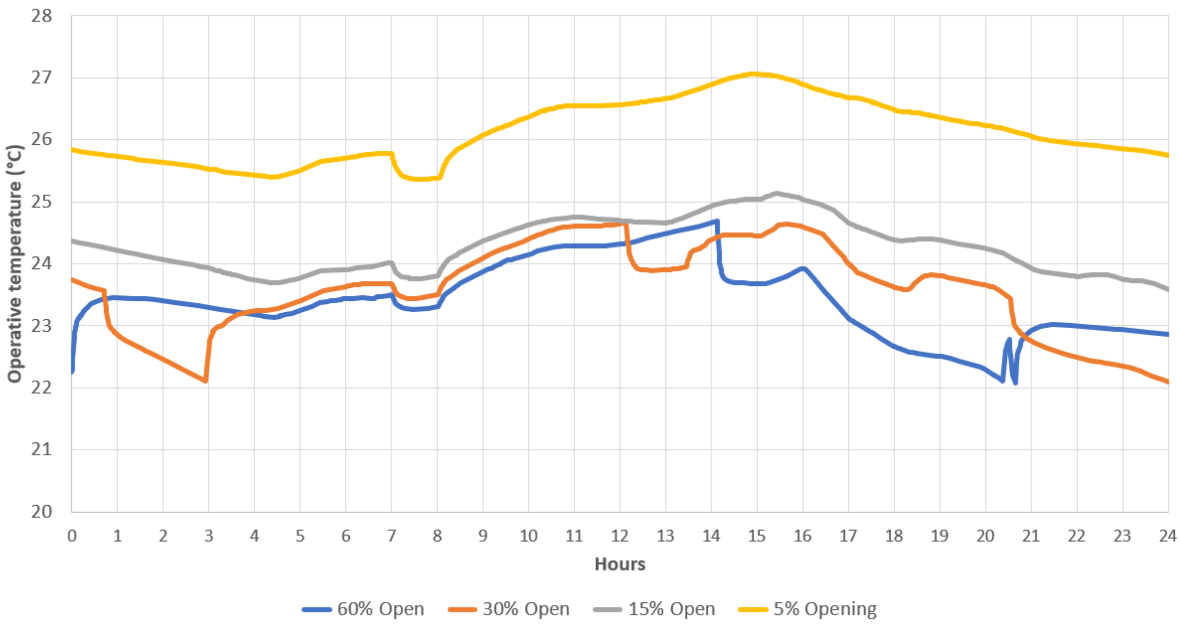


Figure 6.15.: *The resulting temperature measurements on the* 27^{th} *of June with the opening percentages* $60\,\%$*,* $30\,\%$*,* $15\,\%$ *and* $5\,\%$

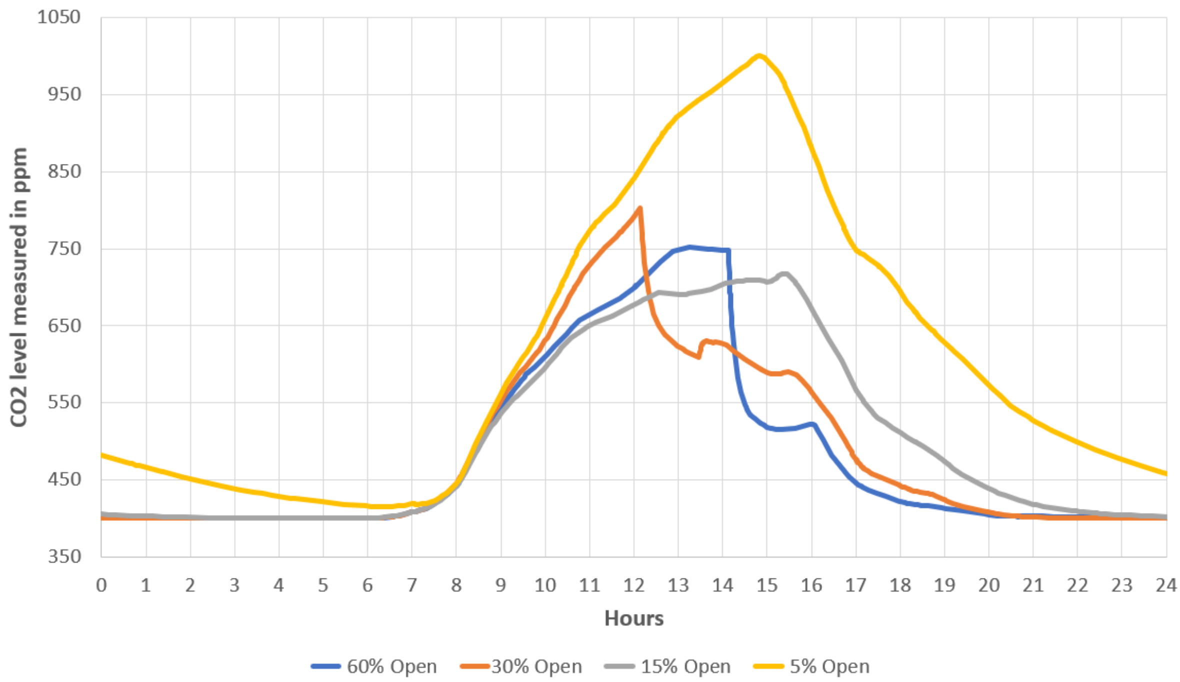


Figure 6.16.: *The resulting* CO_2 *levels on the* 27^{th} *of June with the opening percentages* $60\,\%$*,* $30\,\%$*,* $15\,\%$ *and* $5\,\%$

The resulting temperature and CO_2 measurements show that the opening percentage of the windows does not affect the thermal or the atmospheric environment largely. The indoor temperature of the zone increases with the reduction of opening percentage, indicating that a higher air exchange rate is achieved with increasing opening percentage. The CO_2 level follows the same indication, showing a slight decrease in concentration with decreasing opening percentage. Interestingly, the resulting CO_2 concentration of the 30 % opening scenario was higher than that of the 15 % and 60 % opening scenario, although not substantially. The results of the different scenarios do not deviate mainly from one another except for the 5 % opening scenario, where both the temperature and CO_2 concentration exceeds the acceptable value of 26 °C and 900 ppm. At a certain point, it is assumed that the delivered airflow rate from the windows becomes too low to combat the accumulation of air pollutants, and both CO_2 concentrations and temperatures increase.

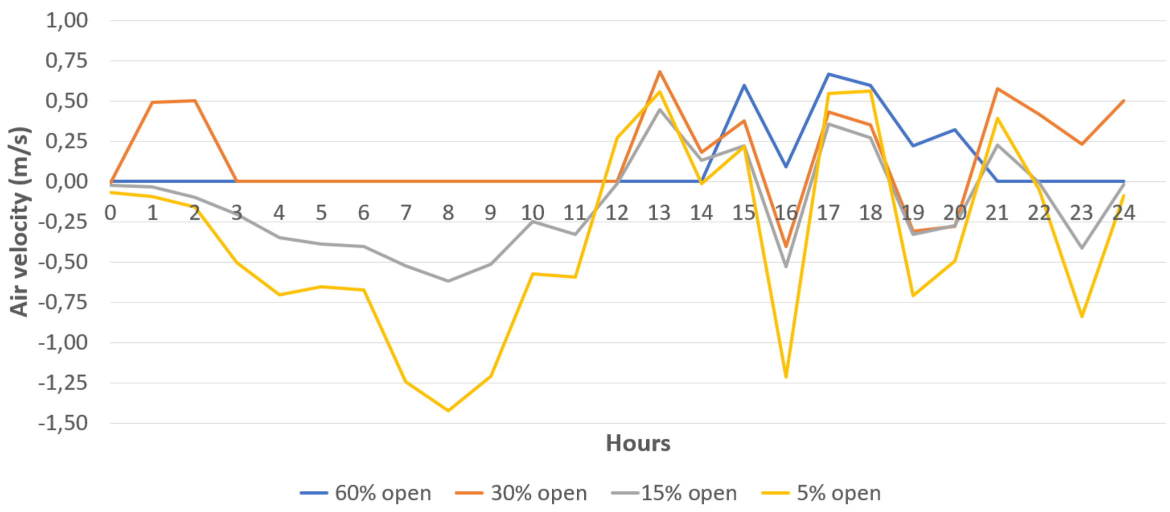


Figure 6.17.: *The resulting air velocity through the south side window on the second floor. The* 27^{th} *of June was simulated with the window opening percentages* 60 %, 30 %, 15 % *and* 5 %

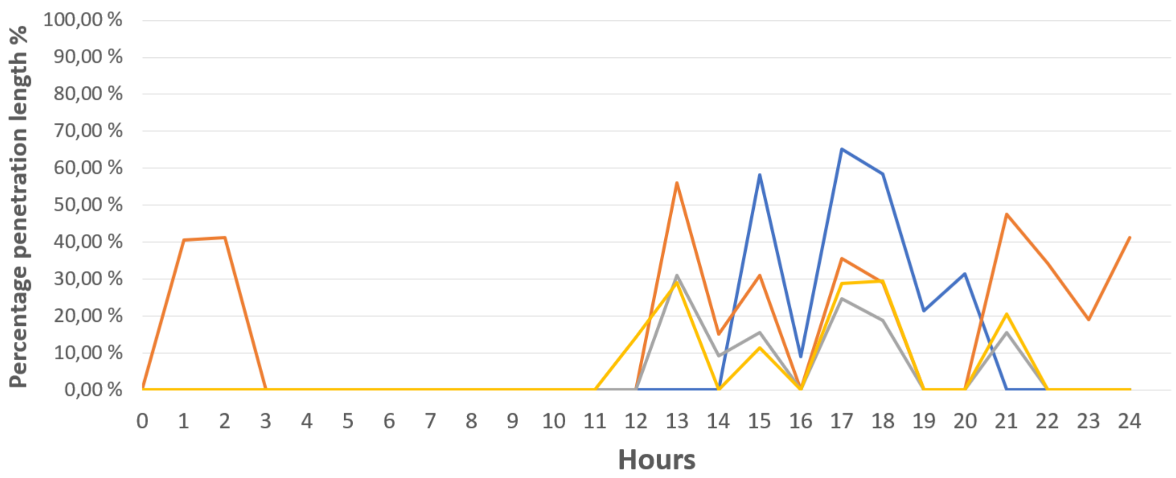


Figure 6.18.: *The resulting percentage penetration length of the airflow through the south side window on the second floor. The* 27^{th} *of June was simulated with the window opening percentages* 60 %, 30 %, 15 % *and* 5 %

The resulting air velocities from the simulated scenarios illustrate a sensitive pressure profile over the model. The different elevation of the zones, the temperature changes, and the number of windows implemented on the facades result in a complex network of factors that affect the airflow through the facades and within the building. A temperature change or an increase in airflow rate through a specific window may have unpredictable effects on the airflow rate or direction through another. This is depicted in the resulting air velocities as they change in intensity and direction throughout the day. With $60\,\%$ opening percentage, the air flows exclusively into the building on the second floor. Inspecting the airflow of the building in its entirety shows that minimal air exchange occurs through the facade on the first and second floor, indicating that a large opening percentage results in a pressure drop on the building. The flow is mainly exchanged between the ground and third floor, indicating a potential short circuit occurring. The resulting temperatures and CO_2 concentrations on the first and second floor do not comply with this theory as both are within the acceptable level. However, in a real-world scenario, a steady rise in temperature and CO_2 level would occur with no air exchange in a zone. However, this is not expected to happen in a real-world scenario as increased temperatures in a zone due to the buoyancy effect would encourage opening of windows flow in the building.

The remaining opening scenarios show results with a more considerable degree of disorder. The air velocities frequently change, both in size and direction, for all simulated opening percentages. They change with varying degree, smaller opening percentages generally resulting in more considerable changes, further depicting the airflow pattern in the building as sensitive, complex, and therefore hard to predict and control.

From the results presented in the graphs above, the $60\,\%$ opening scenario seems to be the solution that results in the best thermal and atmospheric environment. When examining the results from the first floor zone, temperatures and CO_2 are within acceptable values, but there is no airflow passing through the facade even though the windows are open. An indication that the airflow between the ground and third floor is large enough to cover the airflow rate demand of the first floor zone as well. The resulting airflow rates through the day on the second floor zone are limited to the period between $2\,\mathrm{pm}$ and $9\,\mathrm{pm}$. A short circuit on the first and second floors is speculated to have occurred in this modeled scenario. It is assumed that this will not happen in a real-life situation as the inevitable increase in zone temperature would eventually create a pressure difference that would force airflow. However, it does indicate that $60\,\%$ opening on windows may result in an unnecessarily high airflow rate. The $15\,\%$ and $5\,\%$ opening scenarios result is achieved temperatures above $24\,^{\circ}\mathrm{C}$. Therefore, a $30\,\%$ opening percentage is seen as the most acceptable solution based on the presented result.

6.3.4. Realistic Weather File

Every simulation conducted thus far has used a standard weather file downloaded from the EQUAs database. The weather file is not made from recorded values but constructed to portray a potential year with realistic temperature peaks and slopes. To investigate the potential sensitivity of the model, a weather file with recorded climate data from 2020, taken from a measuring device on the roof of the "Varmeteknisk laboratorium" at Gloshaugen campus, was used. The model was simulated through the summer period with the setpoint $23\,^{\circ}\mathrm{C}$ for ventilative cooling, both day-time and night-time, a $30\,\%$ opening percentage and the mechanical ventilation control algorithm. The building is still considered "semi-exposed" compared to the exposure of the climate meter where measurements were taken. Figure 6.19 presents a carpet plot of the resulting temperatures in the second floor zone through the summer period.

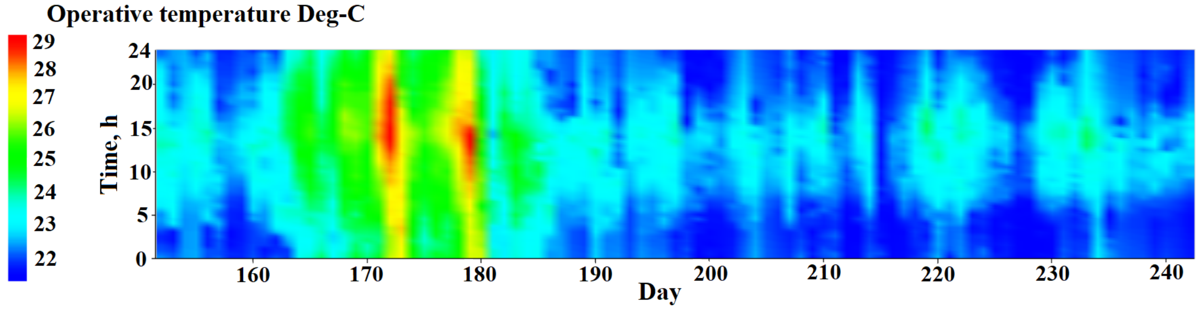
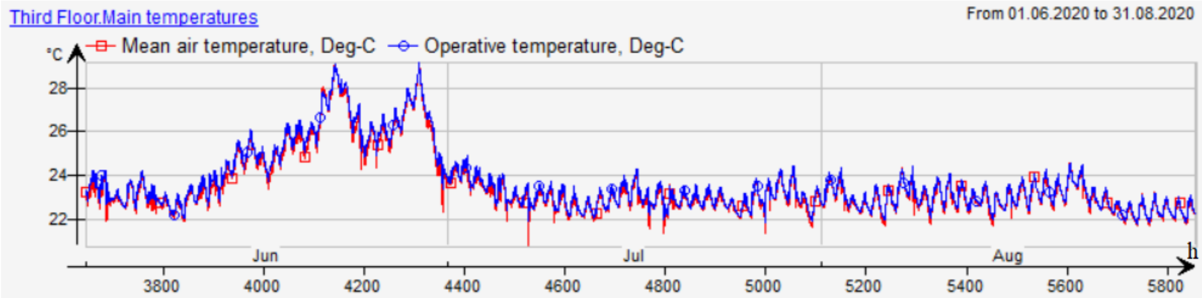


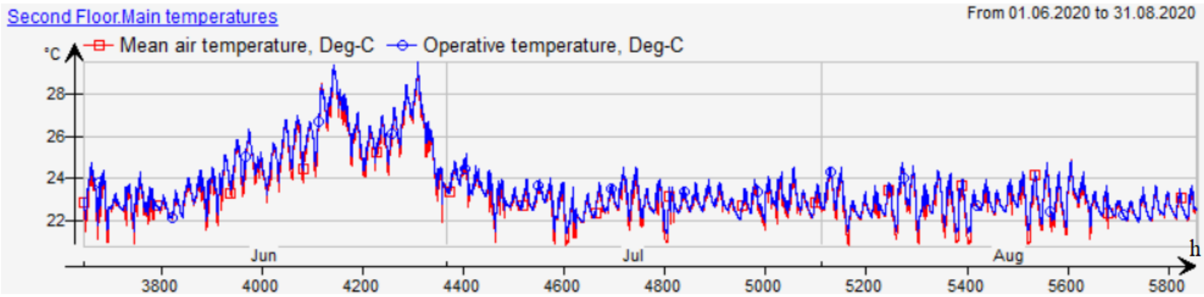
Figure 6.19.: *Carpet plot of the recorded temperatures of Second Floor zone through the summer period. Simulation conducted with a climate data recorded at Gloshaugen campus*

The ambient temperatures of the Gloshaugen climate file are generally lower through the summer than that of the previously used EQUA file. The resulting zone temperatures showcase this as the achieved indoor temperatures rarely exceed 24 °C, lower than that of previous simulated scenarios. The ambient temperature peaks do not follow this trend as they reach 31 °C, an indication that real-world scenarios have a larger variation in the ambient conditions acting on the building. The results of the simulation illustrate a healthy indoor thermal environment. No hours of discomfort is caused by indoor temperatures exceeding 26 °C, due to the high ambient temperatures, and the indoor temperatures are generally below 24 °C through the remaining part of the summer. The CO_2 levels stay within acceptable values for the entirety of the simulated period. The resulting airflow rates and air velocities do no deviate largely from the previously presented results.

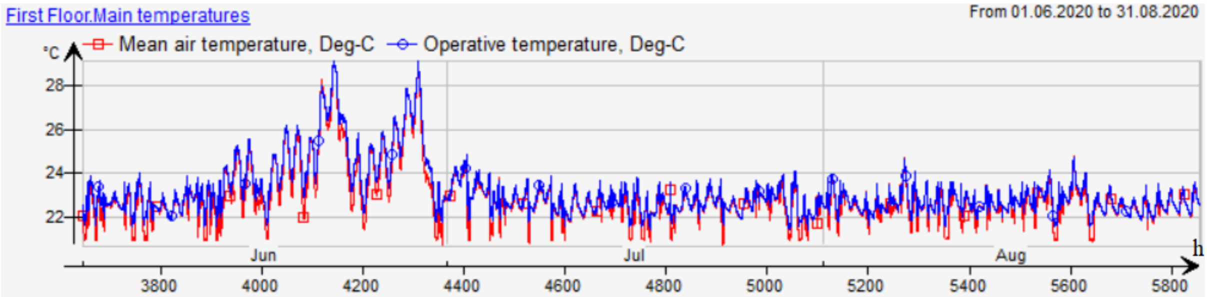
Examining the resulting temperatures further shows that the temperature of the ground floor zone has slightly higher peak temperature during the extra hot periods that occur during the end of June, and a slightly higher general temperature through the remaining simulation. Figure 6.20 presents the temperature curves of the floor zones in the modeled ZEB Laboratory.



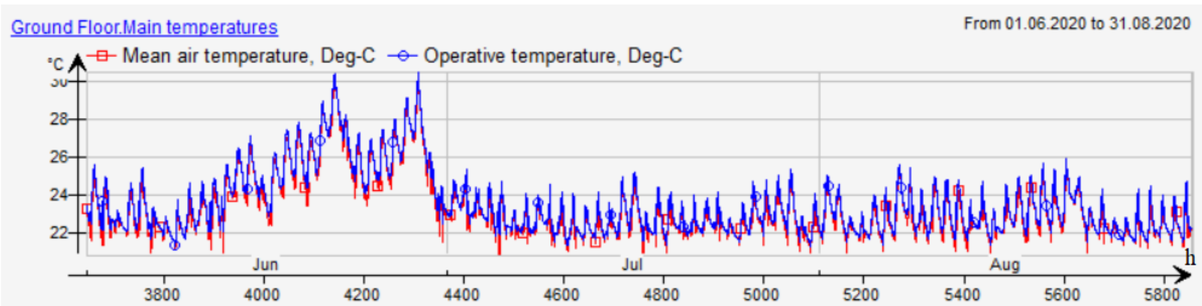
(a) *Resulting temperature in the third floor of the ZEB Laboratory through the summer period*



(b) *Resulting temperature in the second floor of the ZEB Laboratory through the summer period*



(c) *Resulting temperature in the first floor of the ZEB Laboratory through the summer period*



(d) *Resulting temperature in the ground floor of the ZEB Laboratory through the summer period*

Figure 6.20.: *Carpet plot of the recorded temperatures in the floor zones of the modeled ZEB Laboratory, of the simulated scenario. Simulation conducted with ventilative cooling and the presented mechanical ventilation control algorithm*

The generally higher achieved temperatures on the ground floor are assumed to be caused by a low air exchange rate in this zone. The ground floor zones main air supply comes from the small window on the east-side facade. A larger area of automatically operable windows should be investigated on the floor.

The conditions which allow for ventilative cooling are achieved in a larger part of the simulated year when the Gloshaugen climate file is used. Ventilative cooling is used in intervals until the start of October and even some periods of early November. The temperatures of the remaining part of the year stay between 24 °C and 21 °C and CO_2 concentration never exceeds 900 ppm showing that the ventilative cooling can be used in other parts of the year without causing discomfort. This is highly dependent on the ambient conditions. The total energy saving potential of ventilative cooling of the simulated scenario is presented in Table 6.7. The cooling setpoints 26 °C, 25 °C and 24 °C was investigated.

Table 6.7.: *The resulting energy saving potential of ventilative cooling of the Gloshaugen climate file scenario, year round simulation with the cooling setpoints* 26 °C*,* 25 °C *and* 24 °C

Cooling setpoint	**Cooling Demand**	**Peak power demand**	**Power demand per square meter**
26 °C	1520 kWh	15.84 kW	0.900 W/m^2
25 °C	2121 kWh	16.41 kW	0.932 W/m^2
24 °C	3195 kWh	16.98 kW	0.965 W/m^2

6.4. Suggestions for New Window Design

This section will present new suggestions on window design for the ZEB Laboratory to increase the quality of the indoor environment when ventilating the building with natural ventilation.

6.4.1. New Window Design on Ground Floor

The ground floor is supplied with fresh air through the East-side window and the entrance door. As the entrance door is controlled manually and only opens when occupants enter or leave the building, it is assumed that the primary air supply comes from the East-side window. The window is placed very close to the building's main staircase, which acts as a large extract duct for the entire building. A significant concern regarding short-circuiting of the ground floor is speculated due to this design. With a large enough air velocity, the airflow could penetrate far enough into the hallway to achieve circulation from the hallway, into the cafeteria, and back to the staircase. However, the resulting air velocity would have to be unrealistically high to achieve this. In addition, this would likely cause discomfort for occupants as the large air velocities in the occupied zone would cause draught and potential sound problems could be caused by the large air velocity through the relatively small window opening.

Implementing another opening on the West-side of the building will ensure a better spread of the supplied air and better circulation throughout the zone as more air is activated. This would also result in a higher airflow rate in the zone, potentially resulting in a lower indoor temperature, as was shown to be a problem in the simulation scenario presented in Section 6.3.4. An identical window to the one placed on the east-side facade of the ground floor was implemented on the

West-side facade to investigate its effect on the indoor environment. Figure 6.21 presents the resulting temperatures.

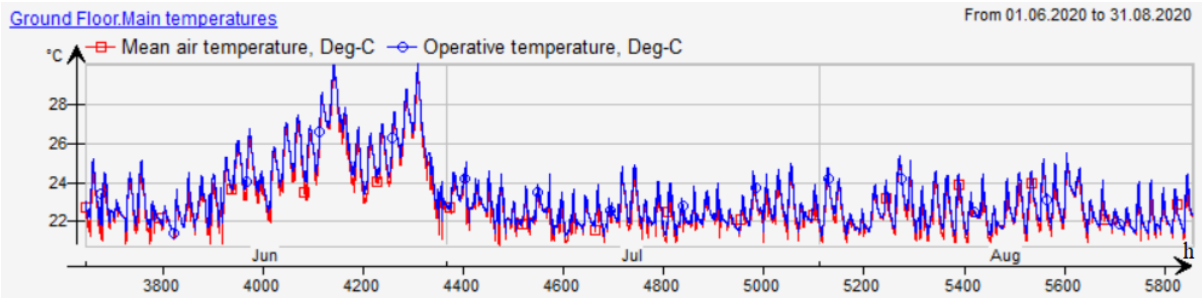


Figure 6.21.: *Resulting indoor temperatures in the ground floor zone of the simulated model with implemented window on the west-side facade*

The resulting temperatures show that the peak and general temperatures in the ground floor zone have slightly lowered, indicating that the indoor environment has benefited from the window implementation. The actual effects of implementing an additional window on the ground floor cannot be realized with the presented simulation method of this thesis. A more complex simulation tool that can predict and evaluate air movement within a zone should be utilized to investigate this further. Physical experiments could also be used but implementing new windows on the facade has a substantial investment cost. The proposed solution to this assumed problem should be evaluated thoroughly before testing.

6.4.2. New Window Design on the Third Floor

The main staircase of the ZEB Laboratory behaves as mention, as the main extract duct of the building. The two large, automatically controlled windows on the upper part of the north facade have, as expected from the theory of buoyancy and shown through the presented results, functioned as the primary openings for extraction of polluted air from the building. Illustrated in Figure 4.2c, one of the windows is placed on the right side of the facade, within the auditorium, and the other is place on the far left side of the facade. Due to the partition between the auditorium and the touch-down area on the third floor, it is speculated that there will not be much air exchange between the two areas. If this is the case, then the air supplied to the auditorium would be exclusively supplied through the west-side window when clean natural ventilation is used. Due to the small opening area of this window, it is speculated that the air velocity through the window would become large due to accumulating temperatures and pressure, and the total airflow rate would be too small for an area with 41 expected occupants. The proposed solution would be to increase the total area of automatically controlled windows. Placement on the south is suggested. The incoming airflow would activate more air movement within the zone, avoiding potential circuiting. The south side of ZEB Laboratory is facing away from the sun, supplying the coldest possible air to the building. Since ZEB Laboratory was modeled without zonal division of the floors, this proposed solution can not be investigated further.

7. Discussion

An extensive literature review forms the basis of this thesis and covers the general theory of indoor environment, building ventilation, and window automation. The literature available regarding the indoor environment and traditional building ventilation is extensive and covered by earlier theses, studies, reports, and standards. Resources at NTNU such as Oria and other databases like Google Scholar and ScienceDirect have been used to ensure the credibility of the collected sources. They have proven helpful in finding relevant information to supplement the review. The literature review on the adaptive thermal comfort model, building automation and window control, and energy-efficient ventilation strategies have been essential in defining the setpoints and parameters of the control algorithms, the presented method, and evaluation of the results. The revised studies and reports showcase that buildings in Northern climates could reduce the cooling demand entirely with ventilative cooling while achieving a comfortable indoor environment, indicating a vast potential for ventilative cooling of the ZEB Laboratory. However, minimal information regarding the adaptability of said strategies to ZEB offices and educational buildings in northern climates was found. Mainly due to the lack of such buildings in operation today or documentation of their daily operation. The ZEB Laboratory has a large thermal transmittance and is therefore expected to have a more considerable cooling demand than the revised ventilative cooled buildings of the Annex 62 studies. Therefore, the presented solutions from the literature review can not be assumed applicable to the ZEB Laboratory without testing and evaluation.

The main purpose of the conducted simulations has been to investigate and determine the ventilative cooling potential of the ZEB Laboratory. The base case scenario showed that cooling is required to achieve an acceptable thermal environment during the simulated summer period. The second floor zone, where the highest percentage hours of discomfort were recorded, measured 230 hours above the acceptable 50-hour maximum. An implemented ideal cooler with a cooling setpoint of $25\,°\mathrm{C}$ was required to achieve a comfortable thermal environment, resulting in energy use of $1141\,\mathrm{kWh}$ and a power demand $0.821\,\mathrm{W/m^2}$. Using the setpoint $24\,°\mathrm{C}$ resulted in a energy use of $2258\,\mathrm{kWh}$ and a power demand $0.880\,\mathrm{W/m^2}$. The required cooling and power demand is small compared to the installed power for space heating, as it is about 200 times smaller. Raising the question, can the initial investment cost of a cooling unit be justified when there are other options and the power demand is so low? Increasing the appeal of ventilative cooling. Implementation of the presented window control algorithm eliminates the cooling demand of the building while maintaining both a healthy atmospheric and thermal environment in all zones of the model. The percentage hours of predicted discomfort was registered below 50 hours throughout the summer season. Increasing the setpoint range of ventilative cooling algorithm showed that indoor temperatures on a "average summer day" could mostly be kept under $24\,°\mathrm{C}$ with a cooling setpoint of $23\,°\mathrm{C}$ or lower.

The resulting $\mathrm{CO_2}$ concentrations of the simulated scenarios show that a satisfactory atmospheric environment is achievable when mechanical or hybrid ventilation is utilized in the ZEB Laboratory. The mechanical ventilation system is designed to cover the entire air pollution load. Therefore, it is not surprising that ventilating with either mechanical or hybrid ventilation is a satisfactory solution for hygienic ventilation.

IDA ICE cannot evaluate air velocities within a zone and, therefore, cannot determine the draught risk a given occupant experiences. In addition, the experience draught from air movement is highly affected by both indoor temperature and the thermal sensation of individual occupants. High indoor air velocities that would traditionally be experienced as uncomfortable may feel like a welcoming breeze in some situations. The calculated separation distance and resulting penetration lengths of the air jets were in all cases lower than the desired percentages of 50 to $60\,\%$. However, this does not necessarily mean that the occupants experience draught. The corresponding separation distance of 50 to $60\,\%$ is desired when supplied air is cold. The exact temperature is not specified, but the ambient air temperature in periods where cooling is required is thought to be generally acceptable, as the temperature difference between the supplied air jet and the indoor temperature does not deviate largely from one another. In addition, the resulting air velocities of the simulated scenario do not stray far from the generally accepted value of $0.3\,\mathrm{m/s}$, indicating that the air velocity of the descending air is also low. The draught risk that occurs when utilizing ventilative cooling during periods of overheating is expected to be low, as high levels of air movement are in most cases considered comfortable when indoor temperatures are high.

The clean natural ventilation solution was unable to achieve an acceptable indoor environment. The defined parameters of the window control algorithm do not allow for ventilation during specified ambient conditions resulting in periods without ventilation. The general airflow rate supplied to the building by the natural ventilation was sufficient in correlation to the requirements of TEK17 but due to the nature of the controller some periods with $\mathrm{CO_2}$ concentrations of $2000\,\mathrm{ppm}$ were measured. Under the presented circumstances, the ZEB Laboratory cannot be ventilated with clean natural ventilation resulting in accumulation of $\mathrm{CO_2}$ and an increase in indoor temperatures. However, with the implemented mechanical ventilation control algorithm, acceptable indoor conditions were achieved, and $68\,\%$ of hours where mechanical ventilation would have been used could be replaced with natural ventilation. Resulting in a fan power reduction of $65\,\%$, $5.51\,\mathrm{kWh}$ to $1.93\,\mathrm{kWh}$.

The simple on-off window control algorithm was found to achieve a healthy indoor environment during the summer season. It is speculated that the presented fan power reduction could be reduced further with a more complex algorithmic solution on both the window and mechanical ventilation control algorithm. Examining the results, it is shown that unnecessary mechanical ventilation occurs during short periods throughout the simulations. This occurs when the ambient temperature or wind velocity achieves unacceptable values for natural ventilation for short periods. The low $\mathrm{CO_2}$ accumulation during the short period where natural ventilation is inoperable does not justify the need for mechanical ventilation. A more complex automatic controller, like a feed-forward control or machine learning system, could in theory, anticipate the length of this period and determine the best possible action with regards to the thermal and atmospheric environment. The potential reduction in fan power requirement could potentially be reduced further.

The draught risk becomes substantially higher when ventilating the ZEB Laboratory with natural ventilation. The same window control algorithm is used, but due to accumulating $\mathrm{CO_2}$ levels from the absence of mechanical ventilation, windows are open for a longer total period through the summer, and during periods with substantially lower ambient temperatures. The window control algorithm allows for operation until ambient temperatures fall below $12\,°\mathrm{C}$. However, due to increasing drought risk with falling supply temperatures, this setpoint is speculated to be to low due to the increasing temperature difference between the indoor and supplied air temperature. A lower setpoint would require an increase in mechanical ventilation which would increase the fan power requirement. The resulting air velocities and separation distances from the investigation of the window opening percentage on the second floor, south-side window showcase a considerable degree of variation through the simulated day. The air velocity rapidly increases and decreases in intensity and changes from inflow to outflow and back over short periods, rarely holding a particular air velocity value for an extended period and resulting in short bursts of large airflow at varying velocities throughout the day. This variation is not expected to cause any draught due to the short periods these high air velocities are achieved. The separation distance of the supplied air-jet follows the pattern of the varying air velocity. The resulting separation length is generally lower than the desired 50 to 60 % room length, causing concern regarding periods of draught. The presented results illustrate a turbulent picture that complicates the evaluation of the draught risk.

Giving occupants the ability to control the ventilative cooling is essential. As presented in the literature, occupants with the ability to interact with the building and its systems significantly increase the satisfaction towards the thermal environment. Some concerns occur in giving occupants control. With no implemented cooling, ventilative cooling is proven essential to eliminate the overheating of the ZEB Laboratory. An override and closing of the windows would cause a decrease in the quality of the thermal environment as temperatures increase and air movement within the building decreases. It is assumed as a low-risk solution as occupants, due to behavioral adjustments, will generally open windows in periods of high temperatures as a natural reaction to their thermal discomfort. To avoid this potential problem altogether, an ambient temperature setpoint where occupants cannot interact with the system of, for example, 26 °C and above, could be used.

To summarize, draught risk is a factor that is complicated to determine. Many factors influence the thermal sensation of an individual occupant, and the results IDA ICE provides are complicated and somewhat limited. The draught risk when using ventilative cooling to combat overheating is assumed low due to these period's high indoor and ambient temperatures. The draught risk when ventilating the building naturally is uncertain and is assumed, with the presented natural ventilation strategy and the setpoints used, to large in periods where ambient temperatures fall close to 12 °C. This strategy is not recommended without further and more detailed research and evaluation.
It must also be mention that the pressure profile of the building will potentially change when simulating the building with the correct zone division. The addition of partitions between the different rooms would increase the internal friction of the model, which would potentially result in an entirely different airflow pattern in the building. CFD simulation on a more realistic model or actual physical experiments in the ZEB Laboratory should be considered if further investigation of the proposed solution's are intended. The potential draught risk and applicability is recommended to explore first.

Nigh-time cooling was investigated, and it was shown that the natural and mechanical ventilation of the ZEB Laboratory could benefit from utilizing the ventilative cooling strategy. This was to some surprise as the thermal mass of the solid tree construction of the ZEB Laboratory is not very large, and therefore, the cooling potential of night-cooling was expected to be low. The natural ventilated scenarios showed that an acceptable indoor environment without night-time cooling could be achieved. However, a more satisfactory indoor environment with temperatures below 24 °C was achieved when night-time cooling was implemented. The mechanical ventilation scenarios achieved acceptable indoor temperatures with and without night-time cooling on the "average day". Results of the simulated "hot day" show that night-time cooling was required to achieve temperatures below 27 °C. The night-time cooling strategy should be utilized in ventilative cooling strategy of the ZEB Laboratory. It is assumed that a higher air exchange rate, which was provided by the night-time cooling solution, was necessary to remove the undesired heat from the indoor air.

The low energy reduction possibility from natural ventilation utilization raises the question, is clean natural ventilation of the ZEB Laboratory a justified solution? Windows have to be open for a more extended period to achieve sufficient airflow rate for pollution removal of the building. As discussed, the actual draught risk is hard to predict, and the results show a high degree of uncertainty. The longer the windows are open, the higher the risk of discomfort occur. Clean natural ventilation would have been a more interesting solution to evaluate and discuss if the building's mechanical ventilation system was not already implemented, as investment cost would be part of the equation. However, mechanical extraction of air is required in specific rooms in buildings, such as bathrooms which makes a mechanical ventilation system a requirement. The relatively low energy requirement for fan operation makes it hard to debate against the use of the mechanical ventilation system as the general quality and indoor environment increases.

The proposed new window design on the ground floor is a necessary implementation for natural ventilation to be an appropriate solution to the ZEB Laboratory. A considerable concern regarding short-circuiting of the ground floor is speculated due to this present design, furthering the cost of clean natural ventilation implementation. However, implementing a window on the West-side of the ground floor may benefit the building's indoor environment regardless of the chosen ventilation solution. During the hotter periods of the summer, air movement, as mention, may increase comfort in occupants, which could be achieved with a window implementation in this area. The cafeteria door which have not been discussed in this report thus far could achieve the same mixing effect and air movement in the zone. However, large air velocities would be achieved in the occupied area, as the door supplies air at occupancy height and along the floor. Due to the uncertainty of drought, the cafeteria door should only be manually controlled.
The implementation of the additional window was also supported by the results of the realistic weather file simulation scenario. The simulation was conducted with the intention of testing the sensitivity of the model. The realistic file provided both lower and higher ambient temperatures and the results showed that a satisfactory indoor environment was achieved through the summer period. The temperature in the ground floor zone showed higher peak temperatures during the extra hot periods of the summer, indicating that the airflow rate provided on the ground floor was not sufficient in combating a heat supply of this magnitude. An increase in window area on the ground floor was an assumed solution. In addition, a window placement on the West facade would increase the air movement within the zone which could help combat overheating of occupants during periods of overheating.
The speculated short circuit achieved on the third floor was not possible to investigate as the ZEB Laboratory was modeled without zone division on the floors of the building. The potential short circuit should be investigated through physical experiments as there is some uncertainty if the speculated problem exists.

The model of this thesis was constructed without zone division on the floors of the building. Chosen to reduce the computational time and complexity of the simulation and was assumed acceptable due to the open airflow nature of the floor designs of the building. This simplification was initially deemed acceptable as the resulting ventilative solution was supposed to be tested in the before-mentioned realistic ZEB Laboratory model, but was not utilized due to technical problems. A model with more realistic zone division would have been created and utilized for the thesis if the complication was discovered earlier in the master thesis work period.

The ZEB Laboratory was created as a test facility with the adaptability to investigate new technologies and ventilation solutions. Though clean natural ventilation of the ZEB Laboratory was found to be an insufficient solution, it was interesting to investigate the possibility and results of clean natural ventilation. Buildings like the Nydalen Vy are now and may continue to use clean natural ventilation solutions in the future. The ZEB Laboratory needs to be able to test similar solutions to live up to its intended purpose, which the presented results of the simulated natural ventilation scenario shows promise towards.

8. Conclusion

The primary purpose of the research and conducted simulations of this thesis have been to investigate the ventilative cooling potential of the ZEB Laboratory. A literature review researching existing ventilative cooling strategies and how best to utilize them in northern climates was conducted in preparation for this task. The results show that the ambient conditions of the northern climate have a substantial potential for cooling and that different ventilative cooling strategies had been found successful in both office and educational buildings. However, minimal information regarding the adaptability of these strategies to the ZEB office and educational buildings located in northern climates was found. Therefore, the presented strategies could not be assumed applicable to the ZEB Laboratory without further investigation. However, an earlier conducted study of hybrid ventilation on the ZEB Laboratory has investigated optimal ventilation methods during the different seasons, concluding that a satisfactory indoor environment could be achieved with a clean natural ventilation system during the summer season.

Through simulation in the tool IDA ICE, the ventilative cooling potential of the ZEB Laboratory was investigated. The results of the conducted simulation scenarios were evaluated based on thermal comfort, indoor air quality, and energy consumption. Firstly, a hybrid ventilation solution was investigated, where ventilative cooling applied through automatically controlled windows was used to combat periods of overheating. The preferred controller was an on-off control that used indoor temperature, CO_2 concentration and weather conditions to determine optimal window operations for ventilative cooling, a control strategy presented in the literature review, and proven applicable to the ZEB Laboratory through simulation. The proposed strategy was found successful in achieving a healthy indoor environment with acceptable levels of occupancy discomfort and CO_2 concentration. Without the use of ventilative cooling, an ideal cooler with a cooling setpoint of $25\,°C$ was required to achieve a comfortable thermal environment, resulting in a energy use of $1141\,kWh$ and a power demand $0.821\,W/m^2$ through the summer period. Using a cooling setpoint of $24\,°C$ resulted in a energy use of $2258\,kWh$ and a power demand $0.880\,W/m^2$. The presented simulation results show that the cooling demand could be completely removed when the window control algorithm was applied while a healthy atmospheric and thermal environment was maintained in the building. The potential discomfort caused by drought was deemed minimal as ventilative cooling would mainly be utilized in periods with high indoor temperatures, periods where draught, more often than not, would have a comfortable cooling effect.

Secondly, clean natural ventilation was investigated. As was concluded in earlier research on the ZEB Laboratory, the achieved airflow rates supplied to the building were above recommended levels, and the achieved indoor environment was within satisfactory conditions. However, this was only the case in periods where ambient conditions allowed for window operation. High temperatures and CO_2 concentrations were achieved outside these periods, and mechanical ventilation was required to achieve an acceptable indoor environment throughout the summer season. Clean natural ventilation was discovered to not an acceptable ventilative solution of the ZEB Laboratory under the presented conditions. With the implementation of a created mechanical ventilation control algorithm, $68\,\%$ of hours of mechanical ventilation could be replaced with natural ventilation, resulting in a fan power requirement reduction of $3.58\,kWh$ from $5.51\,kWh$. The draught risk was substantially larger and more uncertain for the natural ventilation solution than the hybrid solution, and a possible short circuit on the ground and the third floor was speculated. Additional windows on the West facade of the ground floor and the south facade of the third floor were suggested as possible solutions to the speculated short circuit.

The lowest total energy consumption of the ZEB Laboratory was achieved when ventilating the building with natural ventilation supported by the mechanical ventilation control algorithm. However, the hybrid ventilation solution with automatically controlled windows for ventilative cooling was chosen as the best way to ventilate and apply ventilative cooling to the ZEB Laboratory. The difference in fan power requirement of the two solutions was relatively small, and the potential reduction of energy use achieved with the natural ventilation solution was overshadowed by the disadvantages of increased drought risk and potential short circuits.

9. Further Work

The first thing that should be addressed for further work should be implementing the presented ventilative cooling strategy to a more realistic model, as was initially planned for this thesis. Realistic zone division of the floors may affect the flow pattern of the building and should therefore be investigated. The realistic modeled ZEB Laboratory which has been mention in the thesis could be utilized, or in another building.

If there is an interest in further investigating the natural ventilation solution of the ZEB Laboratory, physical experiments or CFD simulation should be utilized. Draught risk and potential short circuits were shown to be the primary concern of this strategy. Different uncertain factors made it hard to give a definitive answer, and a more precise method is suggested.

A study on the effect of dynamic solar shading in combination with ventilative cooling could be investigated. The modeled ZEB Laboratory operated with constant solar shading during this study which is not an optimal solution as daylight can help increase comfort in occupants.

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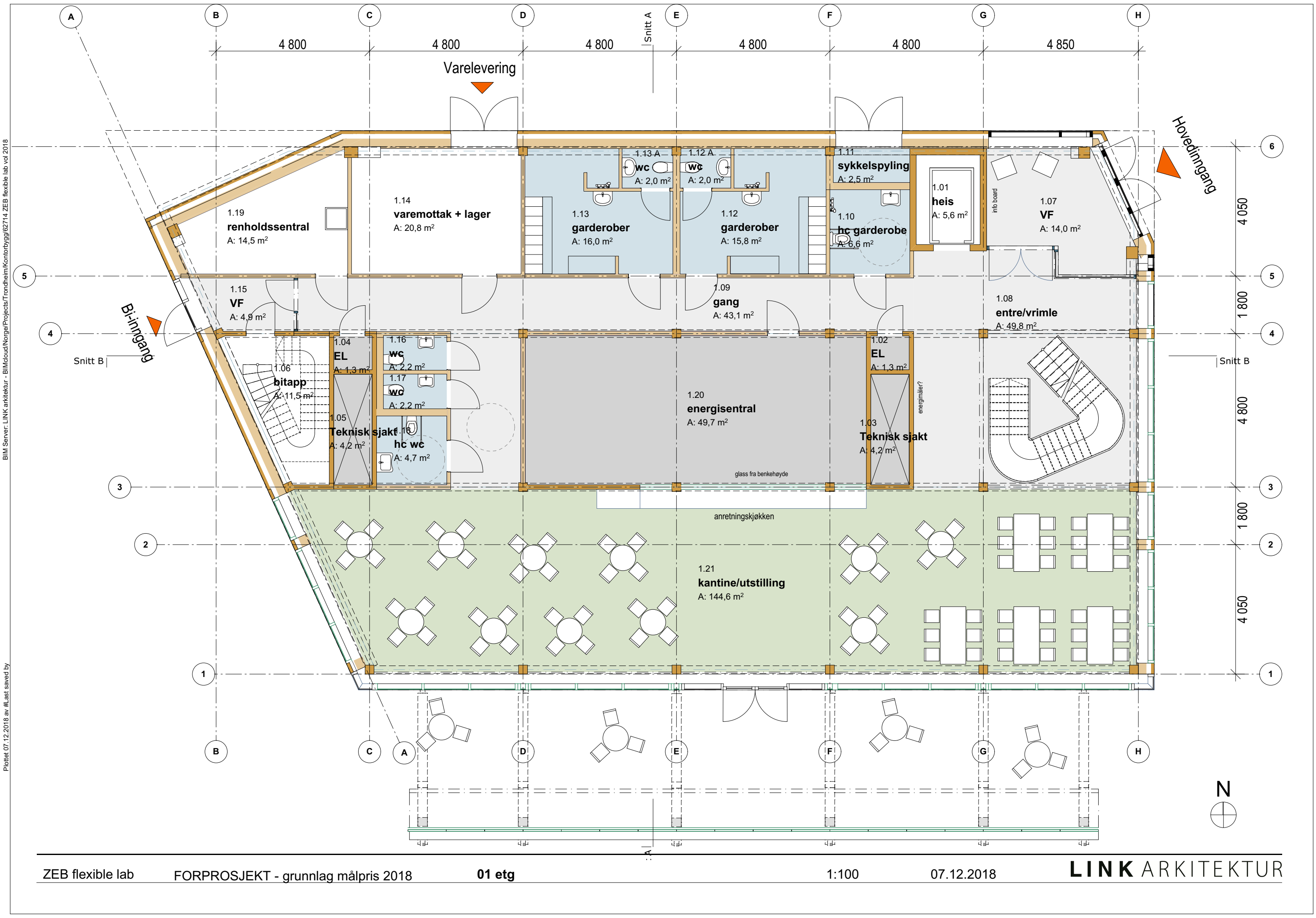
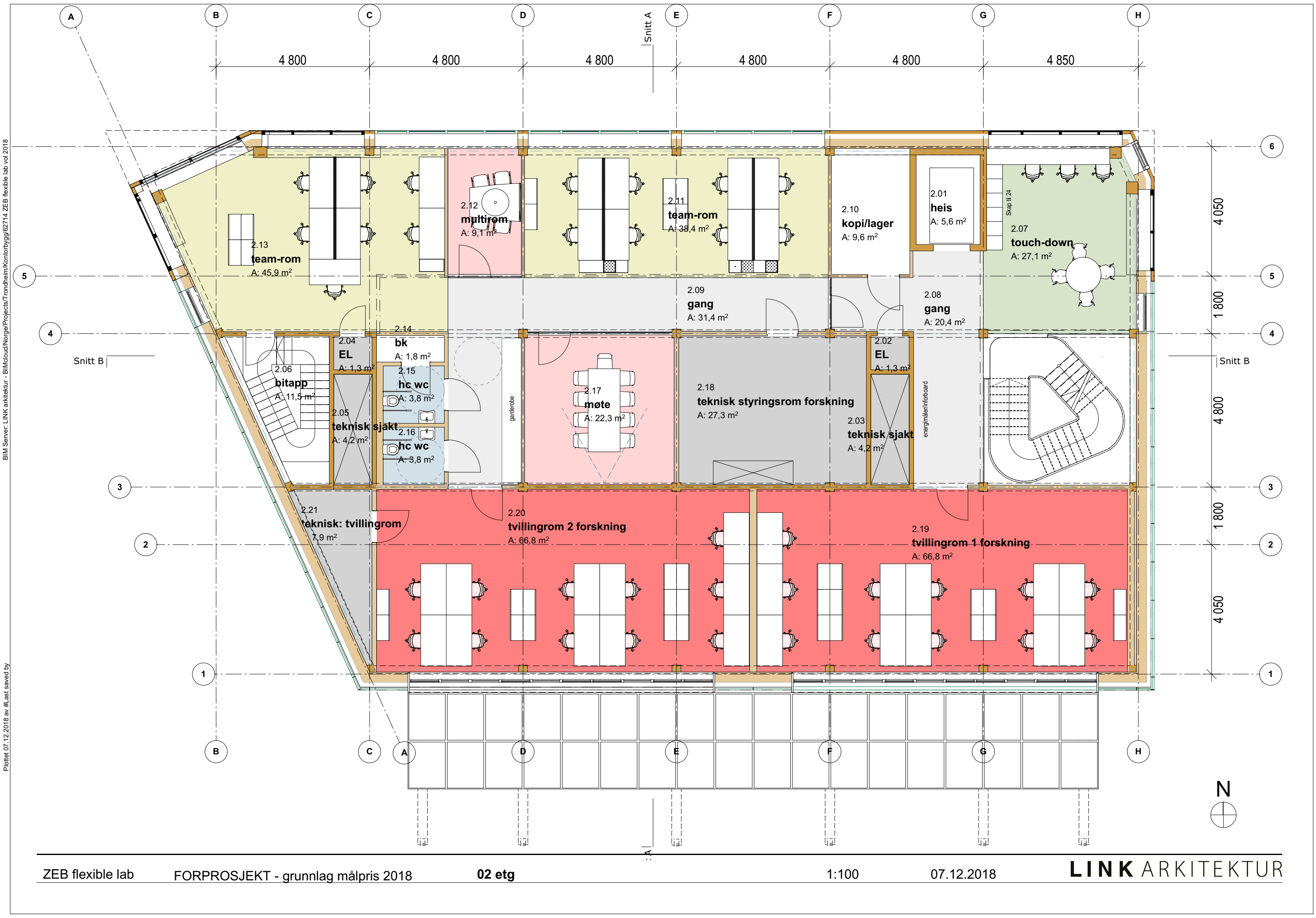
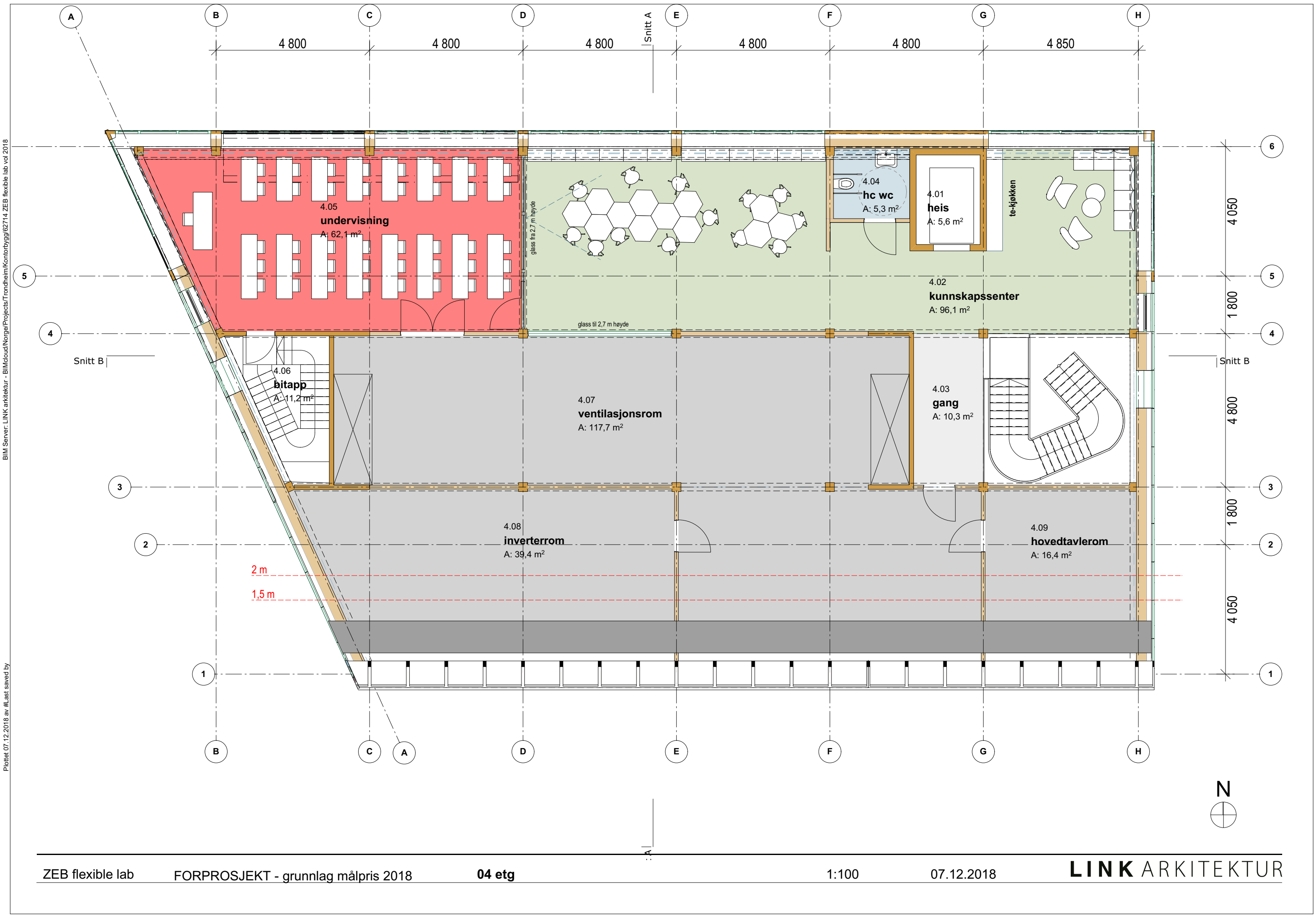
Appendices

A. ZEB Laboratory Floor Plans




2.13
team-rom
A: 45,9 m²
2.12
multirom
A: 9,1 m²
2.11
team-rom
A: 38,4 m²
2.10
kopi/lager
A: 9,6 m²
2.01
heis
A: 5,6 m²
2.07
touch-down
A: 27,1 m²
2.09
gang
A: 31,4 m²
2.08
gang
A: 20,4 m²
2.14
bk
A: 1,8 m²
2.04
EL
A: 1,3 m²
2.06
bitapp
A: 11,5 m²
2.15
hc wc
A: 3,8 m²
2.05
teknisk sjakt
A: 4,2 m²
2.16
hc wc
A: 3,8 m²
2.17
møte
A: 22,3 m²
2.18
teknisk styringsrom forskning
A: 27,3 m²
2.02
EL
A: 1,3 m²
2.03
teknisk sjakt
A: 4,2 m²
2.21
teknisk: tvillingrom
A: 7,9 m²
2.20
tvillingrom 2 forskning
A: 66,8 m²
2.19
tvillingrom 1 forskning
A: 66,8 m²
Snitt A
Snitt B
4 800
4 850
4 050
1 800
N
ZEB flexible lab
FORPROSJEKT - grunnlag målpris 2018
02 etg
1:100
07.12.2018
LINK ARKITEKTUR

ZEB flexible lab	FORPROSJEKT - grunnlag målpris 2018	**03 etg**	1:100	07.12.2018	**LINK** ARKITEKTUR


Snitt A
4 800
4 800
4 800
4 800
4 800
4 850
4.05
undervisning
A: 62,1 m²
glass fra 2,7 m høyde
4.04
hc wc
A: 5,3 m²
4.01
heis
A: 5,6 m²
te-kjøkken
4.02
kunnskapssenter
A: 96,1 m²
glass til 2,7 m høyde
Snitt B
4.06
bitapp
A: 11,2 m²
4.07
ventilasjonsrom
A: 117,7 m²
4.03
gang
A: 10,3 m²
4.08
inverterrom
A: 39,4 m²
4.09
hovedtavlerom
A: 16,4 m²
2 m
1,5 m
4 050
1 800
4 800
1 800
4 050
N
ZEB flexible lab
FORPROSJEKT - grunnlag målpris 2018
04 etg
1:100
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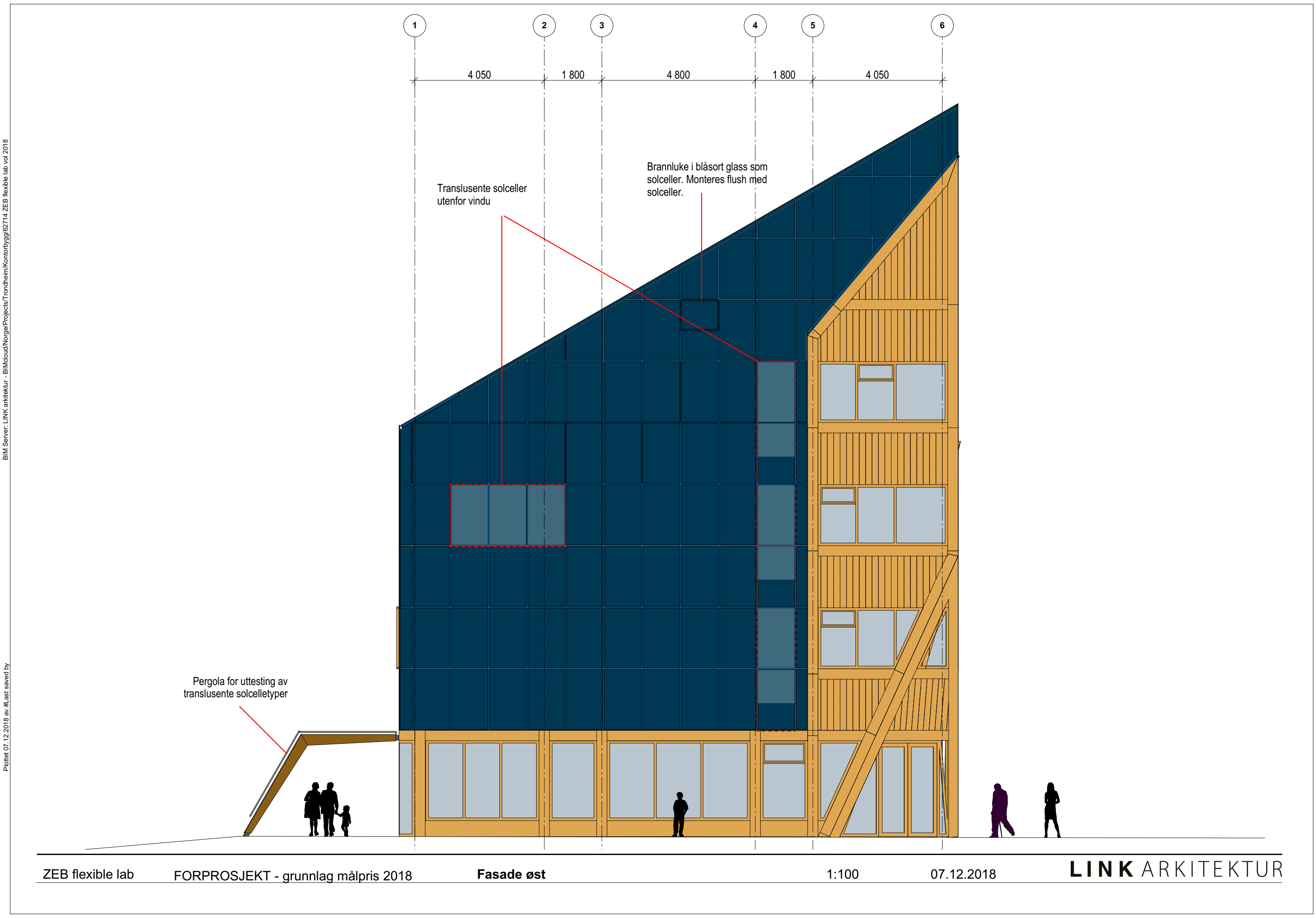
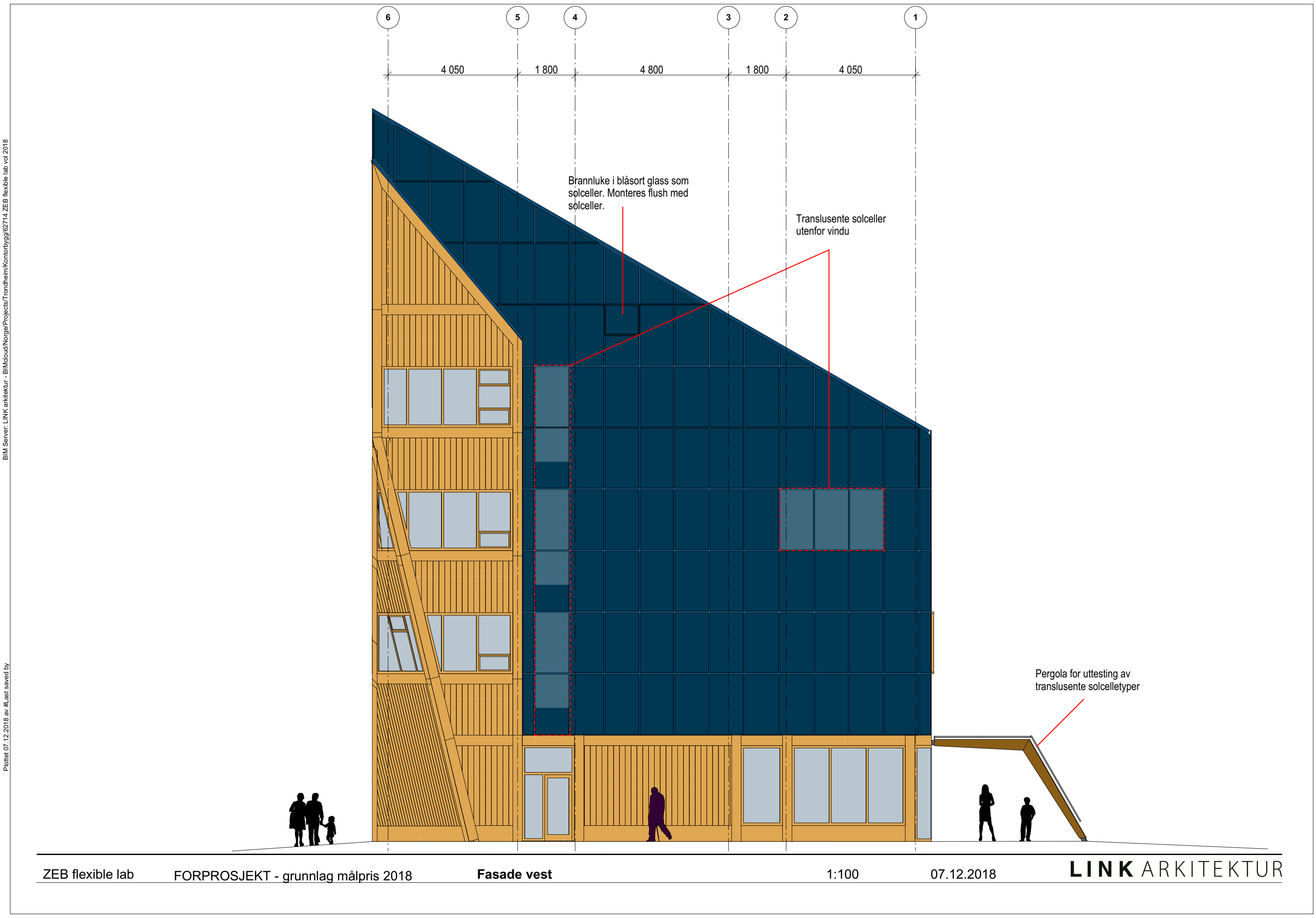
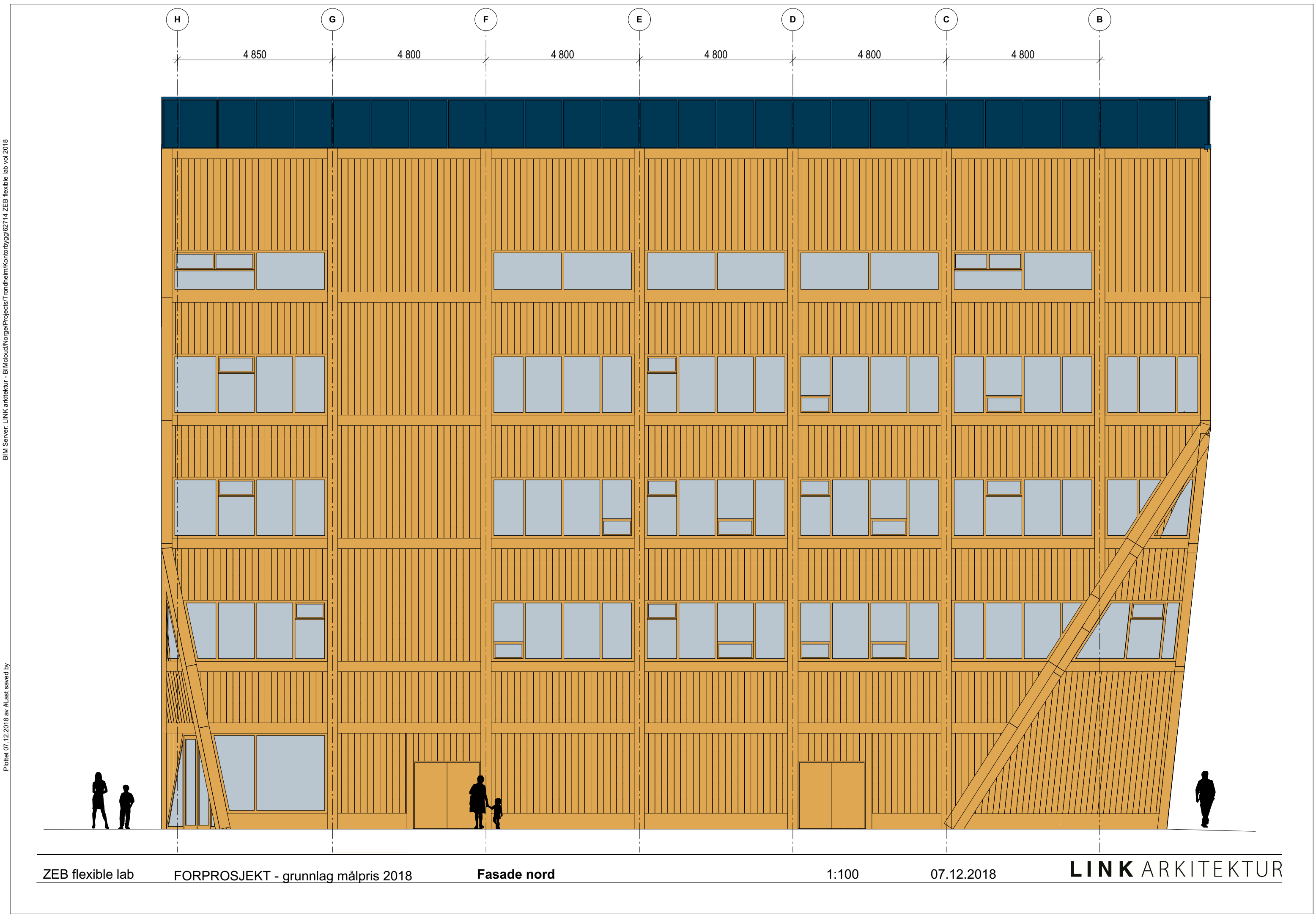
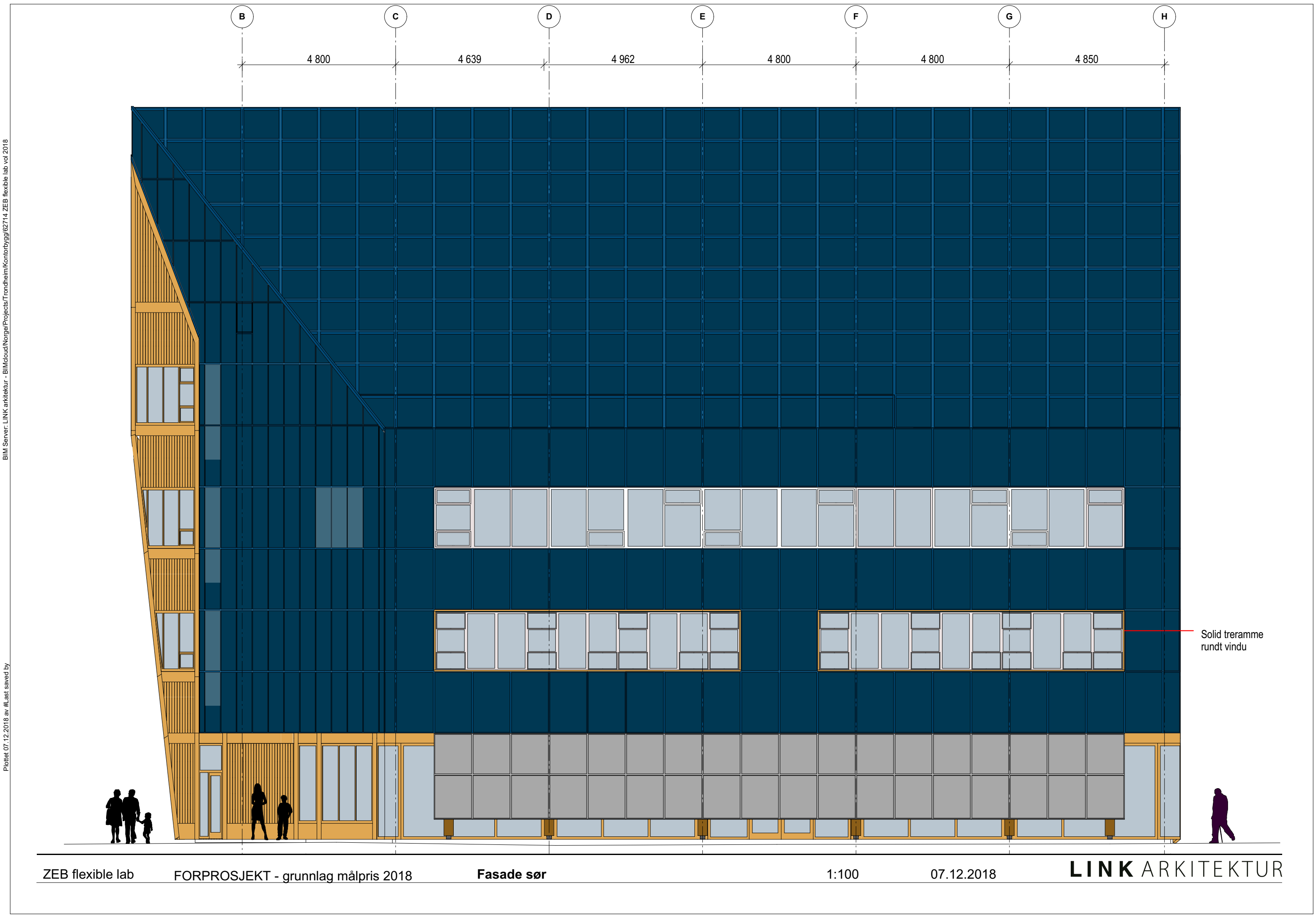
LINK ARKITEKTUR

B. ZEB Laboratory Facade Drawings




6
5
4
3
2
1
4 050
1 800
4 800
1 800
4 050
Brannluke i blåsort glass som solceller. Monteres flush med solceller.
Translusente solceller utenfor vindu
Pergola for uttesting av translusente solcelletyper
ZEB flexible lab
FORPROSJEKT - grunnlag målpris 2018
Fasade vest
1:100
07.12.2018
LINK ARKITEKTUR


H
G
F
E
D
C
B
4 850
4 800
4 800
4 800
4 800
4 800
ZEB flexible lab
FORPROSJEKT - grunnlag målpris 2018
Fasade nord
1:100
07.12.2018
LINK ARKITEKTUR

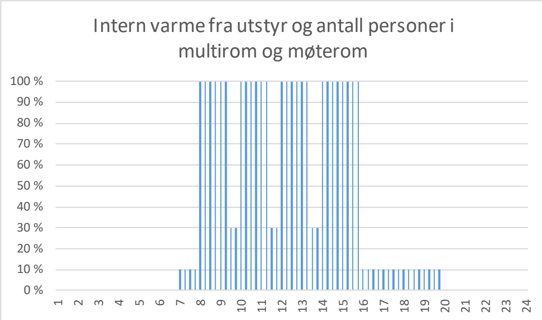

B
C
D
E
F
G
H
4 800
4 639
4 962
4 800
4 800
4 850
Solid treramme rundt vindu
ZEB flexible lab
FORPROSJEKT - grunnlag målpris 2018
Fasade sør
1:100
07.12.2018
LINK ARKITEKTUR

C. User Profiles of the ZEB Laboratory

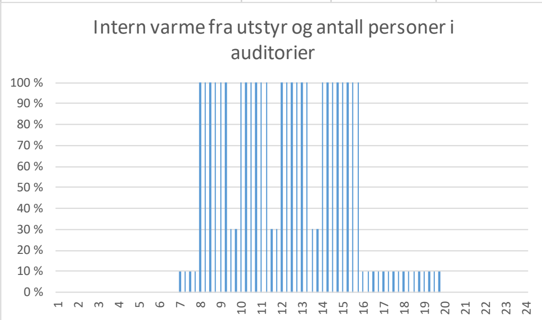
LINK arkitektur AS   Dok.navn: 62714 ZEB Flexible Lab – Energibudsjett premissnotat.   Revidert: **28.11.2017**

Brukerprofiler

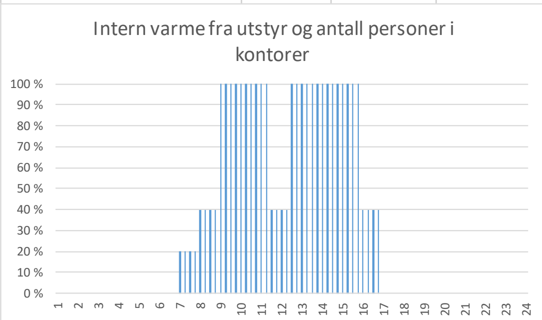
Møterum/multirom



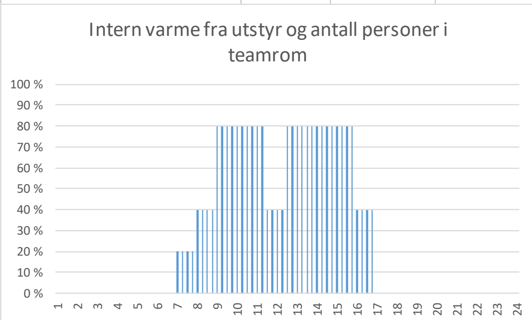
Auditorier



Kontorer



Team rom



Kantine/touch down arbeidsplasser

D. Window Control Algorithm for Ventilative Cooling

Figure D.1 presents the window control algorithm of the modelled ZEB Laboratory. The illustration is been marked with letters which, coupled with a step-by-step description presented below the figure, explains the working mechanism of the controller.

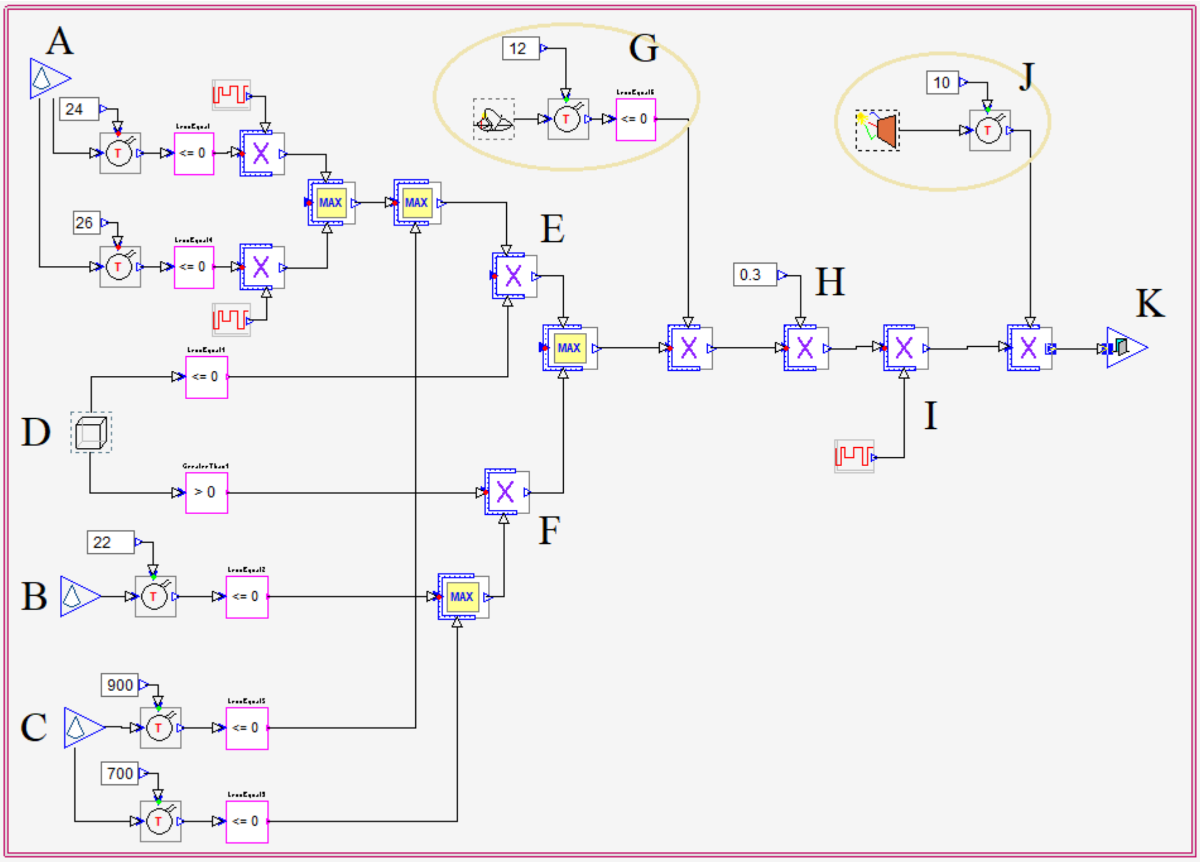


Figure D.1.: *Window control algorithm of the modelled ZEB Laboratory, marked*

Below follows a step-by-step description of the working mechanism of the presented control algorithm:

- A Sends the maximum air temperature of the specified zone
- B Sends the minimum air temperature of the specified zone
- C Sends the maximum CO_2 concentration of the specified zone
- D Sends 1 if the window is open or 0 if the window is closed
- E Sends 1 if the maximum air temperature, while the window is closed, exceeds either 24 °C during the day or 26 °C during the night, or the maximum CO_2 concentration exceeds 900 ppm
- F Sends 1 if the windows is open and either the minimum air temperature is above 22 °C, or the CO_2 concentration exceeds 700 ppm
- G Sends 1 if the ambient temperature is above 12 °C

H Specifies the opening percentage of the implemented automatically controlled windows
I Sends 0 at the time stamps 07:00 am and 18:00 pm to switch between day time and night time ventilation
J Sends 1 if the wind velocity of the ambient conditions do not exceed $10\,\mathrm{m/s}$
K Opens the window if the resulting signal is 1 and closes the window if the resulting signal is 0

E. Mechanical Ventilation Control Algorithm

Figure E.1 presents the mechanical ventilation control algorithm. The illustration has been marked with letters which, coupled with a step-by-step description presented below the figure, explains the working mechanism of the controlled.

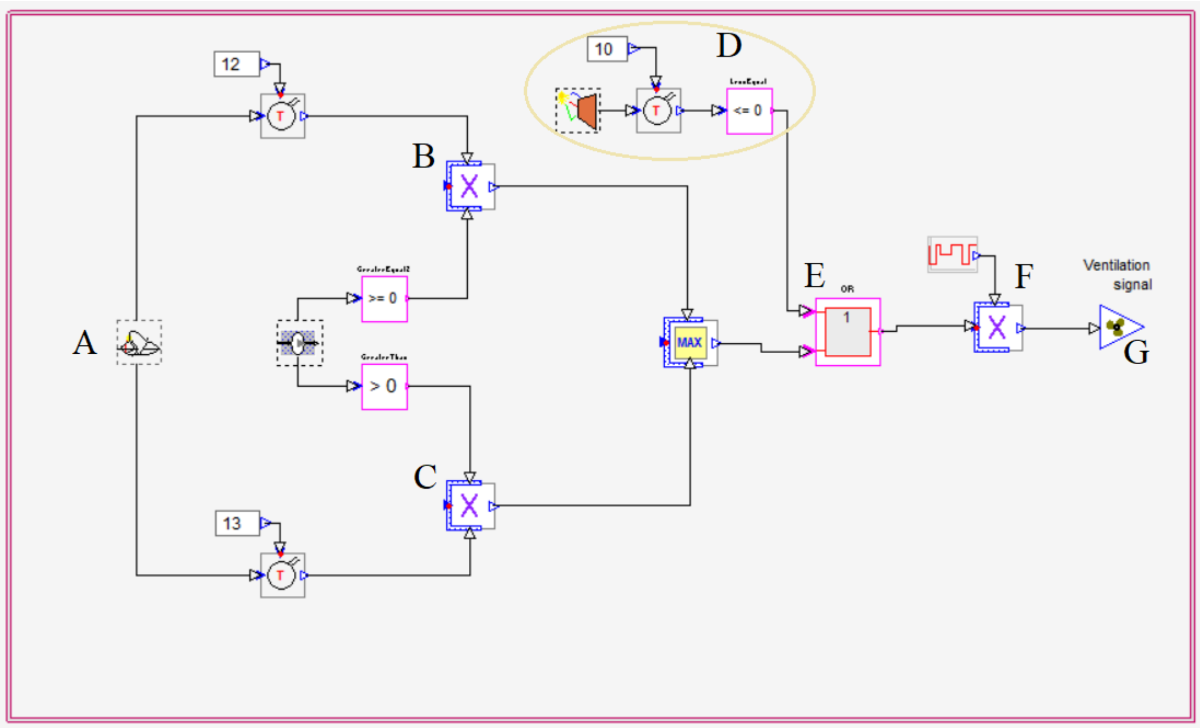


Figure E.1.: *The mechanical ventilation control algorithm of the modelled ZEB Laboratory, marked*

Below follows a step-by-step description of the working mechanism of the presented control algorithm:

- A Sends the ambient air temperature
- B Sends 1 if the mechanical fan of the HVAC system is of and the ambient temperature is below $12\,°\mathrm{C}$.
- C Sends 1 if the fan of the HVAC system is active and the ambient temperature is below $13\,°\mathrm{C}$
- D Sends 1 if the wind velocity acting on the specific facade exceeds $10\,\mathrm{m/s}$
- E Sends 1 if either the ambient temperature falls below $12\,°\mathrm{C}$ or the ambient wind velocity acting on the specific facade is greater than $10\,\mathrm{m/s}$
- F Specifies the capacity of the fan power. Sends 1 if there are occupants present in the zone, otherwise 0,1.
- G Activates the mechanical ventilation if the resulting signal is larger than 0 and deactivates if the resulting signal is 0

Martin Sande

Ventilative Cooling Potential of the ZEB Laboratory

NTNU
Norwegian University of
Science and Technology