Thermal mass Activation

in relation to

Power House One

by

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Preface

I have to thanks the people in my direct environment giving me support during this period of hard work. In this case my future wife Maria Charlotta Brandkvist Lijnen went through a lot. She gave me also support by correcting my thesis so grammatically it would be better.

I'm also grateful for the guidance and good support from my supervisor Matthias Haase. Who gave me good advice for tackling this topic of thermal mass activation.

I have to thank also Bjørn Wachenfeldt from Skanska for giving a lot of good information considering Power House One. His feed back during the process of my thesis was also good and it improved my thesis.

Abstract

In building simulations it is common practice to use standardized occupant behavior and internal gains. Although this is a valid approach for designing systems, the probabilistic nature of these boundary conditions influences the energy demand and achieved thermal comfort of real systems[1]. This paper analyzes the influence of occupant behavior on the energy performance and thermal comfort of three model rooms equipped with thermal mass activation. Three TRNSYS models with weather data form Trondheim were set up.

First, the energy performance and thermal comfort of thermally activated building elements are compared with the performance of idealized cooling with standardized user behavior. Thermal mass activation is able to deliver a better thermal comfort than ventilation heat but results in the parametric studies shows to have a higher energy demand. The influence of the cooling load was investigated by ASHRAE-guidelines. It is shown that the ceiling is the best location for tackling the cooling load in an indoor space.

It is also proven that integrating thermal mass activation is a valuable component for improving the thermal comfort of an indoor space. While its performance is better than that of ventilation, there is not thus far a clear answer as to whether TMA is more energy efficient than activating this same thermal mass by ventilation heat. Further research is necessary to investigate this issue.

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Table of symbols

Symbol	Definition	Value
A	Surface area	m^2
q_{TMA}	Heat exchange coefficient thermal mass activation	W/m^2
$\theta_{S,m}$	average surface temperature	°C
θ_i	nominal indoor temperature	°C
$H_{T,ie}$	transmission heat loss coefficient from heated space to the exterior through the building envelope	W/K
$H_{T,iue}$	transmission heat loss coefficient from heated space to the exterior through the unheated space	W/K
$H_{T,ig}$	steady state ground transmission heat loss coefficient from heated pace to the ground	W/K
$H_{T,ij}$	transmission heat loss coefficient from heated space to a neighbouring heated space at a significantly different temperature, i.e. an adjacent heated space within the building entity or a space of an adjacent building entity	W/K
$\theta_{int,i}$	internal design temperature for heating	°C
θ_e	outside temperature	°C
U_{value}	Heat transfer coefficient	W/m^{2}l
$\Phi_{V,i}$	ventilation heat loss	W/m^2
$H_{V,i}$	ventilation heat loss coefficient	W/m^2
$V_{V,i}$	ventilation mass flow rate	m^3/h
ρ	density of air (=1.23kg/m3)	kg/m^3
c_p	heat capacity of air(=1003,5J/kg·K)	J/kg·K
θ_a	indoor air temperature	°C
θ_v	temperature ventilation air	°C
E_t	total solar radiation incident on surface (TRNSYS)	W/m^2
α	absorption of cladding	

h_o	coefficient of heat transfer by long-wave radiation and convection at outer surface	W/m^2K
$SHGC(\theta)$	beam solar heat gain coefficient as a function of incident angle θ	
$\langle SHGC \rangle_D$	diffuse solar heat gain coefficient	
U	overall U-factor, including the frame	W/m^2K
$IAC(\theta\,\Omega)$	indoor solar attenuation coefficient for beam solar heat gain coefficient; 1.0 if no inside shading device.	
IAC_D	indoor solar attenuation coefficient for beam solar heat gain coefficient; 1.0 if no inside shading device.	
C_p	specific heating capacity of water	J/kgK
SPP	specific pump power	kW/l/s
SFP	specific fan power	kW/m3/
P_{sp_norm}	specific norm power	kJ/hm^2
U_{wrx}	heat tranfer coefficient	kJ/hm^2K
R_w	thermal resistance fluid to pipe	m^2K/W
R_r	thermal resistance pipe	m^2K/W
R_x	thermal resistance x-direction	m^2K/W
$\dot{m}_H$	flow rate	l/s
t	time	hour
T	pipe spacing	cm
$\dot{m}_{H,Sp}$	specific mass flow rate	kg/hm^2
d_a	external diameter of pipe	m
s_r	thickness of pipe wall	m
λ_r	thermal conductivity pipe wall	W/mK
L_r	length of circuit	m
$h_{Fl,H}$	heat capacity floor heating	W/m^2K
$h_{Ceil,H}$	heat capacity ceiling heating	W/m^2K

$h_{Fl,Co}$	heat capacity floor cooling	W/m^2K
$h_{Ceil,Co}$	heat capacity ceiling cooling	W/m^2K
$R_{Concr.Fl.}$	concrete on top of the pipe	W/m^2K
$R_{Concr.Ceil.}$	concrete under the pipe	W/m^2K
$\sigma_{Co.}$	temperature difference water inlet outlet	°C
Q	total energy	kWh
$E_{del,C,el}$	total fan and pumping energy	kWh

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CHAPTER 1 : INTRODUCTION

1.1. Background

Activating the thermal mass in Norway is now done by ventilation heat. The most energy efficient office, “Sparebank1” in Trondheim, is making use of this technology. Power House One is planning on using the same technology. This will be also one of the goals in this thesis. Make a comparison between heating with ventilation air and heating by embedded water pipes, TMA. Which technology creates the best thermal comfort and which technology is most energy efficient?

1.2. Case study

A case study was developed for finding out the results given in this paper. This case study was build up around a plus energy project “Power House One” that will be located in Trondheim, Norway. It will be the most northern plus energy project in the world.

The following three model rooms are used for this case study;

- Office room with 40% window
- Office room with 100% window
- Meeting room with 40% window

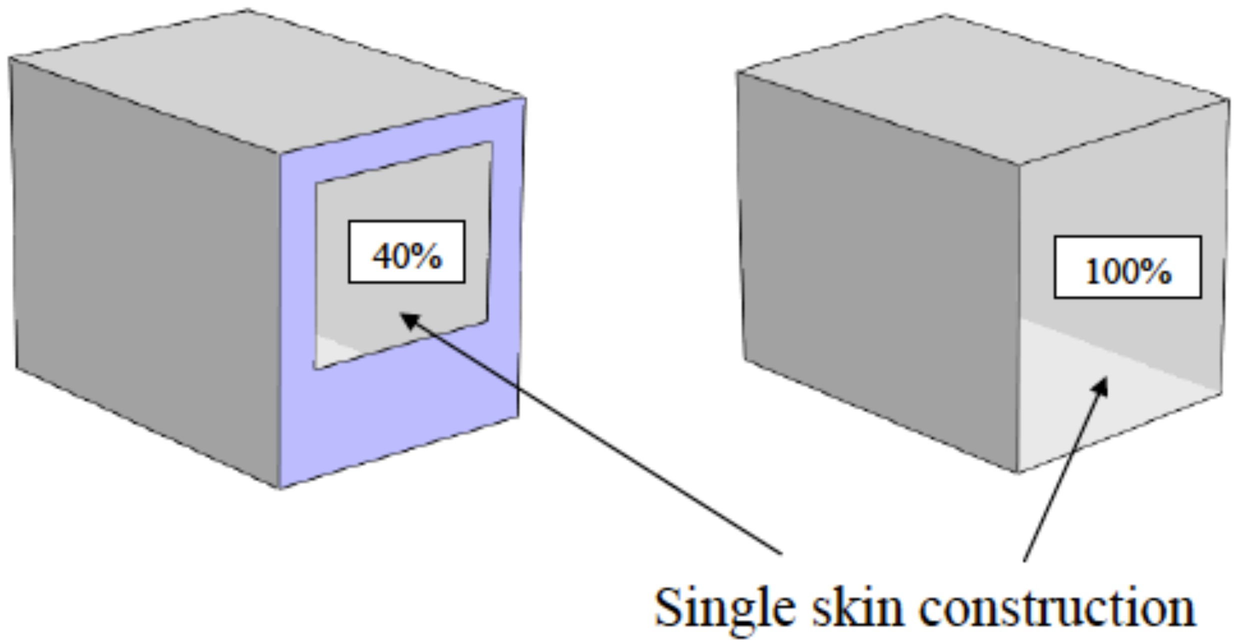


Fig.1. Office room with 40% and 100% window in Power House1

These rooms will first be simulated and analyzed without TMA in order to find the performance for the following parameters that greatly influence the indoor climate.

- Shading device;
    - External shading
    - Lower dead band and upper dead band shading
    - Yearly shading schedule for getting more effort out the passive solar gains
- Ventilation;
    - Demand controlled ventilation

The parameters of these models will be discussed in the following parameters.

1.2.1. Office room with 40% window

The following table show the parameters from an office room with 40% southwest facing windows on one wall.

PARAMETER	
Room depth (m)	3.5
Room width (m)	2.5
Room height (m)	2.7
Floor area (m^2)	8.75
Wall area facing S-W (m^2)	6.75
Window area %	40
Room volume (m^3)	23.63
Orientation	232
Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains (W/m^2)	14.43
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	20C-26C
Tour – retour temp. heat ventilation	45C – 35C
Temperature of airflow	20
Air change of ventilation (1/h)	2
Air change rate per person (m^3/h)	26
Humidity %	50

Table 1. Parameters for an office with 40% window

1.2.2. Office room with 100% window

The following table show the parameters from an office room with 100% southwest facing windows on one wall.

PARAMETERS	
Room depth (m)	3.5
Room width (m)	2.5
Room height (m)	2.7
Floor area (m^2)	8.75
Wall area facing S-W (m^2)	6.75
Window area %	100
Room volume (m^3)	23.63
Orientation	232
Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains (W/m^2)	14.43
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	21C-26C
Tour – retour temp. heat ventilation	45C – 35C
Temperature of airflow	20
Air change of ventilation (1/h)	2
Air change rate per person (m^3/h)	26
Humidity %	50

Table 2. Parameters from an office with 100% window

1.2.3. Meeting room with 40% window

The following table show the parameters from a meeting room with 40% southwest facing windows on one wall.

PARAMETERS	
Room depth (m)	5.3
Room width (m)	3.3
Room height (m)	2.7
Floor area (m^2)	17.52
Wall area facing S-W (m^2)	8.91
Window area %	40
Room volume (m^3)	47.3

Orientation	232
Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains - 8persons (W/m^2)	28.76
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	21C-26C
Tour – retour temp. heat ventilation	45C – 35C
Temperature of airflow (deg C)	19
Air change of ventilation (1/h)	3.5
Air change rate per person (m^3/h)	26
Humidity %	50

Table 3. Parameters from a meeting room with 40% window

1.3. Literature review on thermal mass activation

A literature review will be performed on the following aspects:

- Principles of thermal mass activation,
- Heat exchange between TMA, and
- Heat exchange coefficient between surface and space

This literature review will improve the understanding of thermal mass activation that will be located in the floor. Some advantages and disadvantages from TMA will also be discussed.

1.3.1. Thermal mass activation principles

The principle of Thermal Mass Activation (TMA) is based on activating the building mass by thermal energy. Activating these elements will be achieved by embedded pipes that are circulating thermal energy through the use of water. In general, thermal mass is readily available in every big building by the amount of concrete that is already a part of the building structure. A floor slab is the best place for integrating this technology. This low heating and cooling system facilitates the possible use of renewable energy sources such as solar collectors, ground source heat pump, free cooling or ground source heat exchangers. The thermal mass has high conductivity and thereby a high inertia making the response on high temperature fluctuations slow. TMA is thereby not suited for every kind of building. Office buildings usually have a predictable heating and cooling demand

and the internal heating fluctuations are low. TMA is therefore a good medium for creating good internal comfort with a stable and good indoor climate in office buildings.

Concrete has a high inertia and thereby a high thermal storage capacity. This storage capacity gives TMA the ability to release the thermal energy in a later period of the day. TMA will make use of the total mass to store the thermal energy. This is another main difference between floor heating and TMA. This issue will also be clarified in the parametric study on the time delay factor on a building with high amount thermal mass and a building with a low amount of thermal mass, see 3.2.

The criteria for TMA are according to NS-EN15377-1. The thermal mass thickness divided by the pipe spacing should be equal or higher than 0,3 and the outside diameter divided by the pipe spacing should be equal or smaller than 0,2.

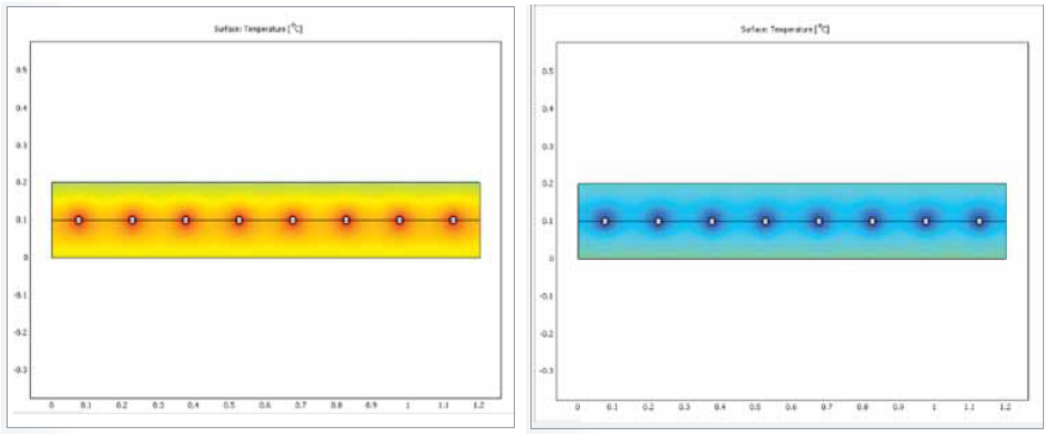


Fig.2 Picture to the left shows TMA heating and the picture to the right shows TMA cooling.(Thermac handboek)

TMA heats the total mass, the concrete, as long as the indoor temperature is not reached. Once the internal temperature gets higher than the activated mass, the purpose of TMA will reverse and it will cool the area instead of heating it. This is called the self-regulating capability of concrete and will reduce the peak times for heating and cooling.

In order to have a good heat exchange between the TMA and the zone, the thermal resistance between the thermal mass and the zone should be as low as possible. Floor covering such as carpets or parquet have a very low conductivity and thereby reduce the heat exchange between the thermal mass and the zone. The same phenomenon will occur when there is a lowered ceiling or when the ceiling is covered by an acoustic absorbing material. It is recommended to cover the thermal activated floor or ceiling with a material that has a very low thermal resistance so it will affect the heat exchange as little as possible. The best scenario is when there is no covering at all allowing the heat exchange between the thermal mass and the zone to be optimal.

The advantage of having a low covering grade, on the floor and on the ceiling, is that it will improve the internal volume resulting in more free space. This will give the architects more freedom to easily achieve a desired zone height of 2.80m – 2.90m. By leaving the extra floor height or the lowered ceiling, it will also improve the total building height by having more free height per floor. This means that by integrating TMA, the total height of the building will be lower but the building can still have the same internal volume dimensions and the same amount of floors as a higher building with heightened floors or lowered ceilings. The envelope of the building would be lower resulting in an improvement by reducing material requirements. This would not only be energy efficient but also cost effective.

Traditional Building Building with TMA

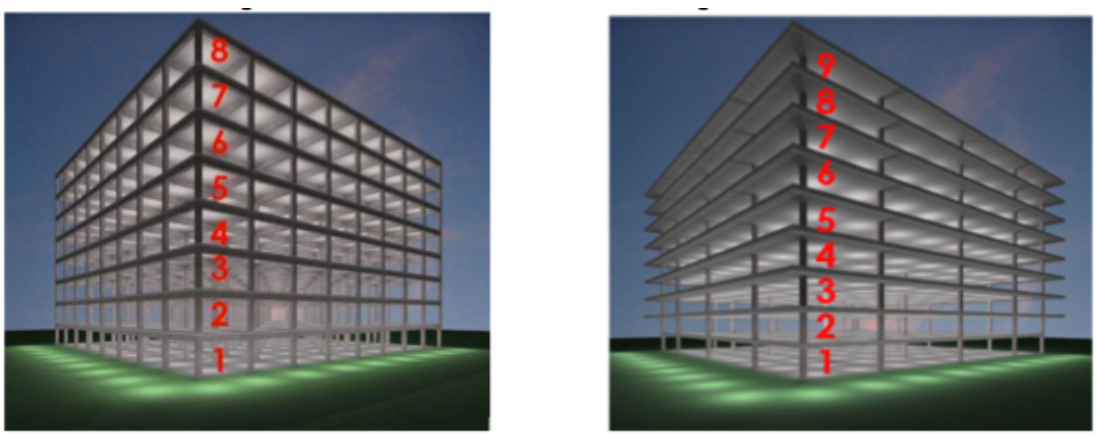


Fig.3 Building to the left has lowered ceilings and heightened floors while the building to the right makes use of TMA. Both buildings have the same height but the building to the right with TMA has an increased number of floors.(Airdeck presentation)

The main disadvantage of TMA is that it will reduce the flexibility of the building. A solution must be found for the vacant pipes, the ventilation channels and the electrical supplies. An alternative to acoustic improvements must also be found. The maintenance, the repairs and the adjustments of the electrical devices will also be more difficult. By integrating the technical devices into the thermal mass of the building, it will be less flexible but at the same time the chance that a problem will occur with embedded systems is almost zero.

Due to the reduction of the flexibility and because of the high cooling load in energy efficient buildings, the ceiling is usually the most efficient place for integrating TMA in Central Europe. The ceiling is often used for activating the thermal mass and for reducing the cooling load as the heat exchange coefficient of the ceiling for cooling is higher (See fig.4).This is due to the fact that most of the internal heating is convective and therefore the ceiling area is usually the warmest area of the internal zones.

This is the reason why the technical devices are usually placed in heightened floors. Another option is to centralise all the technical devices into the central elevator rooms and from there divide the electrical devices.

A study was conducted by Rasmus Z.Høseggen, Hans M.Mathisen and Sten O.Hansen on the effect of suspended ceilings on energy performance and thermal comfort. This study shows that exposed concrete in the ceiling both reduces the number of hours with excessive temperatures considerably and creates a better and more stable thermal environment during the working day. Exposed concrete also increases the achievements of utilizing night free cooling significantly.

1.3.2. Heat exchange coefficient between surface and space

The heat exchange q_{TAM} coefficient between surface and space is according to the following standard NS 15377. The heat and cold exchange of a floor and ceiling can be calculated by the following formula.

$$q_{TMA} = 8{,}92 * (\theta_{S,m} - \theta_i)^{1,1} \qquad (W/m^2)$$

Where;

$\theta_{S,m}$ average surface temperature in °C

θ_i nominal indoor temperature in °C

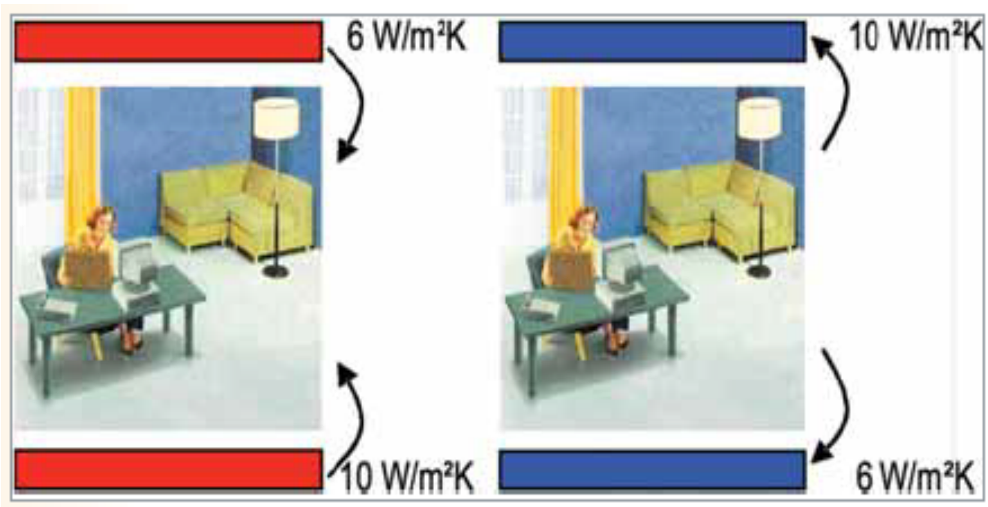


Fig.4 Most common values for heat capacity to the left and the most common values for cooling to the right. (Thermac boek)

For other types of surface heating and cooling systems, the heat flow intensity q is given by;

Ceiling heating: $q = 6 * (|\theta_{S,m} - \theta_i|)$ (W/m^2)

Floor cooling: $q = 7 * (|\theta_{S,m} - \theta_i|)$ (W/m^2)

When the temperature difference between the surface and the indoor increases, the heating or cooling capacity will also increase parallel. When the indoor temperature is higher than the surface temperature of the thermal activated slab, the slab will act as a cooling element. When the indoor temperature is lower than the surface temperature of the thermal activated slab, the slab will act as a heating element. The temperature difference cannot be too high otherwise surface condensation can occur. This must be avoided. This of course is related to the dew point temperature and the humidity level. This will be discussed in further detail in the parametric study done on the floor and ceiling in section *3.1.*

The heat and cold exchange of TMA occurs by 2/3 radiation and by 1/3 convection. A research team that published Thermac Handboek states the following;

The radiation component is influenced by;

- The surface temperature of the construction element
- The material property such as the reflection factor and the emission value
- The shape factor and the surface area

The convection component is influenced by;

- The temperature difference between the surface of the element and the indoor temperature.
- The air velocity
- The type of air stream (laminar or turbulent)
- The direction of the heat flow
- The roughness of the area
- The geometry of the ceiling

As warm air rises and cold air descends, the cooling capacity of a ceiling is higher than the heating capacity. In reverse, the floor's heat capacity will be higher than the cooling capacity. This was investigated by using a TRNSYS16 simulation (operation time 06.00-18.00). In the example of temperature profiles, a 12h operation was found to be the most efficient.

CHAPTER 2 : METHODOLOGY

2.1. Introduction

The thermal comfort and the yearly energy performance of two office rooms and one meeting room will be simulated with the dynamical simulation program TRNSYS16. The multi-zone building (TYPE 56) is used to model the office rooms and meeting room. The active elements are modeled with the built-in active layer model, which uses the RC-representation from Koschenz and Lehman [9]. This study uses the internal calculation temperature dependent heat transfer coefficients defined in TRNSYS [10]. Simulations are done according to weather data from Trondheim, Norway.

A detailed parametric study on the cooling load will be presented in the following section 3.6. The study split of the gains in a convective and radiative part is done according to ASHRAE-guidelines [8]. The time delay factor for radiation will also be integrated and analyzed to see how high the influence of re-radiation is. Two models, low thermal mass and high thermal mass, will be compared with each other and this is also done according to ASHRAE-guidelines. This study together with the heat loss can be used as guidelines for dimensioning the thermal mass.

2.2. Heat loss and cooling load

Heat loss and cooling load are two important parameters for dimensioning the TMA. The following section will give a detailed understanding of this issue.

2.2.1. Heat loss

Heat loss is calculated according to the method described in national standards EN12831 and NS3031. For the calculation and the presentation of the external design temperature, national or public bodies refer to EN ISO 15927-5. Another approach to determining the external design temperature is to use the lowest two-day mean temperatures that have been registered ten times over a twenty-year period. For these calculations, the internal gains, the external heat from the sun and the long wave radiation are set to zero. Transmission heat loss and ventilation heat loss are also separated from each other.

Transmission heat loss

Here the formula will be shown so there is a better understanding of how the results were found in the following chapter 3: parametric studies and results. The design transmission heat loss for a heated space is calculated according to the following formula:

$$\Phi_{T,i} = \left(H_{T,ie} + H_{T,iue} + H_{T,ig} + H_{T,ij}\right)\cdot\left(\theta_{\text{int},i} - \theta_e\right)$$

$H_{T,ie}$	transmission heat loss coefficient from heated space to the exterior through the building envelope	W/K
$H_{T,iue}$	transmission heat loss coefficient from heated space to the exterior through the unheated space	W/K
$H_{T,ig}$	steady state ground transmission heat loss coefficient from heated pace to the ground	W/K
$H_{T,ij}$	transmission heat loss coefficient from heated space to a neighbouring heated space at a significantly different temperature, i.e. an adjacent heated space within the building entity or a space of an adjacent building entity	W/K
$\theta_{\text{int},i}$	internal design temperature for heating	°C
θ_e	outside temperature	°C

The internal design temperature of 20°C and an outside temperature -20°C in Trondheim should be brought into account. For these models, heat transmission between adjacent rooms is set to zero because they have the same indoor climate. This inhibits heat transmission to occur so it will not influence the indoor climate. The only heat loss that must be brought into account is through the Southwest envelope. This envelope will be divided into transmission heat loss through the adjacent part $H_{T,wall}$ and the transparent/window part $H_{T,wind.}$.

$$H_{T,wall} = A_{wall}\cdot U_{wall} \qquad H_{T,wind.} = A_{wind.}\cdot U_{wind.} \qquad \text{(W/K)}$$

$$\Phi_{T,i} = \left(A_{wall}\cdot U_{wall} + A_{window}\cdot U_{window}\right)\cdot\left(\theta_{\text{int},i} - \theta_e\right)\cdot\frac{1}{A_{fl}}$$

where:

$\Phi_{T,i}$	transmission heat loss	(W/m^2)
U_{wall}	wall area	(m^2)
U_{window}	window area	(m^2)
U-value	envelope	(W/m^{2}K)
U-value	window	(W/m^{2}K)

Ventilation heat loss

The design ventilation heat loss, $\Phi_{V,i}$, for a heated space is calculated as follows:

$$\Phi_{V,i} = H_{V,i} \cdot (\theta_a - \theta_V)$$

$$H_{V,i} = \dot{V}_{V,i} \cdot \rho \cdot c_p$$

where:

$\Phi_{V,i}$	ventilation heat loss	W/m^2
$H_{V,i}$	ventilation heat loss coefficient	W/m^2
$\dot{V}_{V,i}$	ventilation mass flow rate	40m^3/Power House1
ρ	density of air	1,23kg/m^3
c_p	heat capacity of air	1003,5J/kg·K
θ_a	indoor air temperature	22°C
θ_V	temperature ventilation air	20°C

Heat recovery should also be taken into account because it has a strong influence on the amount of ventilation heat loss.

Total heat loss

The total amount of heat that is needed to compensate for this heat loss is the sum of the transmission heat loss with the ventilation heat loss.

2.2.2. Cooling load

From the previous parametric study it can be concluded that there is need for reducing and having more control over the cooling load.

The cooling load is calculated according to the method described in ASHRAE 2009. The design cooling temperature is set to 26°C. This temperature was chosen according to the requirements of category A2 given in the NS-EN 15251:2007 standard.

Cooling load principles

Cooling loads result from conductive, convective and radiative heat transfer processes throughout the building envelope and from internal sources and system components. The following are building components or phenomena that may affect cooling loads:

- External: walls, windows, skylights, doors, partitions, ceilings and floors
- Internal: lights, people and appliances
- Infiltration: air leakage and moisture migration
- System: outside air, duct leakage and heat gain, reheat, fan and pump energy, and recovery

Cooling loads in practice

Precise calculation of cooling loads is impossible in practice due to several variables. Some of these factors include variation in heat transfer coefficients of typical building materials and composite assemblies: differing motivations and skills of those who construct the building: unknown filtration rates: and the manner in which the building is actually operated.

The following figure shows that the radiated heat that is absorbed by walls, floor, furniture, etc., contributes to space cooling load only after a time lag. Some of this energy is still present in these building elements and radiated after the occupancy period.

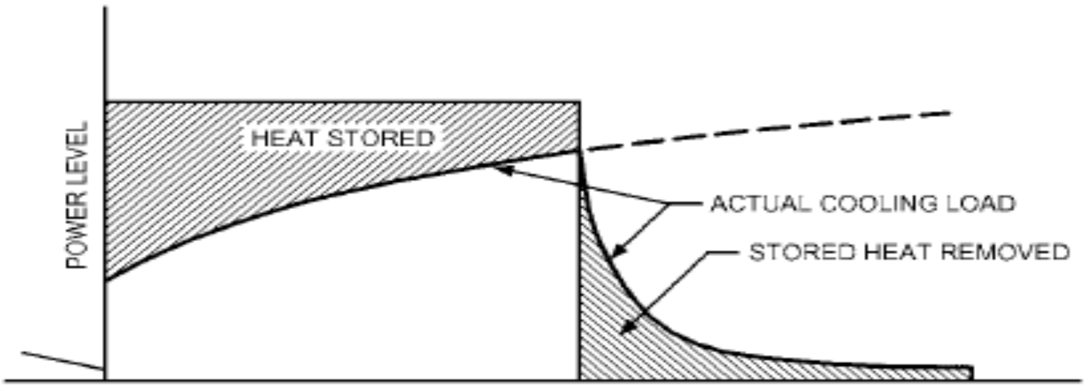


Fig.5 Thermal storage's effect on the cooling load (ASHRAE2009)

An office room with 100% window and a meeting room are analysed in this paper. These rooms face to the southwest. Here, the Radiant Time Series (RTS) method is chosen for calculating the cooling load. RTS is a simplified method for performing design-cooling calculations that is derived from the heat balance (HB) method. The RTS method is suitable for peak design load calculations, but should not be used for annual energy simulations due to its inherent limiting assumptions (ASHRAE 2009).

The following figure illustrates the RTS method for calculating the solar radiation.

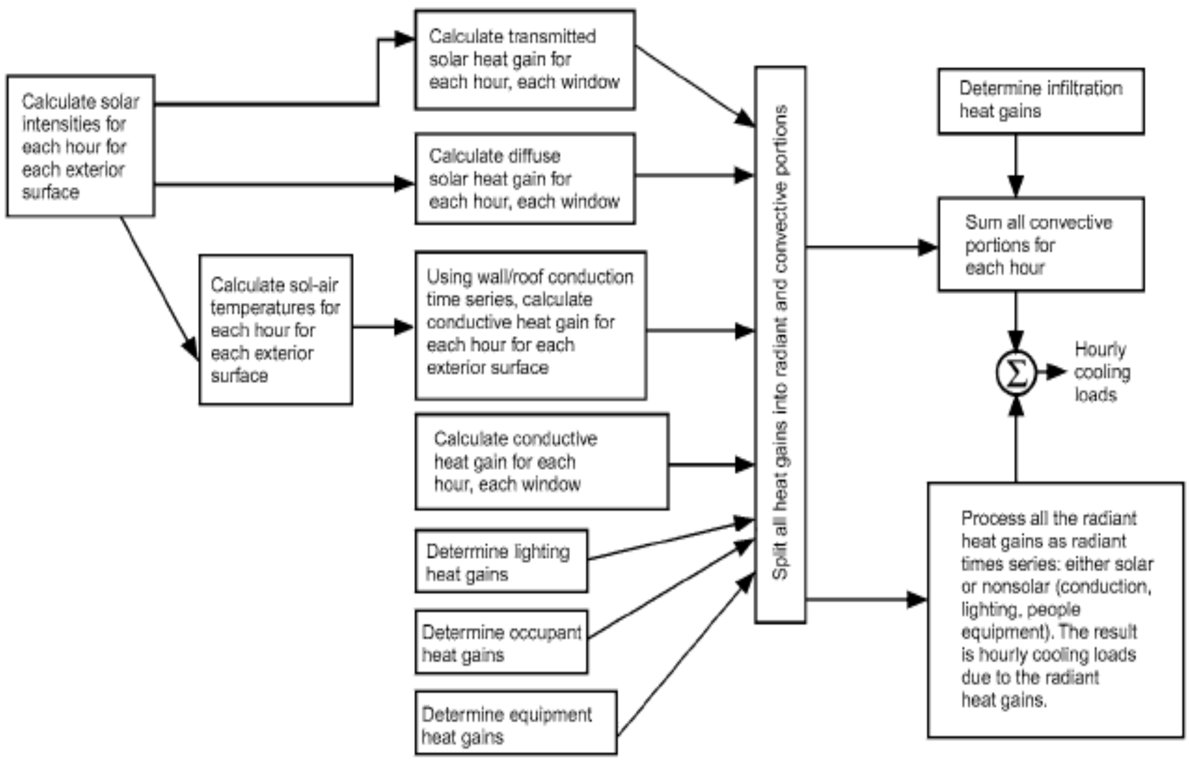


Fig. 6 Gives an overview of the RTS method. When calculating solar radiation, transmitted solar heat gain through the windows, sol-air temperature, and infiltration (ASHRAE 2009).

The following parameters are calculated in this method:

1) Internal gains for an office room and a meeting room,
2) Heat gain through the opaque envelope and
3) Heat gain through the transparent envelope, the window.

Radiant time factors of internal gains

The time lag before the absorbed heat is re-radiated is accounted for by integrating a radiant time factors.

The cooling load for each load component in a particular hour is the sum of the convective portion of the heat gain for that hour plus the time-delayed portion of radiant heat gains for that hour and the previous 23 hours. The convective portion of the heat gain immediately becomes cooling load. The radiative portion must first be absorbed by the finishes and mass of the interior room surfaces, and becomes cooling load only when it is later transferred by convection from those surfaces to the ambient air in the room. Radiant heat gains thus become cooing loads over a delayed period of time (ASHRAE 2009). The RTS converts the radiant portion of hourly heat gains to hourly cooling loads using radiant time factors, which are the coefficients of the radiant time series. Radiant time factors are used to calculate the cooling load for the current hour on the basis of current and past heat gains.

The following table show the radiant time factor for a high thermal mass building with 50% window area and a high thermal mass building with 90% window area. These values will be taken into account for the parametric study performed on the office with 40% window area and for the office with 90% window area, the values are according to ASHRAE guidelines. ASHRAE 2009 states that it is not possible to account for 100% window area due to the fact that a percentage of frameworks always need to be accounted for.

	Heavy weight 50% window	Heavy weight 90% window
Hour	Radiant time factor (%)	Radiant time factor (%)
24	2%	2%
23	2%	2%
22	2%	2%
21	2%	2%
20	2%	2%
19	2%	2%
18	2%	2%
17	2%	2%
16	2%	2%
15	25%	28%
14	9%	9%
13	6%	6%
12	5%	5%
11	5%	4%
10	4%	4%
9	4%	4%
8	4%	4%
7	3%	3%
6	3%	3%
5	3%	3%
4	3%	3%
3	3%	2%
2	3%	2%
1	2%	2%
Total:	100%	100%

Table 4. Radiant time factors for a high thermal mass construction with 50% and 90% window according to ASHRAE 2009 guidelines

In order to obtain a clear understanding how strong the influence of thermal mass is, it is recommended that a parametric study on a low thermal mass building be performed. For a better understanding of how these factors contribute in the calculations of re-radiation, see chapter 3.

Heat gain through the opaque surface

The heat gain through the opaque building envelope is low due to the high insulation level of this element. To calculate the amount of heat that enters the building it is important to first calculate the outside surface temperature. The orientation of the surface to the north, east, south or west, is important due to the influence of the solar gains. The building façade will be clad in wood. The sun exposure façades are also covered by photovoltaic (PV) cells to provide the Power House One building with electricity. The calculation of the following equation is applied to a southwest facing vertical surface $\theta_{s,H'}$:

$$\theta_{Surf.T} = \theta_{T,outs.} + E_t \cdot \frac{\alpha}{h_o}$$

where:

$\theta_{T,outs.}$	outside temperature	°C
E_t	total solar radiation incident on surface (TRNSYS)	W/m^2
α	absorption of cladding, wood 0,41	
h_o	coefficient of heat transfer by long-wave radiation and convection at outer surface	W/m^2K

Heat conduction through the envelope $q_{envelope}$ for every hour by

$$q_{outside} = U_{evelope} \cdot A_{envelope} \cdot (\theta_{Surf,T} - \theta_{IndT})$$

where:

$U_{envelope}$	heat transfer coefficient of the building envelope	W/m^2K
$A_{envelope}$	surface envelope	m^2

Fenestration heat gain and heat loss according to ASHRAE

For spaces with neutral or positive air pressurization, the primary climate-related variable affecting cooling load is solar radiation. These solar gains can be classified as direct, diffuse and conductive gains. The sum of these is the total fenestration heat gain, Q:

$$Q = q_b + q_d + q_c$$

For the total fenestration heat gain, the following equations are used:

Direct beam solar heat gain, q_b:

$$q_D = A \cdot E_{t,b} \cdot SHGC\theta$$

Diffuse solar heat, q_d:

$$q_d = A \cdot \left(E_{t,d} \cdot E_{t,r}\right) \cdot \langle SHGC \rangle_d$$

Conductive heat gain q_c:

$$q_c = U \cdot A \cdot \left(T_{out} - T_{in}\right)$$

where:

A	window area	m^2
$E_{t,b}, E_{t,d}$ and $E_{t,r}$	beam, sky diffuse, and ground-reflected diffuse irradiance	
$SHGC(\theta)$	beam solar heat gain coefficient as a function of incident angle θ	
$\langle SHGC \rangle_D$	diffuse solar heat gain coefficient	
T_{in}	indoor temperature	°C
T_{out}	outdoor temperature	°C
U	overall U-factor, including the frame	W/m^2K
$IAC(\theta.\Omega)$	indoor solar attenuation coefficient for beam solar heat gain coefficient; 1.0 if no inside shading device. is a function of shade type and, depending on type, may also be a function of beam solar angle of incidence θ and shade geometry	
IAC_D	indoor solar attenuation coefficient for beam solar heat gain coefficient; 1.0 if no inside shading device. is a function of shade type and, depending on type, may also be a function of and shade geometry	

The external shading device should also be considered. The shading device will reduce the direct and diffuse solar gains by 80% on the envelope surface facing to the southwest.

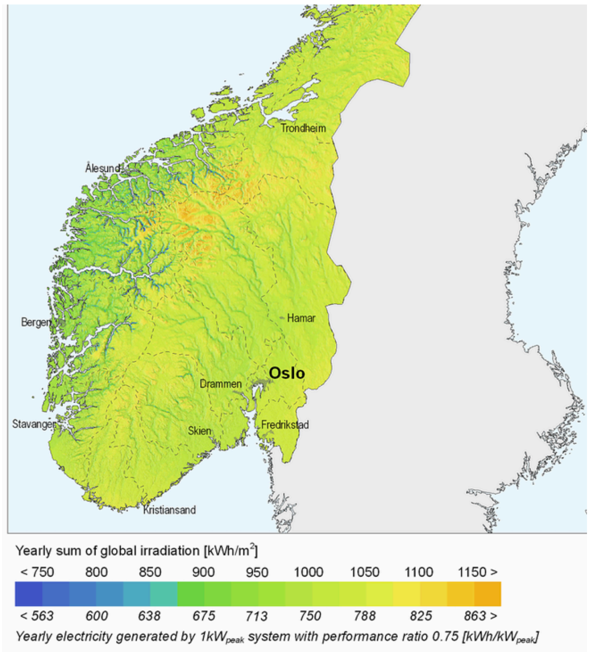


Fig.7 Trondheim is situated at 63°36'N, 10°25'E, where the sun provides 1000W/m2

2.3. Dimensioning of thermal mass activation

For dimensioning the thermal mass activation, the NS-EN15377 standard will be used. This will be also be related to the parametric study of the Power House 1 that is situated in Trondheim. Here the principles and equations are shown here but the eventually calculations are integrated in the tables that will be shown in chapter 3.

2.3.1. Mass flow rate and accessories

First it will be necessary to calculate the flow rate $\dot{m}_H$. There is a heat difference ΔT assumed from proximately 2K (TRNSYS) in the water circuit. This difference is between the input and the output temperature of the water circuit. If a higher mass flow rate would be used then the water temperature would be less high but the pumps need to have a higher dimension. With a smaller mass flow rate the temperature difference would increase and thereby will the heat exchange decrease when it is reaching the end of the pipe. The following formula is for calculating the flow rate.

$$\dot{m}_H = \frac{A \cdot q_k}{\Delta T \cdot C_p}$$

where;

A	heated of cooled surface	(m^2)
q_k	heating or cooling capacity	(W/m^2)
$\Delta\theta$	temperature difference of the fluid loop	(K)
C_p	specific heating capacity of water	(J/kgK)

2.3.2. Energy demand pump

Than the calculation of the energy demand for the pump E_p can be made. This will be according to NS3031:2007.

$$E_P = \dot{m}_H \cdot SPP \cdot t_{dr} \qquad \text{(kWh)}$$

where;

$\dot{m}_H$	flow rate	(l/s)
SPP	specific pump power	(kW/l/s)
t_{dr}	running time	(h)

2.3.3. Calculation of heat transfer coefficient U_{wrx}

The following calculation will prove the correctness of the difference between the mean fluid temperature and the mean surface temperature. If TRNSYS is used as a simulation tool than the heat transfer coefficient U_{wrx} will be calculated. U_{wrx} can also be calculated according to the following equation (5).

$$\Delta T = \frac{P_{sp\_norm}}{U_{wrx}} \qquad \text{(K)}$$

where;

P_{sp_norm}	specific norm power	(kJ/hm^2)
U_{wrx}	heat tranfer coefficient	(kJ/hm^2K)

$$U_{wrx} = \frac{1}{R_w + R_r + R_x} \qquad (W/m^2K)$$

where;

R_w	thermal resistance fluid to pipe	(m^2K/W)
R_r	thermal resistance pipe	(m^2K/W)
R_x	thermal resistance x-direction	(m^2K/W)

2.3.4. Calculation of resistance of the pipe

The resistances are also important for calculating the power like shown above. The resistance between the outer wall temperature of the pipe and the surface temperature of the thermal mass surface will also be important to bring into account. This has to be calculated for floor heating $R_{Fl,H}$ as for ceiling heating $R_{C,H}$ and floor cooling $R_{Fl,Co}$ as for ceiling cooling $R_{C,Co}$.

$$R_x = \frac{T \cdot \ln\left(\frac{T}{\pi - d_a}\right)}{2 \cdot \pi \cdot \lambda_r} \qquad R_r = \frac{T \cdot \ln\left(\frac{d_a}{d_a - 2 \cdot s_r}\right)}{2 \cdot \pi \cdot \lambda_r} \qquad R_w = \frac{T^{0,13}}{8 \cdot \pi}\left(\frac{d_a - 2 \cdot s_r}{\dot{m}_{H.Sp} \cdot L_R}\right)^{0,87}$$

Two conditions shall be fulfilled for application of these equations:

- Equation for R_x is valid only if $s_1 \mid T > 0{,}3$, $s_2 \mid T > 0{,}3$ and $d_a \mid T < 0{,}2$
- Equation for R_z is valid only if $\dot{m}_{H.Sp} \cdot C_w \cdot (R_w + R_r + R_x) \geq \frac{1}{2}$

Data referred to U_{wrx} Ceiling

U_{wrx}	heat transfer coefficient ceiling	(kJ/hm^2K)
T	pipe spacing	(m)
$\dot{m}_{H.Sp}$	specific flow rate	(kg/hm^2)
d_a	external diameter of pipe	(m)
s_r	thickness of pipe wall	(m)
λ_r	thermal conductivity pipe wall	(W/mK)
L_r	length of circuit	(m)

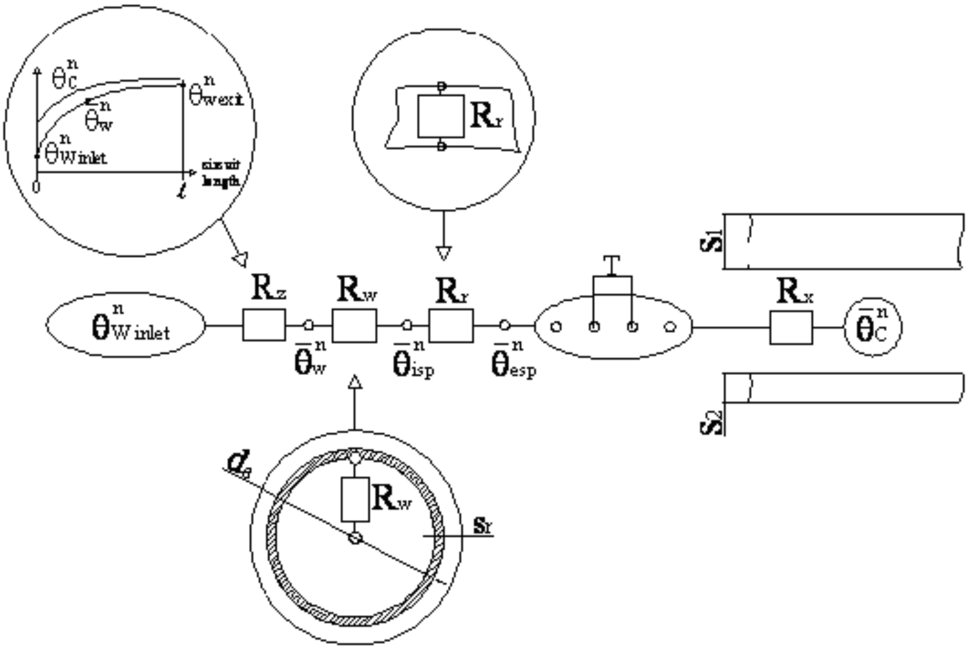


Fig.8 General scheme of the resistance method (NS-EN15377-3:2007)

Heating: $R_{Fl,H}$ and $R_{Ceil,H}$ (m^2K/W)

$$R_{Fl,H} = \frac{1}{h_{Fl,H}} + R_{Concr.Fl} = 0{,}136 \qquad R_{Fl,H} = \frac{1}{h_{Ceil,H}} + R_{Concr.Ceil} = 0{,}203$$

Cooling: $R_{Fl,Co}$ and $R_{Ceil,Co}$.(m^2K/W)

$$R_{Fl,Co} = \frac{1}{h_{Fl,Co}} + R_{Concr.Fl} = 0{,}203 \qquad R_{Ceil,Co} = \frac{1}{h_{Ceil,Co}} + R_{Concr.Ceil} = 0{,}136$$

$h_{Fl,H}$	heat capacity floor heating (see fig.4)	(W/m^2K)
$h_{Ceil,H}$	heat capacity ceiling heating (see fig.4)	(W/m^2K)
$h_{Fl,Co}$	heat capacity floor cooling (see fig.4)	(W/m^2K)
$h_{Ceil,Co}$	heat capacity ceiling cooling (see fig.4)	(W/m^2K)
$R_{Concr.Fl.}$	concrete on top of the pipe (6cm, S1 Fig.8)	(m^2K/W)
$R_{Concr.Ceil.}$	concrete under the pipe (6cm, S2 Fig.8)	(m^2K/W)

2.3.5. Calculation power needed for cooling and heating by active elements

Because the space above and the space under each TMA floor has the same indoor temperature. We can calculate the delivered power for the floor and the ceiling. The heating capacity of the floor is the same as the cooling capacity of the ceiling. See Figure 3. The delivered cooling power q_k will be calculated according to the following formula.

$$q_k = \frac{1}{R_{TMA}} \cdot \left(\bar{\theta}_{m,k} - \bar{\theta}_{i,k}\right) \qquad \text{(W/m}^2\text{)}$$

$$R_{TMA} = R_w + R_r + R_x + \frac{R_{Fl.Co} \cdot R_{Ceil.Co}}{R_{Fl.Co} + R_{Ceil.Co}} \qquad \text{(m}^2\text{K/W)}$$

Where;

R_w	thermal resistance fluid to pipe	(m²K/W)
R_r	thermal resistance pipe	(m²K/W)
R_x	thermal resistance x-direction	(m²K/W)
$R_{Concr.Fl.}$	concrete on top of the pipe (6cm, S1 Fig.10)	(m²K/W)
$R_{Concr.Ceil.}$	concrete under the pipe (6cm, S2 Fig.10)	(m²K/W)
$\bar{\theta}_{m,k}$	surface temperature from medium	(°C)
$\bar{\theta}_{i,k}$	design indoor temperature	(°C)

2.3.6. Design temperature for water in the pipes for cooling

The water temperature in pipes is calculated according to following equations;

$$\bar{\theta}_{m,Co.} = q_{Co.} \cdot R_{eq,Co.} + \theta_{i,Co.}$$

because of;

$$\frac{\sigma_{Co.}}{\theta_{i,Co.} - \theta_{m,Co}} \leq 0{,}5$$

According to prEN15377 the supply temperature θ_s can been calculated by the following formula;

$$\theta_{s,Co.} = \bar{\theta}_{m,Co.} + \frac{\sigma_{Co.}}{2}$$

where;

$\bar{\theta}_{m,Co.}$	design water temperature for cooling	(°C)
$q_{Co.}$	cool capacity	(W/m^2)
$R_{eq,Co.}$	resistance concrete layer	(m^{2}K/W)
$\theta_{i,Co.}$	design indoor temperature cooling	(°C)
$\sigma_{Co.}$	temperature difference water inlet outlet	(°C)
$\theta_{s,Co.}$	supply water temperature cooling	(°C)

By making sure that the temperature are not reaching the benchmark temperatures. The lowest temperature will be on the ceiling because the ceiling is used to cool the zone during the summertime. This will be done as control for avoiding surface condensation, and this must be avoided;

$$\theta_{s2,Co.} = \frac{q_{Ceil,Co.}}{h_{Ceil,Co.}} + \theta_{i,Co.} = \frac{1}{h_{Ceil,Co.}} \cdot \left(\frac{\bar{\theta}_{m,k} - \theta_{i,Co}}{R_{eq,Co}} \frac{R_{Fl,Co}}{R_{Fl,Co} + R_{Ceil,Co}} \right) + \theta_{i,Co}$$

The temperatures need to stay under the boundary temperatures for avoiding surface condensation. With a relative humidity from 60% and an indoor temperature of 25°C will there not occur condensation. According to (Hens,2003b) this will occur by a temperature of 16,7°C.

2.3.7. Design temperature for water in the pipes for heating

For the design calculation of the heating will the same mass flow rate been used. Temperature difference of the water will be;

$$\sigma_H = \frac{q_H \cdot A_F}{m_H \cdot C_H}$$

The design temperature will be;

$$\bar{\theta}_{m,H} = q_k \cdot R_{eq,H} + \theta_{i,H}$$

with;

$$R_{TMA} = R_w + R_r + R_x + R_z$$

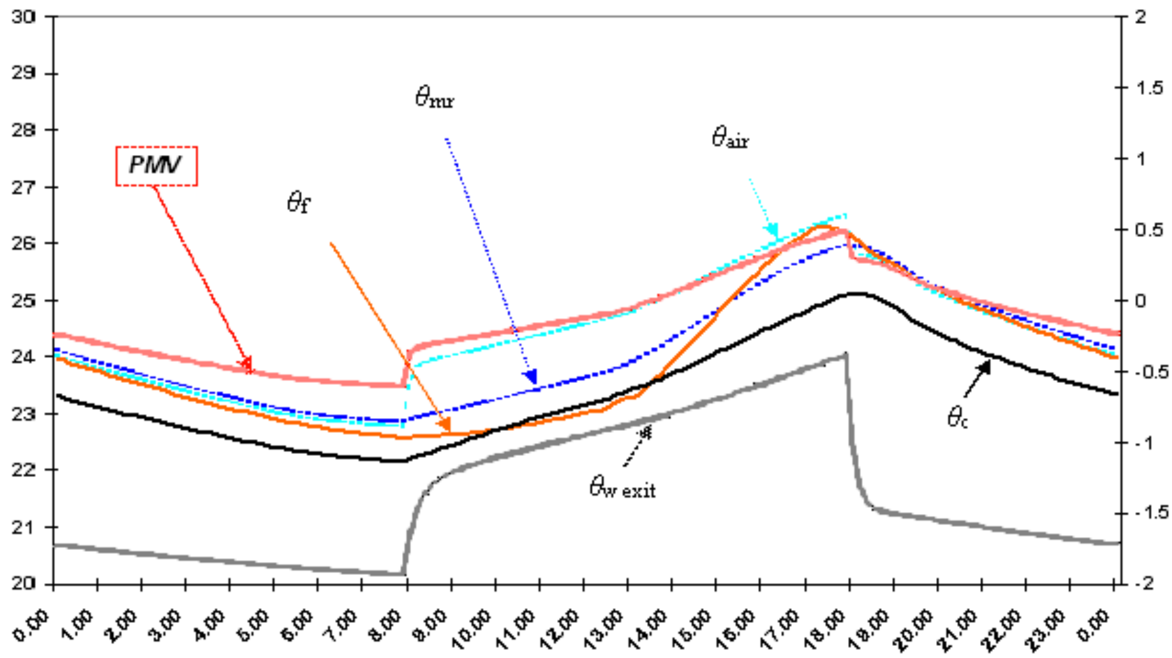
because of;

$$\frac{\sigma_H}{\theta_{m,H} - \theta_{i,H}} \leq 0,5$$

Can according to prEN15377 the supply temperature θ_s been calculated by the following formula;

$$\theta_{s,H} = {}_{m,H} + \frac{\sigma_H}{2}$$

2.3.8. Example of temperature profiles by NS-EN 15377-3:2007



Key

θ_{mr}	mean radiant temperature	X-axis	time of the day
θ_{air}	air temperature	Y-axis, left	temperature °C
θ_f	floor temperature	Y-axis, right	PMV values
θ_c	ceiling temperature		
$\theta_{w\,exit}$	return water temperature		
PMV	Predicted Mean Vote		

Fig. 9 Shows the example of temperature profiles

The diagram shows an example of the relation between internal heat gains, water supply temperature, heat on the room side, hours of operation and heat transfer on the water side. The diagram correspond to a concrete slab with a raised floor (R=0,45m^{2}K/W) and a permissible room temperature range of 21°C to 26°C.

In the upper diagram, the y-axis shows the maximum permissible total heat gain in the space (internal gains plus solar gains) in W/m^2, and on the x-axis represents the required water supply temp. in °C. The lines in the diagram correspond to different hours of operation (8hrs., 12hrs., 16hrs., 24hrs.) and different maximum amounts of energy supplied in Wh/m^2 per day. The lower the diagram shows the cooling power in W/m^2 required on the water side (for dimensioning of the chiller) for thermo-active slabs as a function of supply water temperature and operation time. Further, the amount of energy rejected per day is indicated in Wh/m^2 per day.

The example shows that by having a maximum internal gain of $38W/m^2$ and a 8h operation, a supply water temperature of 18,2°C is required. If, instead, the system is in operation for 12h, a supply water temperature of 19,3°C is required. In total, the amount of energy rejected from the room is approximately $335Wh/m^2$ per day. The required cooling power on the waterside is for a 8h operation is $37W/m^2$ and for a 12h operation it is only $25W/m^2$ per day. Thus, by having a 12h operation, the size of the chiller can be reduced significantly. The total heat rejection on the waterside is approximately $300Wh/m^2$ per day.

2.3.9. TMA in relation to renewable energy

Like mentioned in the principles of TMA. TMA is a low-temperature heating system and a high-temperature cooling system that facilitates the possible use of renewable energy sources such as solar collectors, ground source heat pump, free cooling or ground source heat exchangers.

In thesis the following renewable sources will be discussed but not to far in detail.

- Heat pumps used for heating the project
- Free cooling used for cooling the project

Heat pump

Free cooling is an economical method of using low external air temperatures to assist in chilling water, which can than be used for industrial process, or air conditioning systems. When the ambient air temperature drops to a set temperature, a modulating valve allows all or part of the chilled water to by-pass an existing chiller and run through the Free Cooling system, which uses less power and uses the lower ambient air temperature to cool the water in the system. This can be achieved by installing an air blast cooler with any existing chiller or on its own. During low ambients a processor can by-pass an existing chiller giving energy savings of up to 75%, without compromising cooling requirements (Kelly, David W. “Free Cooling Considerations”,1996).

In this thesis a COP of 4 was used and this was according to a study done by Mark Murphy, 2010. He did a study on thermo-active ceilings and free cooling. This study points out that by an amount of $60W/m^2$ internal gains the coefficient of performance can raise to 19.7.

The coefficient of performance was calculated according to the following equation (NS3031).

$$COP_C = \frac{Q_{C,nd}}{E_{del,C.el}}$$

where;

$Q_{C,nd}$	total cooling energy	(kWh)
$E_{del,C.el}$	total fan and pumping energy	(kWh)

The extra pumping power required to circulate the water through the system is roughly approximated to $2kWh/m^2$, Mark Murphy 2011. For more detailed information see “Project report 52, 2010”.

CHAPTER 3 : PARAMETRIC STUDIES AND RESULTS

The thermal comfort and the annual energy performance of two office rooms and one meeting will be simulated with the dynamical simulation program TRNSYS16. The multi-zone building (TYPE 56) is used to model the offices and meeting. The active elements are modeled with the built-in active layer model, which uses the RC-representation from Koschenz and Lehman [9]. This study uses the internal calculation temperature dependent heat transfer coefficients defined in TRNSYS [10]. Simulations are done according to weather data from Trondheim, Norway.

Several parametric studies have contributed to this thesis. These studies were according to the three models section 2.2. Thermal mass activation will be installed in three building element schemes in the model in order to find the best installation solution for this technology. The three schemes for integrating this technology are:

- Ceiling
- Floor
- Ceiling and floor

The following parametric studies were performed to determine the best performance for thermal mass activation:

- Heat exchange coefficient between surface and space, floor and ceiling
- Thermal storage capacity of concrete
- Shading, external shading device
- Activation and deactivation of the external shading
- Annual schedule for an external shading device in the office with 100% window
- Mass flow rates for the embedded pipes in relation to ventilation
- Radiant time factor
    - Heat gain through opaque surface
    - Heat gain through fenestration
- Cooling load
- Coefficient of performance (COP)

3.1. Heat exchange coefficient between surface and space, floor and ceiling

This study will show that a ceiling is more efficient for cooling and that a floor is more efficient for heating.

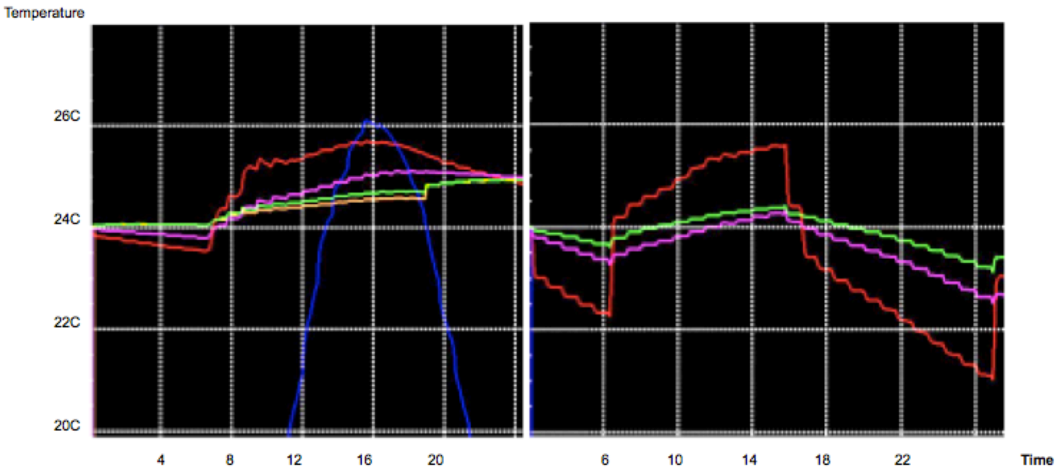


Fig.12 The figure to the left shows a simulation during the summer and the figure to the right shows a simulation during the winter (approximately one day).

From the simulation results, one can see that during the summer, the surface temperature of the floor (pink) is closer to the ambient indoor temperature (red) than the surface temperature of the ceiling (green). During the winter the opposite occurs. This is due to fact that during the summer, the ceiling acts as a cooler, which is why the surface temperature of the ceiling is lower than the surface temperature of the floor. This is shown in the following calculations.

Heat Capacity NS-EN 15377				
Qdes = 8.92(ISurface T. - Indoor T.I)^1.1				
Heat transfer coefficient W/m^2			8.92	
Summer: Warmest moment in the summer: 5493 – 5549h (TRNSYS)				
Winter: Coldest moment in the winter: 242 – 298h (TRNSYS)				
	Temperature Difference Surface – Indoor			
		Surface	**Indoor**	
Summer	Floor	20.21	20.6	
	Ceiling	20.18	20.6	
Winter	Floor	20.15	20.8	
	Ceiling	20.2	20.8	
			Total heat/cool capacity Office (W)	
	Summer	**Winter**	**Summer**	**Winter**
Floor (W/m^2)	3.17	5.55	28.24	49.54
Ceiling (W/m^2)	3.44	5.09	30.64	45.36

Table 5. Values extracted from figure 10.

Conclusion

By calculating the heat capacity according to the NS-EN15377 standard, it can be stated that the floor has a higher heat capacity during the winter while during the summer, the ceiling has a higher cooling capacity.

3.2. Thermal storage capacity of concrete

This study will show that a large amount of thermal mass will reduce the amount of heat required in the morning for returning the indoor climate to a temperature that provides good thermal comfort. This is done by taking into account the internal gains that will act as a cooling load.

The cooling load for each load component in a particular hour is the sum of the convective portion of the heat gain for that hour and the time-delayed portion of radiant heat gains for that hour and the previous 23 hours. The convective part of heat gain immediately becomes cooling load. The radiative part must first be absorbed by the interior finishes and mass of the room surfaces, and becomes cooling load only when it is later transferred by convection from those surfaces to the room air. Thus, radiant heat gains become cooing loads after a delayed period. The RTS converts the radiant portion of hourly heat gains to hourly cooling loads using radiant time factors, which are the coefficients of the radiant time series. Radiant time factors are used to calculate the cooling load for the current hour using the current and past heat gains (ASHRAE).
The following table shows how much of the internal gains is radiated heat and how much is convective heat.

Initial internal gains

	Loads (W/m^2)			
	Office room	**Meeting room**	**Radiant %**	**Convective %**
Person	4.57	18.26	60	40
Artificial light	3	3	95	5
Appliances	6.85	7.5	25	75
Total	14.42	28.76	60	40

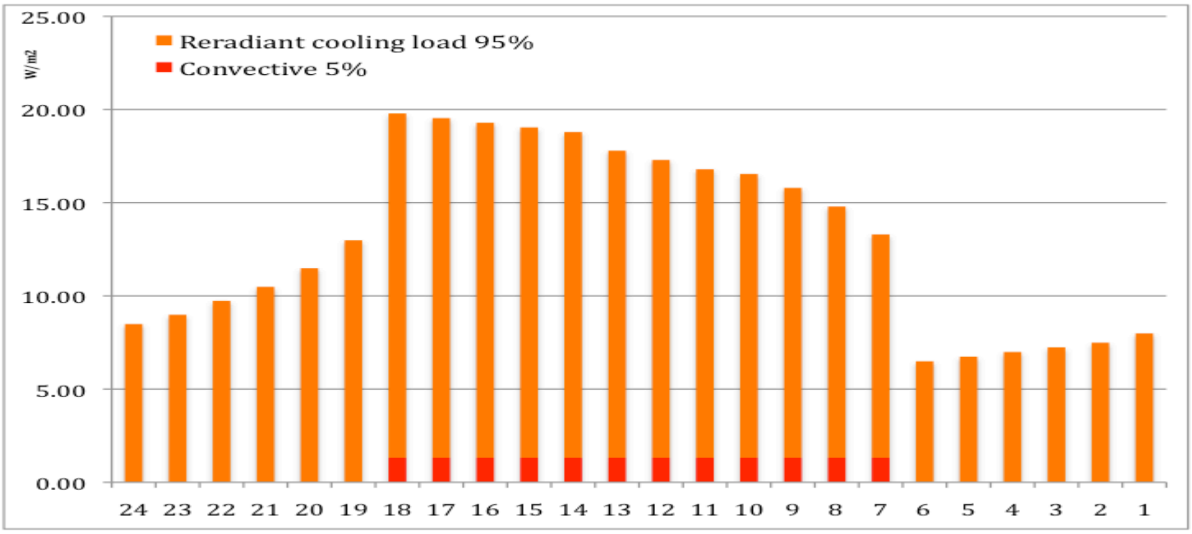
Table 6. Internal heat gains divided into radiant and convective heat gains

Artificial lighting in the office with a floor area of 8.75m^2 and 40% is chosen for showing this calculation because it has a high radiant portion (see fig.13).

3.2.1. High thermal mass construction

First, a construction with a high thermal mass is chosen for this study. This construction follows the guidelines of ASHRAE 2009. The construction is classified as a heavy construction with 50% window and no carpet. As discussed in the principles of TMA, the lack of carpet is the best set-up for promoting heat exchange between the thermal mass and ambient air.

Figure 13 shows the influence from heat gains of lighting in a high thermal mass construction.



		Lighting (Watt)					
Hour	Occup. time %	Lighting	Convective 5%	Radiant 95%	time factor (%)	Reradiant cooling load	Total sensible cooling load
24	0%	0	0	0.00	2%	7.99	7.99
23	0%	0	0	0.00	2%	7.49	7.49
22	0%	0	0	0.00	2%	7.24	7.24
21	0%	0	0	0.00	2%	6.99	6.99
20	0%	0	0	0.00	2%	6.74	6.74
19	0%	0	0	0.00	2%	6.49	6.49
18	100%	26.28	1.314	24.97	2%	11.98	13.29
17	100%	26.28	1.314	24.97	2%	13.48	14.79
16	100%	26.28	1.314	24.97	2%	14.48	15.79
15	100%	26.28	1.314	24.97	25%	15.23	16.54
14	100%	26.28	1.314	24.97	9%	15.48	16.79
13	100%	26.28	1.314	24.97	6%	15.98	17.29
12	100%	26.28	1.314	24.97	5%	16.48	17.79
11	100%	26.28	1.314	24.97	5%	17.48	18.79
10	100%	26.28	1.314	24.97	4%	17.73	19.04
9	100%	26.28	1.314	24.97	4%	17.98	19.29
8	100%	26.28	1.314	24.97	4%	18.23	19.54
7	100%	26.28	1.314	24.97	3%	18.47	19.78
6	0%	0	0	0.00	3%	12.98	12.98
5	0%	0	0	0.00	3%	11.48	11.48
4	0%	0	0	0.00	3%	10.49	10.49
3	0%	0	0	0.00	3%	9.74	9.74
2	0%	0	0	0.00	3%	8.99	8.99
1	0%	0	0	0.00	2%	8.49	8.49
Total:		315.36	15.77	299.59	1.00	298.11	313.88

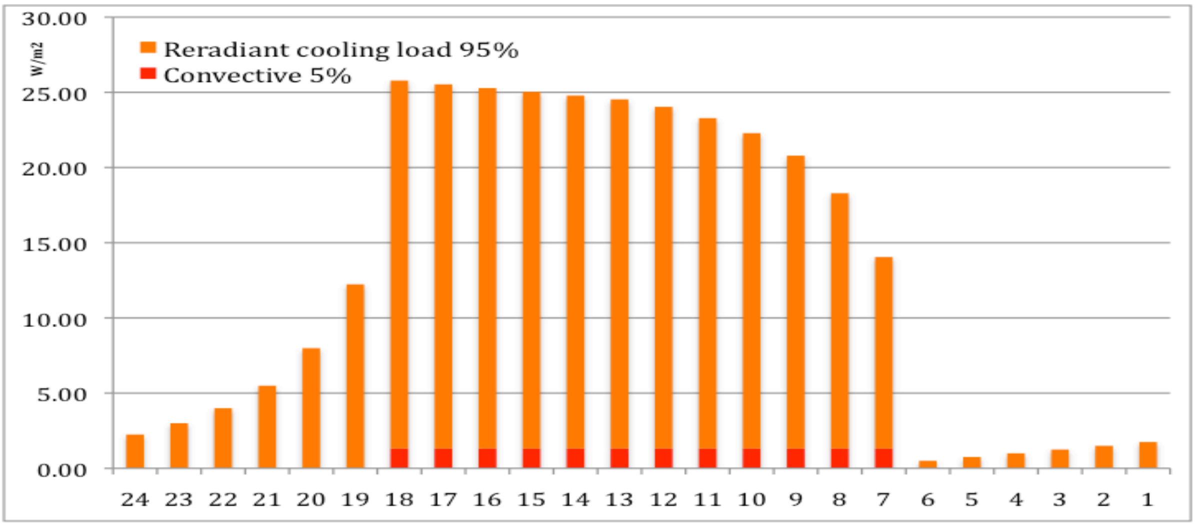
Fig13. Calculation with the radiant time factors of a heavy construction

Figure 13 shows the cooling load calculations for lighting with 5% convective heat gains and 95% radiated heat gains. The calculation is done for a room that is classified (ASHRAE 2009) as a heavy construction with no carpet and 50% glass. The heavy construction has a strong influence on the time delay factor and of the re-radiated heat gain, which is shown in the table and the diagram. The re-radiated cooling load is high. See the attachment "detailed cooling load calculation" for more detailed information on the calculations. The sensible cooling load is the sum of re-radiated values and the convective values. This sum can be compared with total lighting gains, which, in turn ,is the sum of convective values and radiant values.

3.2.2. Low thermal mass construction

The results of the office with 40% window can be compared with those of an identical construction but with a small amount of thermal mass. This will give a clear understanding of the importance and influence of the thermal mass present in a construction. Section 3.6 will summarize the cooling load and include the solar gains.

Figure 14 shows the influence from heat gains of lighting in a low thermal mass construction.



		Lighting (Watt)					
Hour	Occup. time %	Lighting	Convective 5%	Radiant 95%	Radiant time factor	Reradiant cooling load	Total sensible cooling load
24	0%	0.00	0.00	0.00	0%	2.25	2.25
23	0%	0.00	0.00	0.00	0%	3.00	3.00
22	0%	0.00	0.00	0.00	0%	3.99	3.99
21	0%	0.00	0.00	0.00	0%	5.49	5.49
20	0%	0.00	0.00	0.00	0%	7.99	7.99
19	0%	0.00	0.00	0.00	0%	12.23	12.23
18	100%	26.28	1.31	24.97	0%	24.47	25.78
17	100%	26.28	1.31	24.97	0%	24.22	25.53
16	100%	26.28	1.31	24.97	0%	23.97	25.28
15	100%	26.28	1.31	24.97	50%	23.72	25.03
14	100%	26.28	1.31	24.97	18%	23.47	24.78
13	100%	26.28	1.31	24.97	10%	23.22	24.53
12	100%	26.28	1.31	24.97	6%	22.72	24.03
11	100%	26.28	1.31	24.97	4%	21.97	23.28
10	100%	26.28	1.31	24.97	3%	20.97	22.29
9	100%	26.28	1.31	24.97	2%	19.47	20.79
8	100%	26.28	1.31	24.97	1%	16.98	18.29
7	100%	26.28	1.31	24.97	1%	12.73	14.05
6	0%	0.00	0.00	0.00	1%	0.50	0.50
5	0%	0.00	0.00	0.00	1%	0.75	0.75
4	0%	0.00	0.00	0.00	1%	1.00	1.00
3	0%	0.00	0.00	0.00	1%	1.25	1.25
2	0%	0.00	0.00	0.00	1%	1.50	1.50
1	0%	0.00	0.00	0.00	0%	1.75	1.75
Total:		315.36	15.77	299.59	100.00%	299.59	315.36

Fig 14. Calculation with the radiant time factors of a heavy construction

Figure 14 shows a short time delay and the re-radiated heat values are low. In the heavy weight building (see fig.13), the time delay of absorbed heat gains is long and the re-radiated heat values are high. This is mainly due to the difference in the time delay factor. The total amount of the sensible cooling loads is almost the same. At 6am, there is just 1 W/m^2 floor area in the lightweight building whereas in the heavy weight building, 7 W/m^2 is still available.

Conclusion

A lightweight construction needs double the amount of heat gain to reach the same indoor climate as a heavyweight construction. This leads to the conclusion that in a heavy weight building, that is, a building with high thermal mass, it is easier to keep the temperature stable than in a lightweight building. This is also the reason why office buildings are better suited to having a high thermal mass than a normal house. This fact is also directly related to the occupancy of both buildings types. Since a light weight building, such as a wooden house, is usually unoccupied during the day, there is no need for re-radiated heat from the construction.

3.3. Shading, external shading device

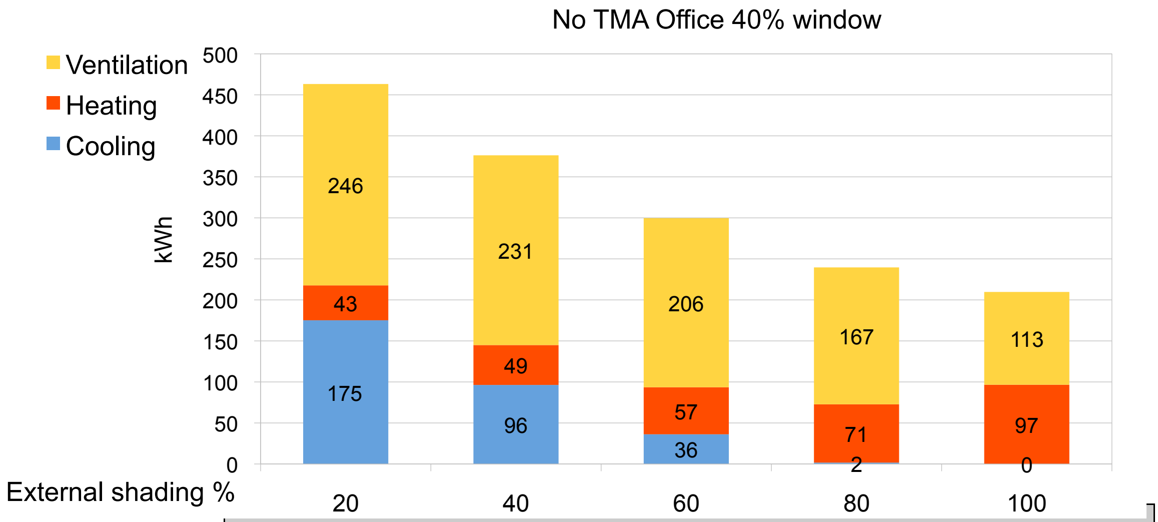
Office with 40% window and no TMA

The goal of this parameter study is to determine the optimal percentage of external shading.

The following table shows the parameters from an office room with 40% window, with this surface is oriented to the southwest.

PARAMETERS	
Room depth (m)	3.5
Room width (m)	2.5
Room height (m)	2.7
Floor area (m^2)	8.75
Wall area facing S-W (m^2)	6.75
Window area %	40
Room volume (m^3)	23.63
Orientation	232
Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains (W/m^2)	14.43
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	20C-26C
Inlet-outlet temp., heat ventilation	45C – 35C
Temperature of airflow	20
Air change of ventilation (1/h)	2
Air change rate per person (m^3/h)	26
Humidity %	50

Table 7. Parameters from an office with 40% window



	140 Watt Upper Dead band and 120 Watt Lower dead band				
External shading %	20	40	60	80	100
Cooling (kWh)	175.20	96.35	36.19	1.67	0.00
Heating (kWh)	42.52	48.63	57.34	71.05	96.61
Ventilation (kWh)	245.50	231.30	206.20	166.90	113.10
Total (kWh)	463.22	376.28	299.73	239.62	209.71

Fig.15 Different external shading devices for an office room with 40% window

The external shading increases as the ventilation and cooling decreases. This is due to the reduction of the passive solar gains entering the room. An external shading of 80% is chosen in order to allow a higher daylight factor and thus resulting in a better indoor climate.

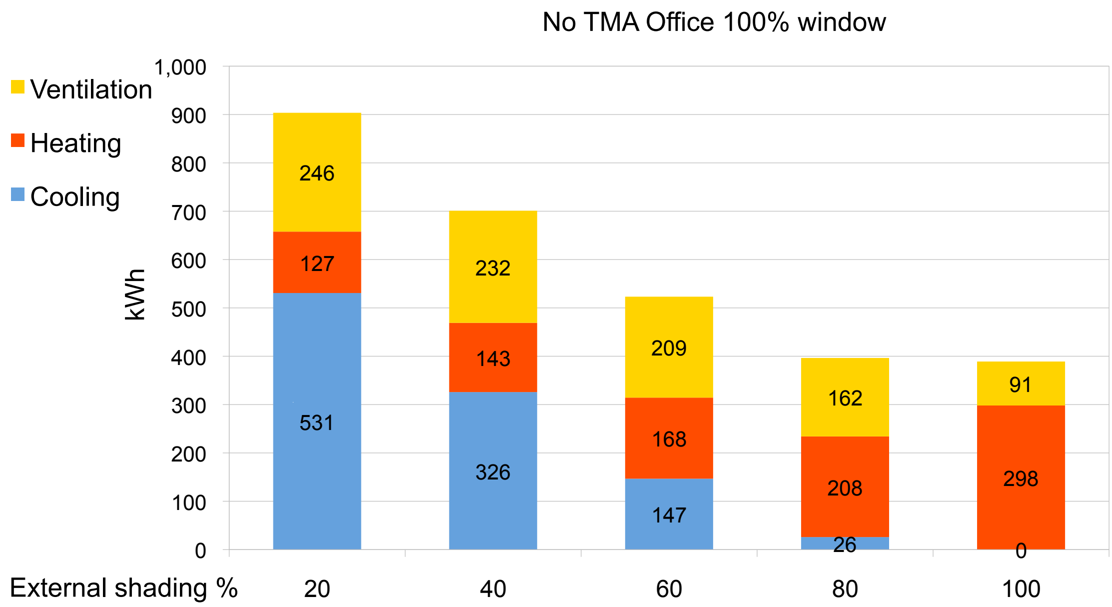
Office with 100% window

The following table show the parameters from an office room with 100% window with this surface oriented to the southwest.

PARAMETERS	
Room depth (m)	3.5
Room width (m)	2.5
Room height (m)	2.7
Floor area (m^2)	8.75
Wall area facing S-W (m^2)	6.75
Window area %	100
Room volume (m^3)	23.63
Orientation	232
Occupation (h)	3120

Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains (W/m^2)	14.43
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	21C-26C
Inlet-outlet temp. heat ventilation	45C – 35C
Temperature of airflow	20
Air change of ventilation (1/h)	2
Air change rate per person (m^3/h)	26
Humidity %	50

Table 8. Parameters from an office with 100% window



	140 Watt Upper Dead band and 120 Watt Lower dead band				
External shading %	20	40	60	80	100
Cooling (kWh)	530.80	325.90	146.70	25.92	0.00
Heating (kWh)	127.00	143.20	167.70	208.20	298.10
Ventilation (kWh)	245.80	231.90	208.80	162.30	90.96
Total (kWh)	903.60	701.00	523.20	396.42	389.06

Fig.15 External shading devices for an office room with 100% window

Here, the external shading has a high influence on the cooling load. With 80% external shading, cooling requirements become almost non-existent. A large difference is noted when these results are compared to the results from the office with 40% window. This can be explained by the lower U-value of the transparent area (0,7W/m^{2}K), as opposed to 0,15 W/m^{2}K of the opaque surface in the office room with 40% window. This makes the transmission heat loss through a window twice as high in addition to the area of the window also being more than double that of the 40% window office.

Conclusion

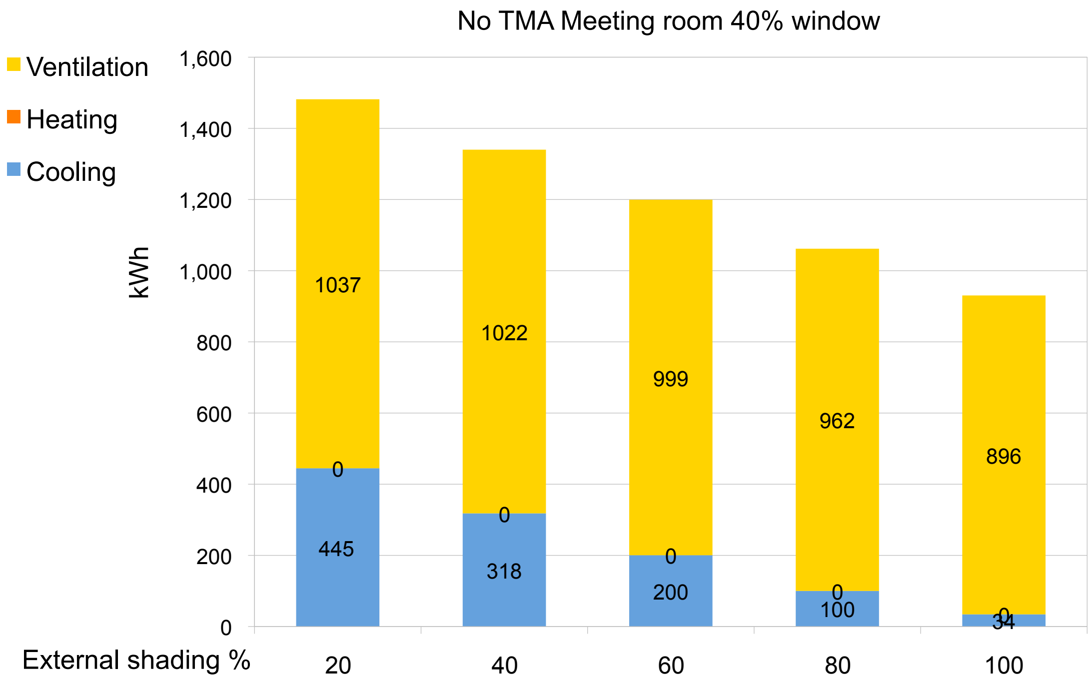
The best performance of the external shading for the office with 100% window is seen when the percentage of the external shading varies for each of the different seasons of the year. By allowing more passive solar gains during the wintertime, the heating demand will be reduced. Section 3.3.4 which discusses the yearly schedule, gives a good overview of the impact of this strategy.

Meeting room with 40% window

The following table show the parameters from a meeting room with 40% window with this surface oriented to the southwest.

PARAMETERS	
Room depth (m)	5.3
Room width (m)	3.3
Room height (m)	2.7
Floor area (m^2)	17.52
Wall area facing S-W (m^2)	8.91
Window area %	40
Room volume (m^3)	47.3
Orientation	232
Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains - 8persons (W/m^2)	28.76
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	21C-26C
Inlet – outlet temp. heat ventilation	45C – 35C
Temperature of airflow	19
Air change of ventilation (1/h)	3.5
Air change rate per person (m^3/h)	26
Humidity %	50

Table 9. Parameters from an meeting room with 40% window



					TMA in ceiling
	140 Watt Upper Dead band and 120 Watt Lower dead band				
External shading %	**40**	**60**	**80**	**100**	**80**
Cooling (kWh)	318.10	200.40	100.10	34.32	0.00
Heating (kWh)	0.00	0.00	0.00	0.00	10.78
Ventilation (kWh)	1022.00	999.10	961.70	896.10	670.00
TMA Cooling					472.36
TMA Heating					0.00
Total (kWh)	1340.10	1199.50	1061.80	930.42	1153.14

Fig.16 External shading devices for a meeting room with 40% window

Conclusion

A good cooling and ventilation strategy will be necessary for this room with high internal gains. The ventilation heat was lowered from 20°C to 19°C and the air change rate increased from 2 – 3.5 h^{-1}. By installing TMA, the cooling load became non-existent and the ventilation requirements were reduced. The indoor climate improved as well, as the indoor climate never surpassed 25°C (see fig.18 output data TRNSYS). Further investigation on the coefficient of performance (COP) suggests that the total energy demand will also decrease. A COP of 4 for TMA cooling was used in this diagram.

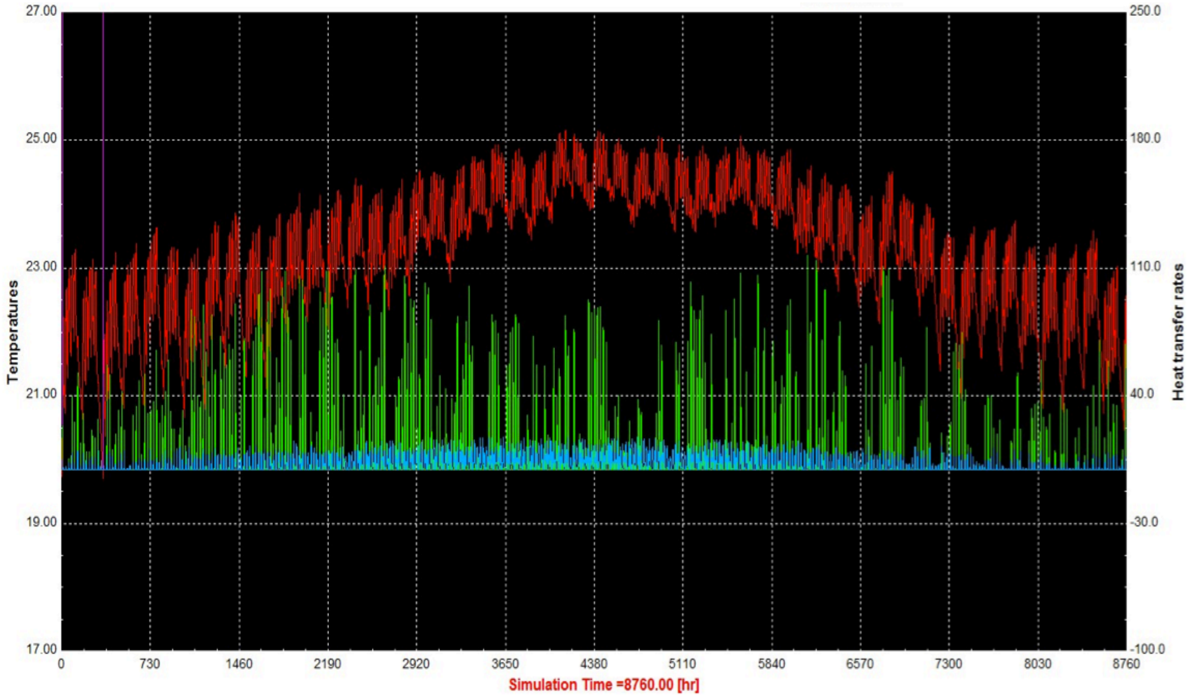


Fig. 17 Output data TRNSYS for a meeting room with TMA and just 1h^{-1} ventilation.

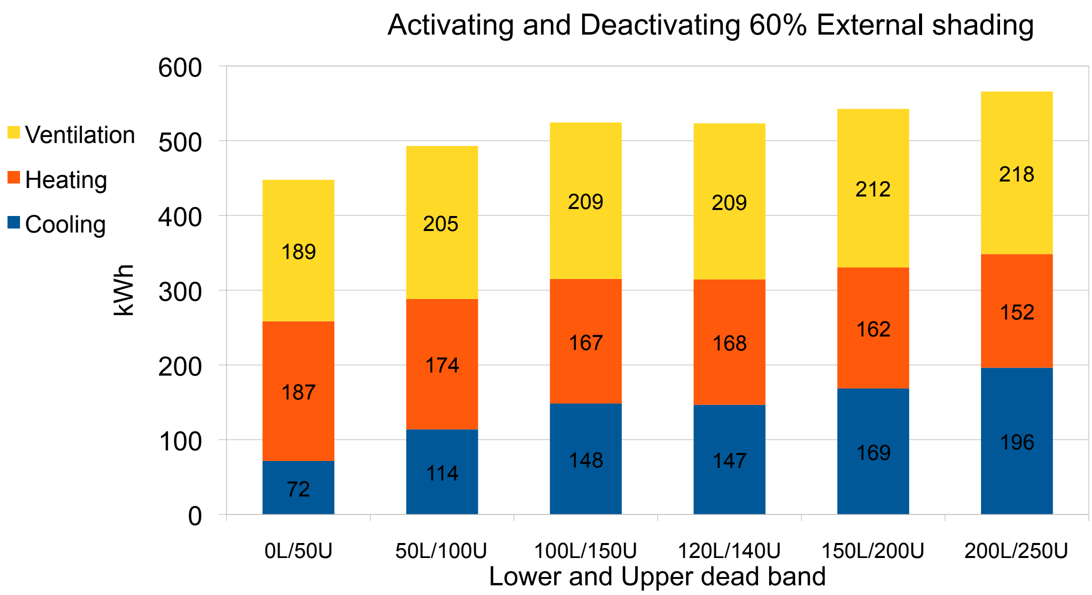
1. Red line stands for indoor climate °C
2. blue and green stands for direct and diffuse solar gains Watt

3.3.2. Activating and deactivating the external shading

This portion of the parameter study will be performed only on the office with 100% window because this room will have the most influence on the control of the shading. Here much effort should be exerted in order to find a good balance between not having an excessively high level of solar gains entering the room while still allowing an adequate amount of daylight to enter the room. The control of artificial lighting is dependent on this issue. Thus external shading will have a strong influence on the energy consumption of lighting.

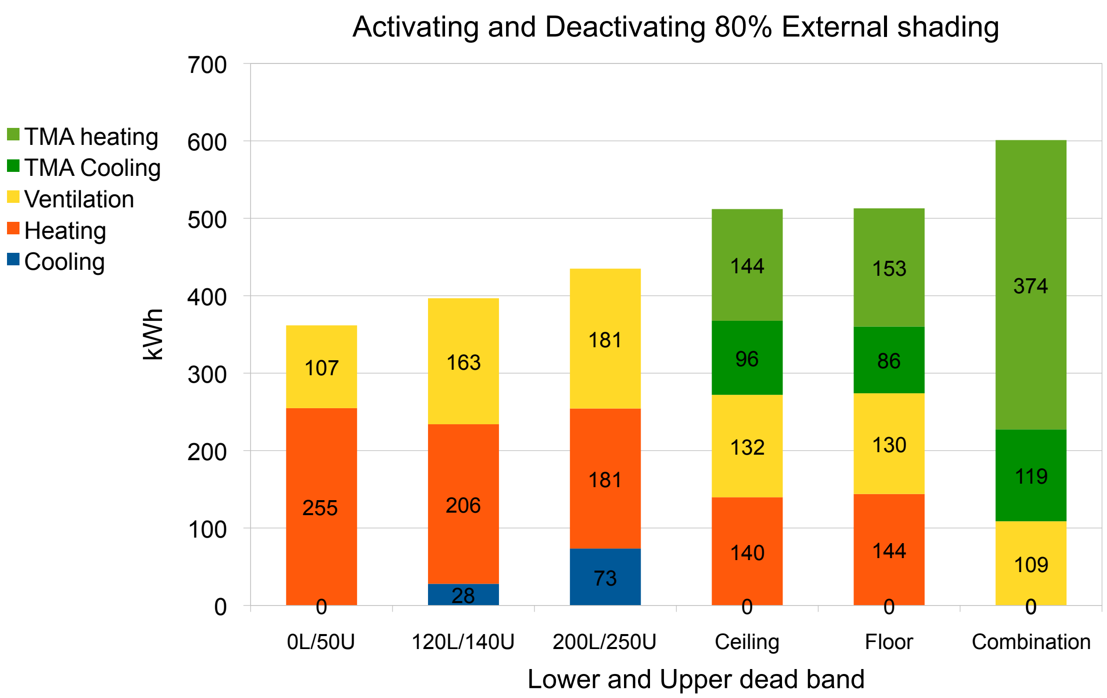
The windows will have an U-value of 0,7 W/m^{2}K and g-value of 0,6 W/m^{2}K. The lower and upper dead bands will control the external shading device, allowing for better control over the amount of horizontal solar gains entering the room. This will have a high influence on the solar gains and, ultimately, the cooling loads. The best performance for the lower and upper dead bands will be analyzed in Figure 18 and 19. These results were simulated and analyzed using TRNSYS.

First, a study is performed on controlling a 60% external shading device and in figure 20, the study is done on controlling an 80% external shading device.



	Office 100% Window and No TMA, 60% External Shading					
	0L/50U	50L/100U	100L/150U	120L/140U	150L/200U	200L/250U
Cooling	71.55	113.90	148.40	146.70	168.60	196.30
Heating (kWh)	186.90	174.30	166.70	167.70	161.80	152.00
Ventilation (kWh)	189.30	204.80	209.30	208.80	212.20	217.50
Total	447.75	493.00	524.40	523.20	542.60	565.80

Fig.18 Activating and deactivating from 60% external shading



	Office 100% Window and TMA 80% ExternalShading					
				TMA 120L – 140U		
	0L/50U	120L/140U	200L/250U	Ceiling	Floor	Combination
Cooling	0.00	27.93	73.47	0.00	0.00	0.00
Heating (kWh)	254.80	206.10	180.90	139.70	143.90	0.00
Ventilation (kWh)	106.90	162.70	180.60	132.30	130.10	108.70
TMA Cooling (kWh)				95.69	86.22	118.70
TMA heating (kWh)				144.24	152.63	373.62
Total	361.70	396.73	434.97	511.92	512.85	601.02

Fig.19 Activating and deactivating from 80% external shading and TMA

Conclusion

No TMA

Solar shading on the southwest façade is activated when the total radiation is greater than 140W/m^2, and it is deactivated when the total radiation is less than 120W/m^2. The solar external shading factor is 80%. Artificial lighting switches on when the total radiation is less than 120W/m^2 and it switches off when the total radiation is greater than 200W/m^2.

TMA integrated

By integrating the TMA technology, the cooling load can be reduced while simultaneously increasing the daylight factor. The external shading will be activated when the horizontal solar gains reach 140W/m^2 and can be deactivated when they are less than 120 W/m^2. Both shading assumptions will be used and discussed later in this thesis. A coefficient of performance for TMA heating, 2.2 (NS 3031) and cooling 4 (NS EN 14511) were also taken into account.

3.3.3. The impact of shading on indoor temperatures

Figure 20 shows the impact of external shading on a warm day with a high level of direct (green line) and diffuse solar gains (blue line). The red line represents the indoor climate temperature. The figure to the left shows the impact of an external shading device of 20% that is activated and deactivated as explained above in a room with no TMA. The figure to the right shows how an external shading device of 80% is activated and deactivated.

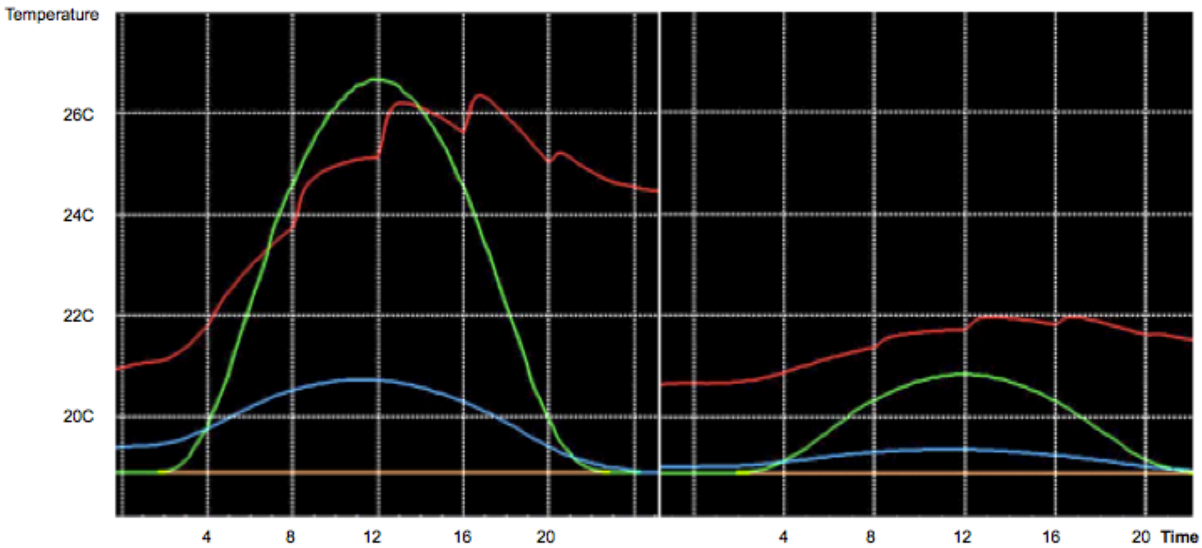


Fig. 20 The indoor climate is reduced by 4 degrees (26°C – 22°C) due to the high impact on the shading on the direct solar gains (TRNSYS).

- Red line represents for indoor climate °C
- Blue and green represent for direct and diffuse solar gains, respectively Watt

Conclusion

Shading has a considerable influence on the indoor climate. A good shading strategy is thus an important issue in the development of a project. The following section explains what could be a good approach for reaching this goal.

3.3.4. Yearly schedule for an external shading device

The parametric study using TRNSYS on the office with 100% window was chosen as this model profits most from the passive solar gains entering the room. The following integrated parameters were used.

PARAMETERS	
Room depth (m)	3.5
Room width (m)	2.5
Room height (m)	2.7
Floor area (m^2)	8.75
Wall area facing S-W (m^2)	6.75
Window area %	100
Room volume (m^3)	23.63
Orientation	232

Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains (W/m^2)	14.43
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	21C-26C
Inlet – outlet temp. heat ventilation	45C – 35C
Temperature of airflow	20
Air change of ventilation (1/h)	0.5
Air change rate per person (m^3/h)	26
Humidity %	50

Table 10. Parameters from the office with 100% window

A small amount of ventilation was used in this study in order to increase the influence on the yearly schedule. COPs for TMA were not considered.

The following integrated systems were chosen for this study:

- TMA in the floor and the ceiling with a different inlet and outlet water temperature for the winter and the summer*.
- TMA in the ceiling with a different inlet and outlet water temperature for the winter and the summer.
- No TMA

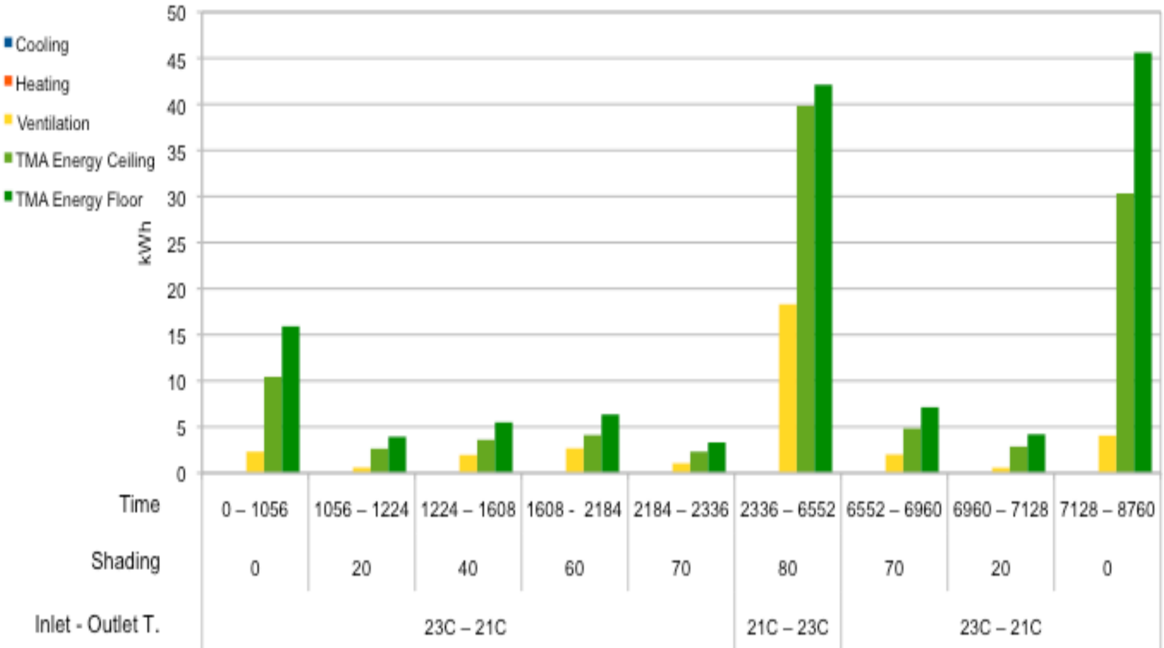
*Different water supply temperatures were chosen for activating the thermal mass element in order to maximize the profit from this technology. A climate analysis of Trondheim was conducted with TRNSYS and the following water temperatures were chosen to activate the elements: inlet water temperatures 23°C and outlet temperature of 21°C were chosen during the winter period. During the summer period, the system works in reverse. The system is adjusted when the temperature difference between the inlet and outlet temperatures are higher than 2°C. This is achieved by adjusting the mass flow rate.

This strategy was used in the literature study of a project located in Stockholm called "Kungsbrohuset" in order to create an optimal indoor climate.

The models were evaluated over a time period of a year (0 - 8760h)

Office 100% window and TMA in floor and ceiling

Figure 21 shows a yearly schedule for an office with 100% window and with TMA installed in the floor and in the ceiling.



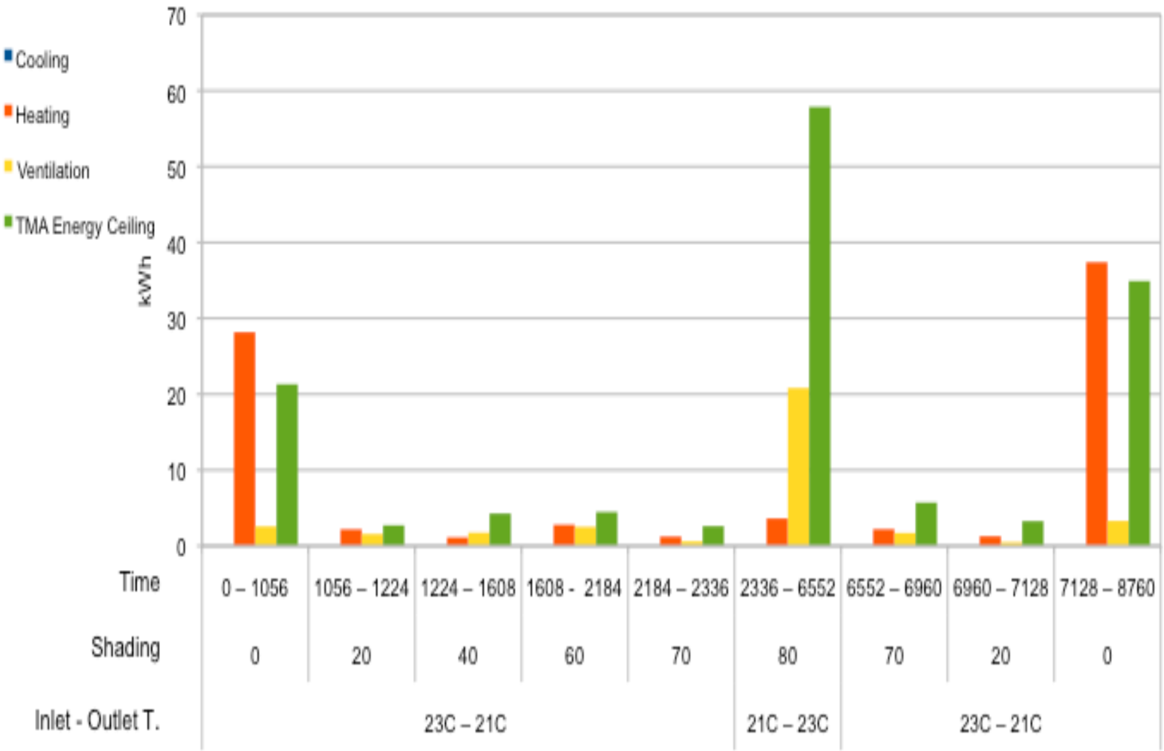
Tur – Retur Temp.	23C – 21C					21C – 23C	23C – 21C		
Shading %	0	20	40	60	70	80	70	20	0
Time (h)	0 – 1056	1056 – 1224	1224 – 1608	1608 - 2184	2184 – 2336	2336 – 6552	6552 – 6960	6960 – 7128	7128 – 8760
Cooling	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ventilation	2.32	0.60	1.97	2.69	1.06	18.31	2.02	0.58	4.06
TMA Energy Ceiling	10.43	2.64	3.61	4.12	2.33	39.83	4.84	2.87	30.33
TMA Energy Floor	15.92	3.94	5.50	6.35	3.31	42.12	7.13	4.20	45.62
Total (kWh)	**28.67**	**7.18**	**11.07**	**13.16**	**6.69**	**100.26**	**13.99**	**7.65**	**80.01**
Total 0 – 8760h (kWh)	**268.70**								

Fig.21 Yearly schedule for the office with 100% window and TMA in the ceiling and the floor

The periods for switching over to another shading device were chosen by first analysing and simulating the indoor temperatures. These periods were determined such that the indoor temperatures did not reach the boundary cooling temperature of 26°C; this study ensured that the indoor temperatures stayed between 21°C and 25°C in order to provide a good indoor climate. This allowed for not only a good indoor temperature but also maximum natural lighting and hence a reduction in the amount of artificial lighting required during the occupied hours.

Office with 100% window and TMA in the ceiling

Figure 22 will show a yearly schedule for an office with 100% window and with TMA installed in the ceiling.



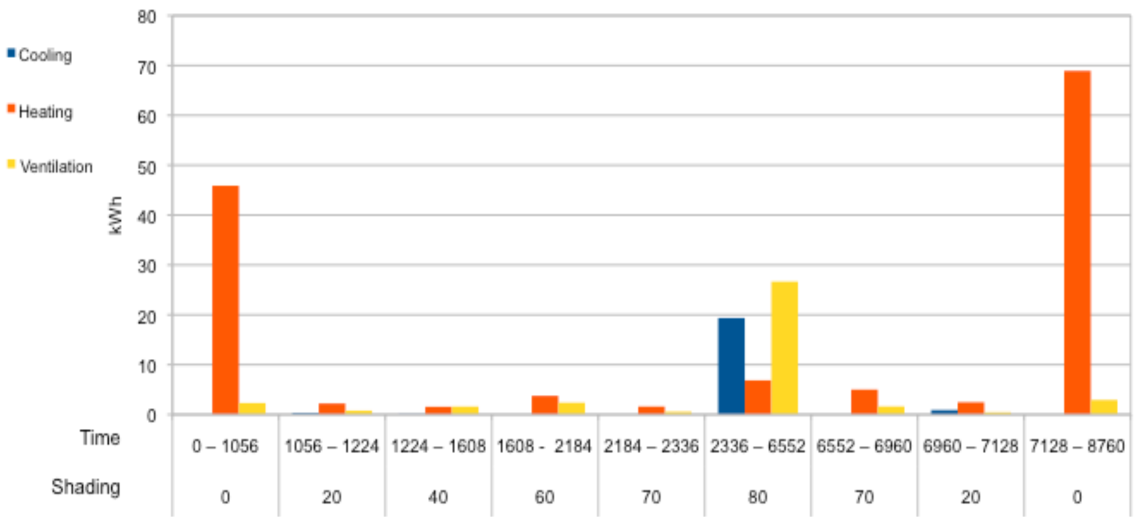
Tur – Retur Temp.	23C – 21C					21C – 23C	23C – 21C		
Shading %	**0**	**20**	**40**	**60**	**70**	**80**	**70**	**20**	**0**
Time (h)	**0 – 1056**	**1056 – 1224**	**1224 – 1608**	**1608 - 2184**	**2184 – 2336**	**2336 – 6552**	**6552 – 6960**	**6960 – 7128**	**7128 – 8760**
Cooling	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating	28.17	2.20	1.13	2.80	1.24	3.63	2.22	1.28	37.40
Ventilation	2.58	1.58	1.75	2.52	0.60	20.83	1.71	0.49	3.30
TMA Energy Ceiling	21.32	2.72	4.27	4.48	2.61	57.88	5.72	3.27	34.92
Total (kWh)	**52.07**	**6.49**	**7.15**	**9.80**	**4.45**	**82.33**	**9.65**	**5.04**	**75.62**
Total 0 – 8760h (kWh)	**252.60**								

Fig. 22 Yearly schedule for the office with 100% window and TMA in the ceiling

These results were compared with the results of the office with 100% window and TMA in both the floor and the ceiling. The total energy for the office with TMA in the ceiling is lower than the office with TMA in the ceiling and the floor. This suggests that the cooling load is higher than the heating demand due to the higher efficiency of cooling from the ceiling. This was proven in the parametric study on the heat capacity in the ceiling and the floor (see section 3.1.).

Office with 100% window, no TMA and heating of 21°C

Figure 23 will show a yearly schedule for an office with 100% window and with no TMA installed and a heating system with a temperature of 21°C.



Shading %	0	20	40	60	70	80	70	20	0
Time (h)	0 – 1056	1056 – 1224	1224 – 1608	1608 - 2184	2184 – 2336	2336 – 6552	6552 – 6960	6960 – 7128	7128 – 8760
Cooling	0.00	0.26	0.06	0.00	0.00	19.35	0.00	0.90	0.00
Heating	45.89	2.23	1.55	3.76	1.61	6.86	5.01	2.48	68.92
Ventilation	2.31	0.78	1.60	2.39	0.57	26.67	1.62	0.45	2.93
Total (kWh)	**48.20**	**3.27**	**3.21**	**6.14**	**2.18**	**52.88**	**6.63**	**3.83**	**71.85**
Total 0 – 8760h (kWh)	**198.18**								

Fig.23 Yearly schedule for the office with 100% window and no TMA

These results were analysed and compared with the ones from the office with TMA in the ceiling. It can be stated that during the summer, there is a need for 20kWh external cooling in the office when there are no thermal active elements. This implies that the ambient indoor temperatures approach the boundary temperature of 26°C whereas the temperatures in the room with TMA stay easily below 25°C. The indoor climate is therefore better in the office with the thermal activated elements. As mentioned in the beginning of this section, no COPs were considered. If a COP for cooling of 2.4 was taken into account according to standard NS3031, then the total cooling energy required drops to 172kWh, which is 26kWh less than the total cooling energy required for the office without active elements. This proves that TMA provides a better indoor climate with less energy.

Conclusion

A yearly schedule has a high reduction potential for the total amount of heat needed due to the optimized use of passive solar gains. The indoor temperatures were also always kept under the boundary temperatures of 25 degrees. This, together with optimized day lighting, increases significantly the indoor climate.

3.4. Water mass flow rates and ventilation

In this parameter study, the optimal water flow rate in TMA elements is found for a meeting room with TMA installed.

The parameters of this room have been previously described in table 3. The only difference is the number of air changes, which has been reduced from $3.5h^{-1}$ to $1h^{-1}$. This change allows for a better understanding on the influence of the mass flow rate for cooling and heating.

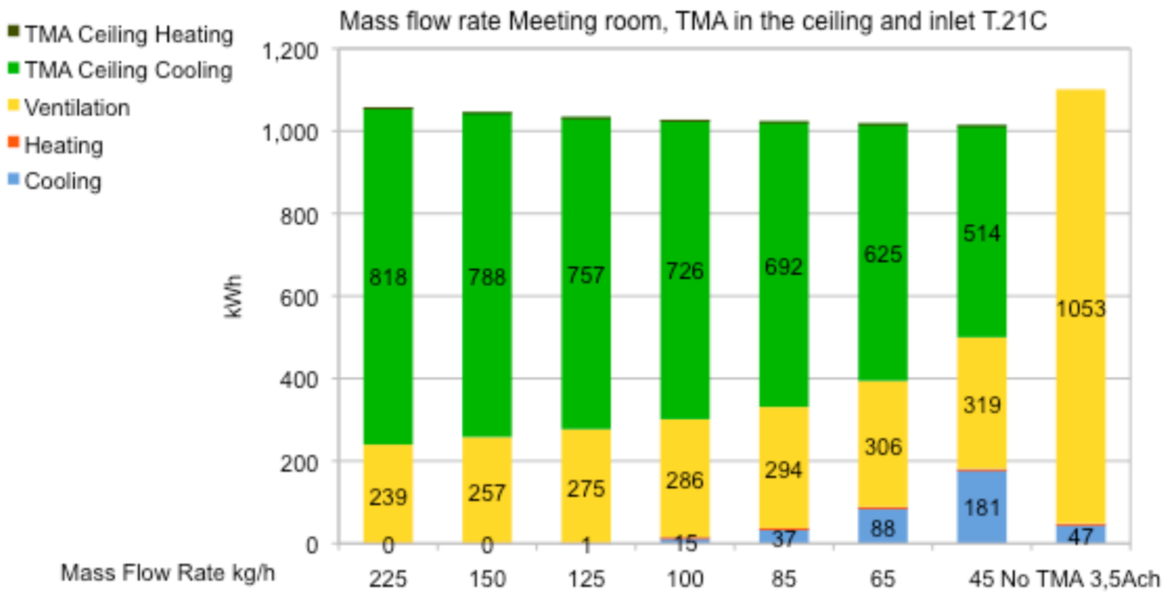
TMA is located in the ceiling and in the second study it is located in the floor as ceiling.

PARAMETERS	
Room depth (m)	5.3
Room width (m)	3.3
Room height (m)	2.7
Floor area (m^2)	17.52
Wall area facing S-W (m^2)	8.91
Window area %	40
Room volume (m^3)	47.3
Orientation	232
Occupation (h)	3120
Clothing factor (clo)	1
Metabolic rate (met)	1.2
Internal gains - 8persons (W/m^2)	28.76
Upper dead band shad.(W/m^2)	140
Lower dead band shad.(W/m^2)	120
Indoor climate	21C-26C
Inlet – outlet temp. heat ventilation	45C – 35C
Temperature of airflow	19
Air change of ventilation (1/h)	1
Air change rate per person (m^3/h)	26
Humidity %	50

Table 11 Parameters meeting room

Meeting room with TMA in the ceiling

Figure 24 shows a study done on the mass flow rate with an inlet water temperature of 21°C for the embedded pipes. Pipes with an inner diameter of 2cm and 2mm wall thickness, spaced 20cm apart are placed 60mm from the surface of the concrete. This layout is in accordance with NS-EN 15377. In this study, the pipe is located 6cm from the ceiling surface. The total floor thickness is 24cm of reinforced concrete, which has high thermal inertia.



	Mass flow rate (kg/h) and 21C							
Mass Flow Rate (kg/h)	**225**	**150**	**125**	**100**	**85**	**65**	**45**	No TMA 3,5Ach
Cooling	0.00	0.00	1.39	14.96	37.02	88.14	180.50	46.53
Heating	0.48	0.39	0.31	0.24	0.21	0.20	0.20	1.68
Ventilation	239.30	257.40	275.10	286.30	293.80	305.60	319.10	1053.00
TMA Ceiling Cooling	817.71	787.72	757.09	725.71	692.44	624.54	514.48	
TMA Ceiling Heating	0.22	0.17	0.15	0.14	0.12	0.11	0.08	
TOTAL (kWh)	**1057.72**	**1045.69**	**1034.04**	**1027.35**	**1023.60**	**1018.58**	**1014.36**	**1101.21**

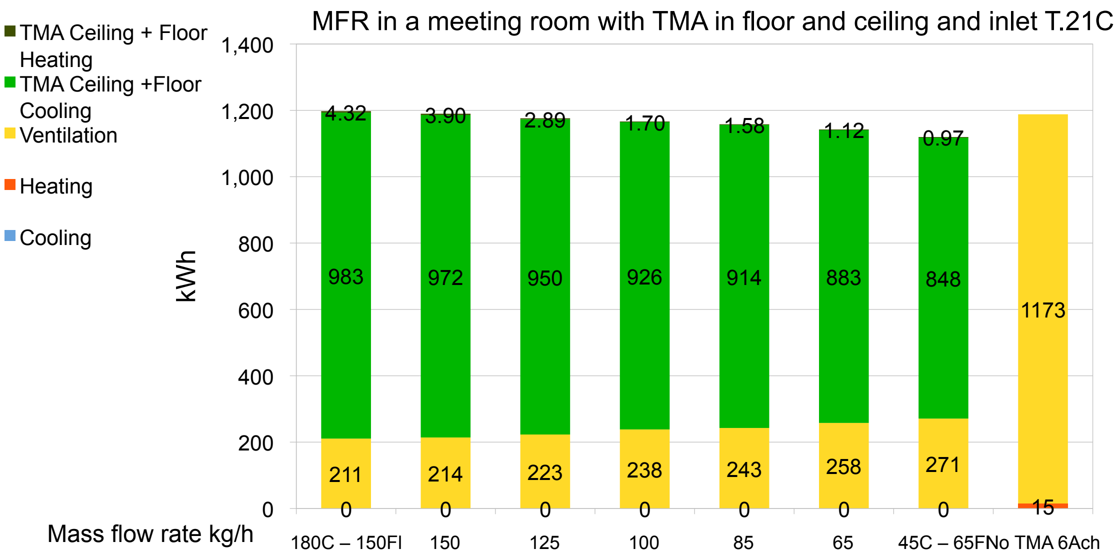
Fig. 24 Several mass flow rates in a meeting room with TMA in the ceiling

This table shows that for a water mass flow of 125kg/h and inlet temperature of 21°C, the cooling load is almost non-existent. For a mass flow of 150kg/h there is no cooling load at all, and from 225kg/h, the maximum cooling temperature can even be lowered to 25°C (see fig.18). To obtain a similar indoor climate in a room without TMA, the ventilation must be increased to $3.5h^{-1}$ and even then, there is still a cooling load of

47kWh. A cooling COP of 4 and a heating COP of 2.22 were used for TMA. These values are in accordance with EN14511 and NS3031. The values for TMA cooling are even higher but COP will be discussed further in section 3.7.

Meeting room with TMA in the ceiling and floor

Figure 25 shows a study done on the mass flow rate with an inlet water temperature of 21°C in the embedded pipes. Pipes with an inner diameter of 2cm and 2mm wall thickness, spaced 20cm apart are located 60mm from the surface of the concrete. This layout is according to NS-EN 15377. In this study, there are two circuits of pipes; one is located 6cm from the ceiling surface and the other is located 6cm from the floor surface. The total floor thickness is 24cm of reinforced concrete, which has high inertia.



	Mass flow rate (kg/h) and 21							
Mass Flow Rate (kg/h)	**180C – 150Fl**	**150**	**125**	**100**	**85**	**65**	**45C – 65Fl**	No TMA 6Ach
Cooling	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating	0.00	0.00	0.00	0.00	0.00	0.00	0.00	15.07
Ventilation	210.70	214.10	223.00	238.30	242.70	257.90	271.00	1173.00
TMA Ceiling +Floor Cooling	982.57	971.80	950.21	926.47	913.89	883.36	847.59	
TMA Ceiling + Floor Heating	4.32	3.90	2.89	1.70	1.58	1.12	0.97	
Total (kWh)	**1197.60**	**1189.80**	**1176.10**	**1166.48**	**1158.17**	**1142.38**	**1119.56**	**1188.07**

Fig.25 Several mass flow rates in a meeting room with TMA in the ceiling

When the water mass flow rate is 45kg/h in the ceiling and 65kg/h in the floor, there is no cooling load. From a water mass flow rate of 100kg/h in both the floor and ceiling TMA, the maximum cooling temperature can be lowered to 25°C. With a mass flow rate of 180kg/h for the ceiling and 150kg/h in the floor, the maximum cooling load temperature can even be lowered to 24°C. With this lowered boundary temperature for cooling, a good indoor climate is assured. In a meeting room without TMA, the ventilation must be increased to $6h^{-1}$ in order to maintain an indoor climate under the 26°C boundary temperature. A cooling COP of 4 and a heating COP of 2.22 were used for TMA. These values are in accordance with EN14511 and NS3031. These values are in accordance with EN14511 and NS3031. The values for active cooling and heating are even higher, but the COP will be discussed further in section 3.7. As previously mentioned, heat recovery cannot be incorporated in TRNSYS. The ventilation rate should be high in this case in order to obtain the same results as an indoor climate with integrated thermal active elements.

Conclusion

The water mass flow rate has a strong influence on cooling load, with sufficient flow rates often eliminating the cooling load. The boundary temperatures for cooling can also be lowered significantly. The results also indicate that the ceiling is the best location for having this technology as demonstrated by the fact that the cooling is almost non-existent from a mass flow rate of 125kg/h and at a water temperature of 21 °C in this scenario. In the parametric study on the heat exchange coefficient, it was also found that the ceiling is the most efficient place for reducing the cooling load. TMA is also more efficient than ventilation in this case because ventilation does not achieve the same level of indoor climate.

These studies indicate that ventilation requirements are reduced significantly with integrated TMA. The reason this occurs is due to the reduction of the cooling load but future research will be necessary.

3.5. Radiant time factor

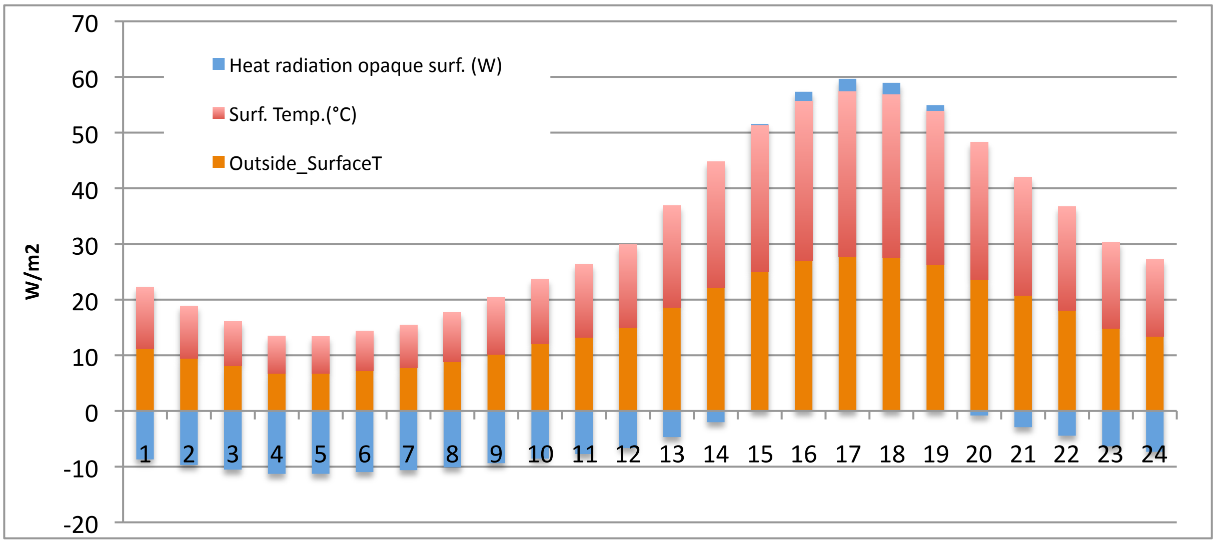
The radiant time factor plays the most important role taking into account the time-delayed heat radiated from internal furniture but mostly from walls, floors and ceilings. This is directly related to the level of inertia of the element and how exposed the element is. This time delayed radiation occurs after the occupation period.

Findings from the parametric study performed in 3.5 indicates that a large quantity of thermal mass has a benefit on the indoor climate in an office building. The radiant factor will be used in the following studies performed in 3.6. cooling load. These studies are performed according to guidelines given in ASHRAE 2009.

3.5.1. Heat gain through opaque surfaces

The office with 40% window was used for this parametric study because it has the largest opaque surface area. The equations used for this parametric study can be found in the section heat gain through the opaque surface 3.5.1.

This study shows that more is heat lost through the opaque surface than gained from outside to inside. Figure 26 shows the heat gain through an opaque surface.



Heat gain through opaque surface		
Total solar radiation (W)	146.81	
Absorbtion wood	0.41	
Heat transfer coefficient	5.50	
U-value wall (W/m^{2}K)	0.15	
U-value window (W/m^{2}K)	0.70	
Surface window	2.70	
Surface opaque wall	4.05	
Outside_SurfaceT	**Surf. Temp.(°C)**	**Heat radiation opaque surf. (W)**
11.12	11.12	-8.7
9.45	9.44	-9.7
8.05	8.04	-10.5
6.73	6.72	-11.3
6.71	6.70	-11.3
7.20	7.19	-11.0
7.74	7.75	-10.6
8.85	8.86	-10.1
10.20	10.22	-9.4
12.04	11.70	-8.5
13.21	13.16	-7.7
14.92	14.95	-6.7
18.59	18.26	-4.7
22.14	22.67	-2.0
25.04	26.32	0.2
26.99	28.70	1.6
27.78	29.66	2.2
27.52	29.37	2.0
26.18	27.73	1.0
23.65	24.67	-0.8
20.79	21.20	-2.9
18.04	18.69	-4.4
14.85	15.47	-6.4
13.38	13.81	-7.3
		-136.98

Figure 26. The heat transfer through the opaque surface of an office with 40% window on a warm day during the summer in Trondheim

Figure 26 shows that there is heat transfer from outside to inside for only 5hrs of the day. For the rest of the day, there is a net heat transfer from the indoor environment to the outdoor environment.

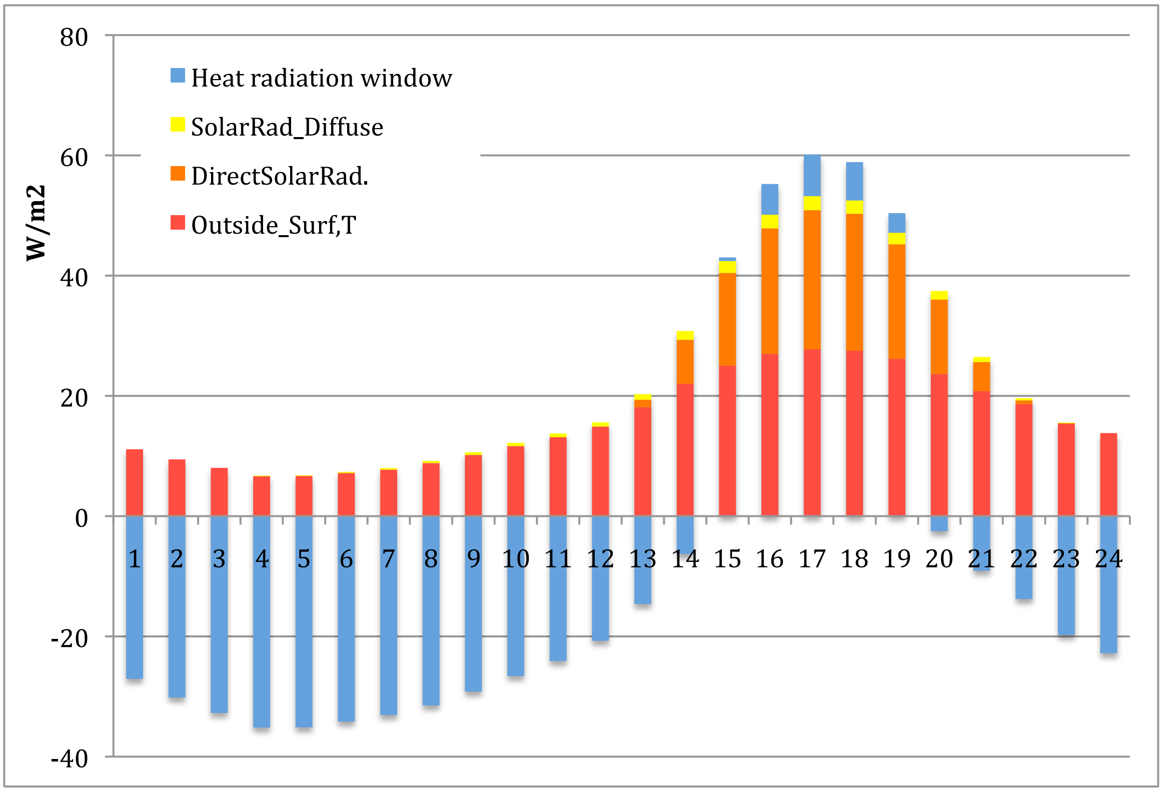
Conclusion

There is more indoor heat loss to the outdoors through the opaque surface than there is a heat gain from the outdoors to the indoor environment.

3.5.2. Fenestration heat gain and heat loss according to ASHRAE 2009

For spaces with neutral or positive air pressure, the primary weather-related variable affecting cooling load is solar radiation. These solar gains can be divided into direct, diffuse and conductive gains, the sum of which is the total fenestration heat gain.

The following figure 27 shows the solar heat gain and heat loss through a window of an office with 40% window area on one wall. The values used are typical of a warm day during the summer in Trondheim.



Total solar radiation (W)	146.81				
Absorbtion wood	0.41				
Heat transfer coefficient	5.50				
U-value wall (W/m²K)	0.15				
U-value window (W/m²K)	0.70				
Surface window	2.70				
Surface opaque wall	4.05				
TIME	**IndoorTemp.**	**Outside_Surf,T**	**DirectSolarRad.**	**SolarRad_Diffuse**	**Heat radiation window**
1	25.44	11.12	0.00	0.00	-27.1
2	25.40	9.44	0.00	0.00	-30.2
3	25.36	8.04	0.00	0.00	-32.7
4	25.32	6.72	0.00	0.01	-35.2
5	25.29	6.70	0.00	0.09	-35.1
6	25.27	7.18	0.00	0.18	-34.2
7	25.26	7.73	0.00	0.27	-33.1
8	25.52	8.83	0.00	0.37	-31.5
9	25.66	10.18	0.00	0.46	-29.2
10	25.77	11.66	0.00	0.55	-26.6
11	25.92	13.12	0.00	0.63	-24.1
12	25.92	14.90	0.00	0.69	-20.7
13	26.00	18.10	1.25	0.95	-14.6
14	26.00	22.01	7.30	1.47	-6.3
15	26.00	25.02	15.42	1.99	0.6
16	26.00	26.98	20.89	2.27	5.1
17	26.00	27.76	23.12	2.34	6.9
18	26.00	27.51	22.78	2.22	6.4
19	26.00	26.16	19.05	1.92	3.3
20	26.00	23.64	12.36	1.44	-2.5
21	26.00	20.77	4.86	0.84	-9.1
22	25.98	18.62	0.64	0.36	-13.8
23	25.93	15.47	0.00	0.09	-19.8
24	25.88	13.81	0.00	0.00	-22.8
			127.67	19.14	-426.17

Figure 27 Shows the heat gain and heat loss through the window of an office with 40% window and with an 80% external shading device

The equations used for these calculations are explained in section 3.5.2. The table shows that there is a heat gain through the window for only 5hrs of the day. For the rest of the day, there is a net heat transfer from inside to outside, corresponding to heat loss. The sum of these heat transfer values indicate that there is a net heat loss through the window than heat gain toward the indoor volume over the entire day. The heat loss here is almost 300W/m^2 higher than the heat loss through the opaque envelope see 3.5.1. The reason for this is the difference in U-value of both the opaque envelope and the window. The U-value of the opaque area (0.15W/m^{2}K) is a considerably better than the U-value of the window (0.7W/m^{2}K). Additionally, the heat loss compared to the heat gain is quite high due to the 80% shading. This shading is necessary for the building; otherwise, the cooling load would be excessively high.

Conclusion

Windows are beneficial in that they provide daylight, a factor which increases the indoor comfort. However, as shown in figure 28, it is recommended to have only a small amount of transparent surfaces since the heat loss is much higher than the solar heat gains with these surfaces. The window area should be related to a performance of the daylight factor. This is also the main reason why heating systems are located in the same area of the room as the windows.

3.6. Cooling load

In this parametric study, the cooling load as explained 2.2 is implemented in this parameter study on the following three models:

- Office with 40% window
- Office with 100% window
- Meeting room with 40% window

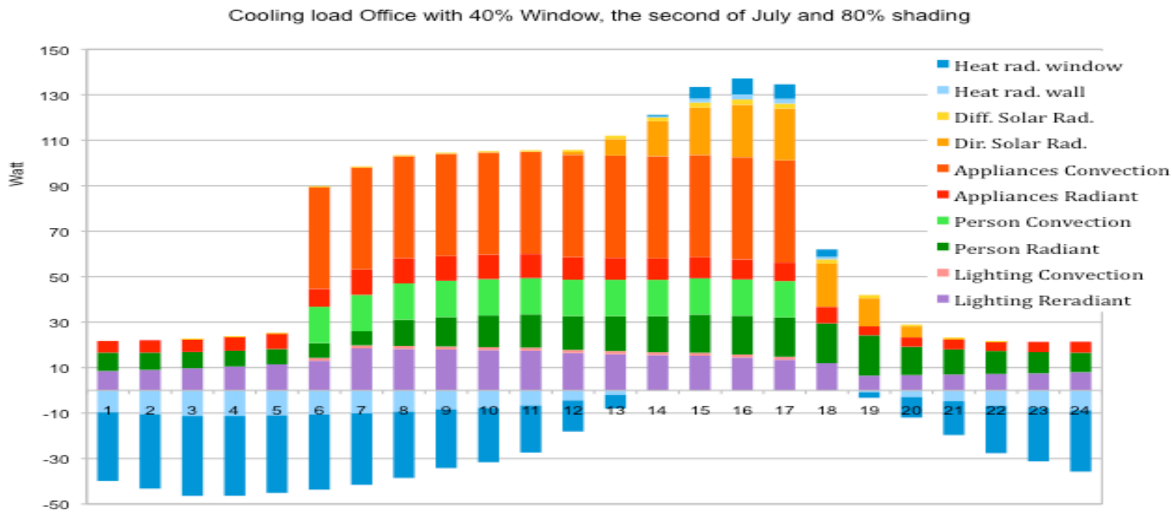
The parameters were explained in section 1.2. The implemented models did not make use of TMA so there is a clear indication of how much cooling load is required in these models. The results will be used for dimensioning the activated elements and will be discussed in the following section 2.3.

In the performance of COP 3.7. an air change of 0.3h^{-1} for office rooms is assumed because this is equivalent to a heat recovery rate of 85%.

Office with 40% window

Figure 28 shows the cooling load calculation for an office with 40% window on a warm summer day in Trondheim.

The cooling load is not high and can be eliminated altogether by having ventilation of 2ach (see table 1.2.1) or through the installation of TMA elements.

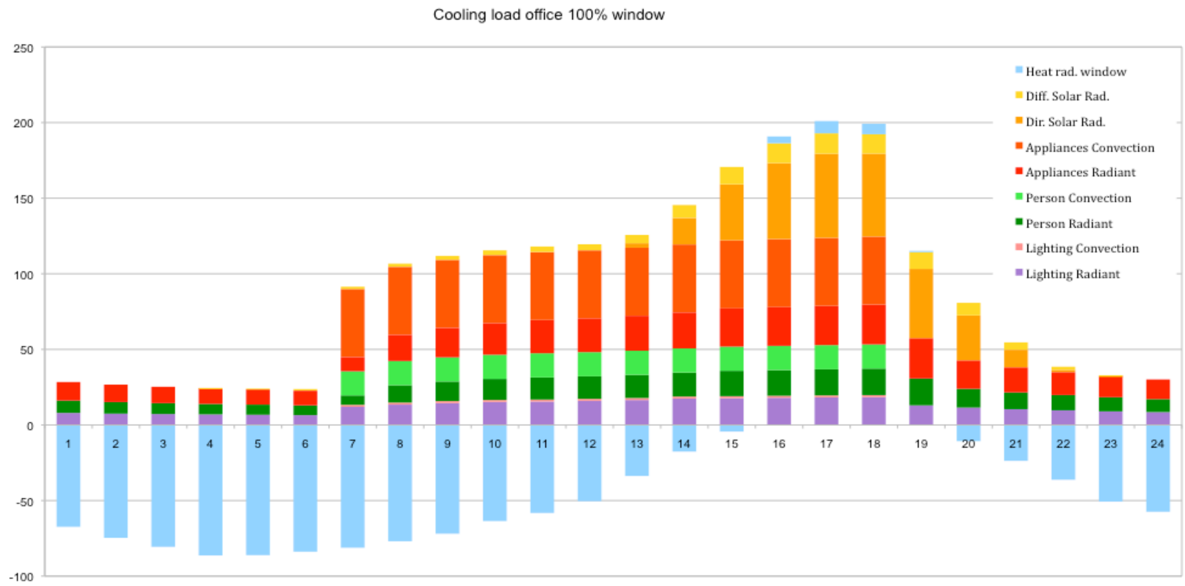


Power House 1: Office 40% Window 80% Shading, 0,5ach										
Cooling load Warmest day 2july & Orientation 232 Southwest in Watt										
	Lighting		Person		Appliances					
Hour	Reradiation	Convection	Reradiation	Convection	Reradiation	Convection	Dir. Solar Rad.	Diff. Solar Rad.	Heat rad. wall	Heat rad. window
24	8.49	0	8.16	0	5.1	0	0.00	0.00	-9.7	-30.2
23	8.99	0	7.68	0	5.4	0	0.00	0.00	-10.5	-32.7
22	9.74	0	7.2	0	5.85	0	0.00	0.01	-11.3	-35.1
21	10.49	0	6.96	0	6.3	0	0.00	0.09	-11.3	-35.1
20	11.48	0	6.72	0	6.9	0	0.00	0.18	-11.0	-34.1
19	12.98	1.314	6.48	16	7.8	45	0.00	0.27	-10.6	-33.1
18	18.47	1.314	6.24	16	11.1	45	0.00	0.37	-10.1	-31.5
17	18.23	1.314	11.52	16	10.95	45	0.00	0.46	-9.4	-29.2
16	17.98	1.314	12.96	16	10.8	45	0.00	0.55	-8.3	-25.9
15	17.73	1.314	13.92	16	10.65	45	0.00	0.63	-7.7	-23.9
14	17.48	1.314	14.64	16	10.5	45	0.00	0.69	-6.7	-20.7
13	16.48	1.314	14.88	16	9.9	45	1.25	0.95	-4.4	-13.7
12	15.98	1.314	15.36	16	9.6	45	7.30	1.47	-1.9	-6.1
11	15.48	1.314	15.84	16	9.3	45	15.42	1.99	0.2	0.6
10	15.23	1.314	16.8	16	9.15	45	20.89	2.27	1.7	5.1
9	14.48	1.314	17.04	16	8.7	45	23.12	2.34	2.2	6.9
8	13.48	1.314	17.28	16	8.1	45	22.78	2.22	2.1	6.4
7	11.98	0	17.52	0	7.2	0	19.05	1.92	1.1	3.3
6	6.49	0	17.76	0	3.9	0	12.36	1.44	-0.8	-2.5
5	6.74	0	12.48	0	4.05	0	4.86	0.84	-2.9	-9.1
4	6.99	0	11.04	0	4.2	0	0.64	0.36	-4.8	-14.9
3	7.24	0	10.08	0	4.35	0	0.00	0.09	-6.7	-20.9
2	7.49	0	9.36	0	4.5	0	0.00	0.00	-7.6	-23.6
1	7.99	0	8.64	0	4.8	0	0.00	0.00	-8.7	-27.1
SubTotal:	298.09	15.77	286.56	192.00	179.10	540.00	127.67	19.14	-137.16	-426.73
Total (Watt):										1094.44

Fig. 28 Heat gain and heat loss in an office with 40% window and 80% external shading.

Office room with 100% window

Figure 30 shows the cooling load calculation for an office with 40% window on a warm day during the summer in Trondheim.



Power House 1: Office 100% Window 80% Shading, 0,5ach

Cooling load Warmest day 2july & Orientation 232 Southwest in Watt

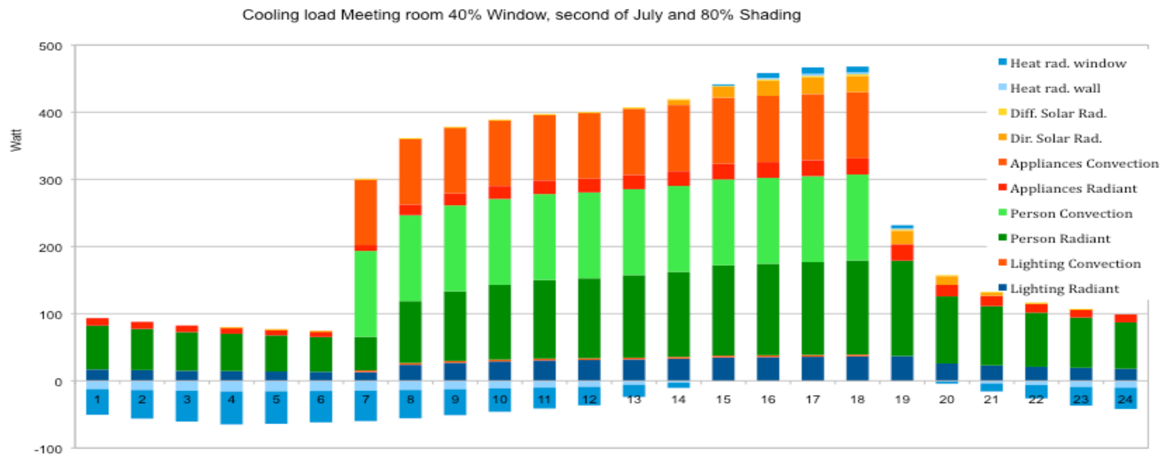
	Lighting		Person		Appliances				
Hour	**Reradiation**	**Convection**	**Reradiation**	**Convection**	**Reradiation**	**Convection**	**Dir. Solar Rad.**	**Diff. Solar Rad.**	**Heat rad. wall**
1	8.24	0	7.92	0	11.88	0	0.00	0.00	-74.7
2	7.74	0	7.44	0	11.16	0	0.00	0.00	-80.6
3	7.24	0	6.96	0	10.44	0	0.00	0.05	-86.3
4	6.99	0	6.72	0	10.08	0	0.00	0.54	-86.1
5	6.74	0	6.48	0	9.72	0	0.00	1.07	-83.8
6	6.49	1.314	6.24	16	9.36	45	0.00	1.55	-81.2
7	5.99	1.314	5.76	16	8.64	45	0.00	2.12	-76.9
8	12.23	1.314	11.76	16	17.64	45	0.00	2.64	-71.9
9	13.98	1.314	13.44	16	20.16	45	0.00	3.18	-63.5
10	14.98	1.314	14.4	16	21.6	45	0.00	3.64	-58.2
11	15.73	1.314	15.12	16	22.68	45	0.00	4.01	-50.4
12	15.73	1.314	15.12	16	22.68	45	3.01	5.50	-33.7
13	16.23	1.314	15.6	16	23.4	45	17.59	8.52	-17.6
14	16.73	1.314	16.08	16	24.12	45	37.13	11.48	-4.4
15	17.73	1.314	17.04	16	25.56	45	50.28	13.13	4.5
16	17.98	1.314	17.28	16	25.92	45	55.65	13.51	8.1
17	18.23	1.314	17.52	16	26.28	45	54.84	12.86	6.9
18	18.47	0	17.76	0	26.64	0	45.86	11.11	0.8
19	18.97	0	18.24	0	27.36	0	29.77	8.34	-10.7
20	12.73	0	12.24	0	18.36	0	11.70	4.83	-23.7
21	10.99	0	10.56	0	15.84	0	1.53	2.05	-36.2
22	9.99	0	9.6	0	14.4	0	0.00	0.50	-50.7
23	9.24	0	8.88	0	13.32	0	0.00	0.00	-57.4
24	8.74	0	8.4	0	12.6	0	0.00	0.00	-67.3
SubTotal:	289.36	15.77	278.16	192.00	417.24	540.00	307.35	110.63	-1027.6
Total (Watt):									1122.88

Fig. 29 Heat gain and heat loss in an office with 100% window and 80% external shading.

The cooling load here is not much higher than in the office with 40% window, but this is due to the high heat loss through the window. The heat loss is twice as high as in the office with 40% window. A good shading schedule for this scenario is thus recommended, see section 3.3.

Meeting room with 40% window

Figure 30 shows the cooling load calculation for an office with 40% window on a warm day during the summer in Trondheim.



Power House 1: Meeting 40% Window 80% Shading, 0,5 ach										
Cooling load Warmest day 2july & Orientation 232 Southwest in Watt										
	Lighting (Watt)		**Person (Watt)**		**Appliances (Watt)**					
Hour	**Radiant**	**Convection**	**Radiant**	**Convection**	**Radiant**	**Convection**	**Dir. Solar Rad.**	**Diff. Solar Rad.**	**Heat rad. wall**	**Heat rad. window**
1	16.98	0.00	65.28	0.00	11.17	0.00	0.00	0.00	-12.3	-38.3
2	15.98	0.00	61.44	0.00	10.51	0.00	0.00	0.00	-13.6	-42.6
3	14.98	0.00	57.6	0.00	9.86	0.00	0.00	0.00	-14.7	-45.9
4	14.48	0.00	55.68	0.00	9.53	0.00	0.00	0.01	-15.7	-49.1
5	13.98	0.00	53.76	0.00	9.20	0.00	0.00	0.10	-15.5	-48.4
6	13.48	0.00	51.84	0.00	8.87	0.00	0.00	0.20	-15.0	-46.8
7	12.98	2.63	49.92	128.00	8.54	98.55	0.00	0.29	-14.5	-45.4
8	23.97	2.63	92.16	128.00	15.77	98.55	0.00	0.40	-13.5	-42.3
9	26.96	2.63	103.68	128.00	17.74	98.55	0.00	0.50	-12.4	-38.7
10	28.96	2.63	111.36	128.00	19.05	98.55	0.00	0.60	-11.2	-34.9
11	30.46	2.63	117.12	128.00	20.04	98.55	0.00	0.69	-10.0	-31.2
12	30.96	2.63	119.04	128.00	20.37	98.55	0.00	0.75	-8.8	-27.6
13	31.96	2.63	122.88	128.00	21.02	98.55	1.32	1.03	-5.8	-18.1
14	32.96	2.63	126.72	128.00	21.68	98.55	7.75	1.63	-2.5	-7.9
15	34.95	2.63	134.4	128.00	23.00	98.55	16.36	2.23	0.3	1.1
16	35.45	2.63	136.32	128.00	23.32	98.55	22.15	2.57	2.3	7.1
17	35.95	2.63	138.24	128.00	23.65	98.55	24.52	2.65	3.0	9.5
18	36.45	2.63	140.16	128.00	23.98	98.55	24.16	2.53	2.8	8.8
19	36.95	0.00	142.08	0.00	24.31	0.00	20.21	2.18	1.5	4.6
20	25.96	0.00	99.84	0.00	17.08	0.00	13.12	1.62	-1.0	-3.1
21	22.97	0.00	88.32	0.00	15.11	0.00	5.15	0.93	-3.8	-11.9
22	20.97	0.00	80.64	0.00	13.80	0.00	0.67	0.39	-6.3	-19.8
23	19.47	0.00	74.88	0.00	12.81	0.00	0.00	0.09	-8.9	-27.9
24	17.98	0.00	69.12	0.00	11.83	0.00	0.00	0.00	-10.1	-31.6
SubTotal:	596.19	31.56	2292.48	1536.00	392.23	1182.60	135.42	21.39	-185.9	-580.6
Total (Watt):										5421.31

Figure 30 Heat gain and heat loss in an office with 100% window and 80% external shading.

The cooling load is the highest in this scenario. This is due to the higher internal gains during the occupation period between 6:00 and 18:00. Here, the influence of integrated active elements would be the greatest.

Conclusion

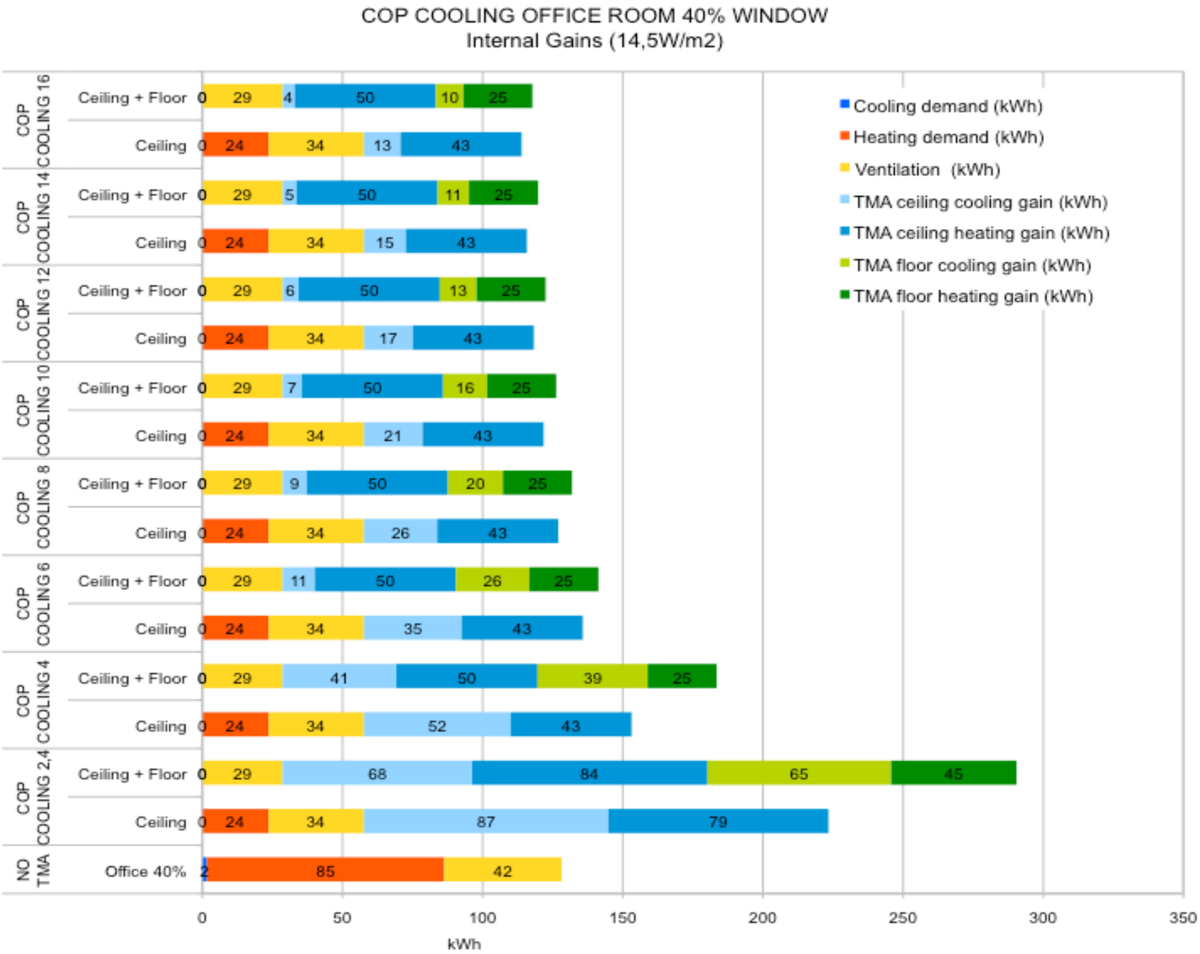
These calculations give a good overview on the cooling load in each room model. The room with 100% window has similar, although somewhat higher cooling load than the office with 40% window. The reason for this is the increase solar gain that reaching the internal space due to increased window area brings negates some of the heat loss.

3.7. Coefficient of performance in the three models

Heat recovery is important because it reduces the heat loss through ventilation. This study investigates the effect 85% heat recovery by changing the ventilation parameter in the three room scenarios. Decreasing the air change rates for ventilation by 85% has an equivalent effect to the implementation of heat recovery, which cannot be modelled using TRNSYS. The values for heating and cooling are obtained by calculations performed according to TEK10 regulations.

Office with 40% window

The COP values of 2.2 for heating and 2.4 for cooling are used in this model, in accordance with NS3031. These values are for water based floor systems.



	Office Room 40% Window and TMA, 80% ExternalShading 2h⁻¹ and H.R. ventilation 85% (0.3h⁻¹)								
	NO TMA	COP COOLING 2,4		COP COOLING 4		COP COOLING 6		COP COOLING 8	
COP HEATING TMA 3,5	Meeting room	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor
Cooling demand (kWh)	1.67	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating demand (kWh)	84.58	23.70	0.00	23.70	0.00	23.70	0.00	23.70	0.00
Ventilation (kWh)	41.91	33.95	28.66	33.95	28.66	33.95	28.66	33.95	28.66
TMA ceiling cooling gain (kWh)		87.22	67.64	52.33	40.59	34.89	11.46	26.17	8.60
TMA ceiling heating gain (kWh)		78.51	83.95	43.18	50.37	43.18	50.37	43.18	50.37
TMA floor cooling gain (kWh)			65.46		39.28		26.18		19.64
TMA floor heating gain (kWh)			44.75		24.61		24.61		24.61
Total (kWh)	128.17	223.38	290.47	153.16	183.51	135.72	141.29	127.00	131.88
	NO TMA	COP COOLING 10		COP COOLING 12		COP COOLING 14		COP COOLING 16	
	Meeting room	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor
Cooling demand (kWh)	1.67	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating demand (kWh)	84.58	23.70	0.00	23.70	0.00	23.70	0.00	23.70	0.00
Ventilation (kWh)	41.91	33.95	28.66	33.95	28.66	33.95	28.66	33.95	28.66
TMA ceiling cooling gain (kWh)		20.93	6.88	17.44	5.73	14.95	4.91	13.08	4.30
TMA ceiling heating gain (kWh)		43.18	50.37	43.18	50.37	43.18	50.37	43.18	50.37
TMA floor cooling gain (kWh)			15.71		13.09		11.22		9.82
TMA floor heating gain (kWh)			24.61		24.61		24.61		24.61
Total (kWh)	128.17	121.76	126.23	118.27	122.47	115.78	119.78	113.91	117.76

Fig. 32 different COPs for TMA cooling and heating in an office with 40% window

For heating in the office with no TMA, a COP value of 0.84 is used. This is due to the fact that heating for the room comes from the district-heating grid and this is the value recommended by NS3031. For no TMA cooling, the COP value is 2.4, also in accordance with NS3031.

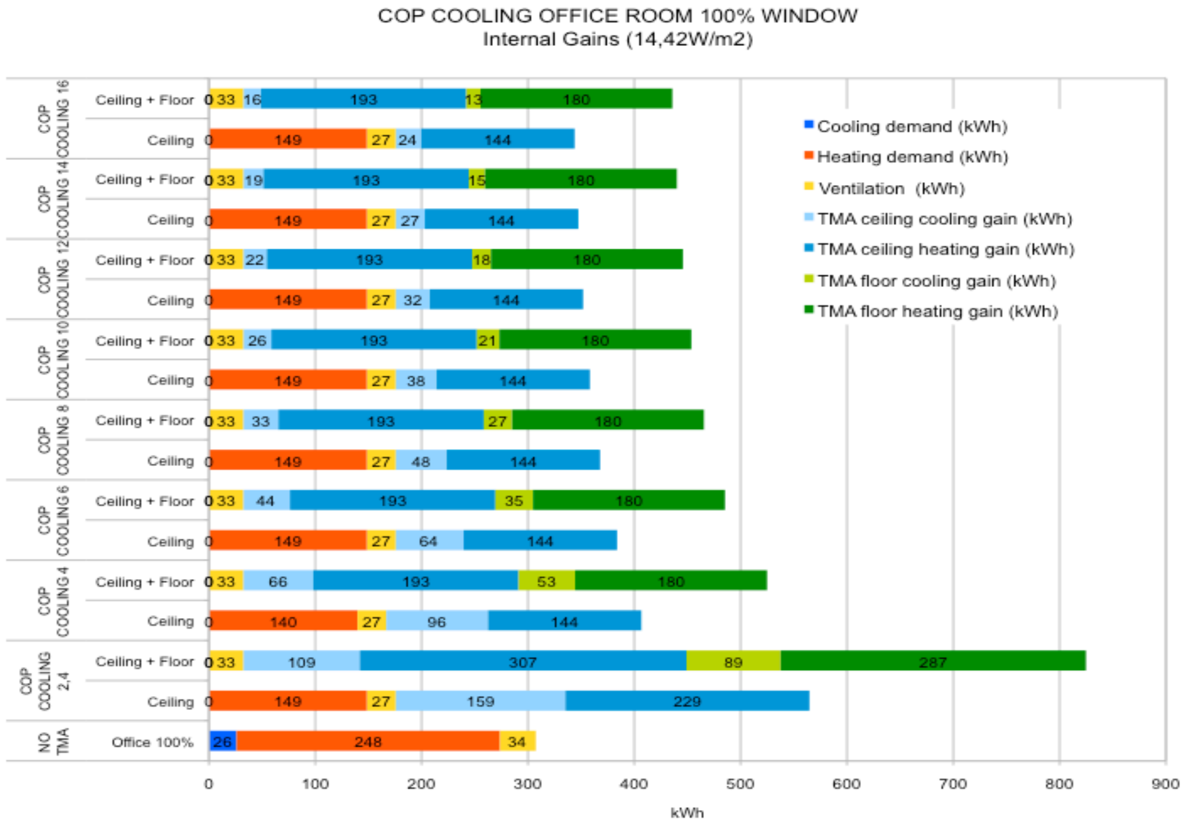
For the COP calculations of COOLING 2.4 there is a value of 2.2 used for TMA heating. This is according to NS3031 for floor heating. The other calculations assume a COP of 3.5 for TMA heating. The energy source for this high COP is a water-to-water heat pump where COP values can increase up to 5.6, (Dimplex.de).

Conclusion

The break-even point between no-TMA and TMA occurs at a COP COOLING of 8. Results for TMA are improved by increasing the COP values. According to a study done by Mark Murphy, TMA in combination with free cooling can reach COP values of 19.6 but this is related to $60 W/m^2$ internal gains.

Office room with 100% window

The following figure 31 shows the COP calculations performed on an office with 100% window. The same strategy is used as before with the office with 40% window.



	Office Room 100% Window and TMA, 80% ExternalShading $2h^{-1}$ and H.R. ventilation 85% ($0.3h^{-1}$									
	NO TMA	COP COOLING 2,4		COP COOLING 4		COP COOLING 6		COP COOLING 8		
COP HEATING TMA 3,5	Office 100%	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	
Cooling demand (kWh)	25.92	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Heating demand (kWh)	247.86	148.70	0.00	139.70	0.00	148.70	0.00	148.70	0.00	
Ventilation (kWh)	33.64	27.08	32.55	27.08	32.55	27.08	32.55	27.08	32.55	
TMA ceiling cooling gain (kWh)		159.48	109.28	95.69	65.57	63.79	43.71	47.84	32.79	
TMA ceiling heating gain (kWh)		229.47	307.27	144.24	193.14	144.24	193.14	144.24	193.14	
TMA floor cooling gain (kWh)			88.55		53.13		35.42		26.57	
TMA floor heating gain (kWh)			287.13		180.48		180.48		180.48	
Total (kWh)	**307.42**	**564.73**	**824.79**	**406.70**	**524.87**	**383.81**	**485.31**	**367.86**	**465.52**	

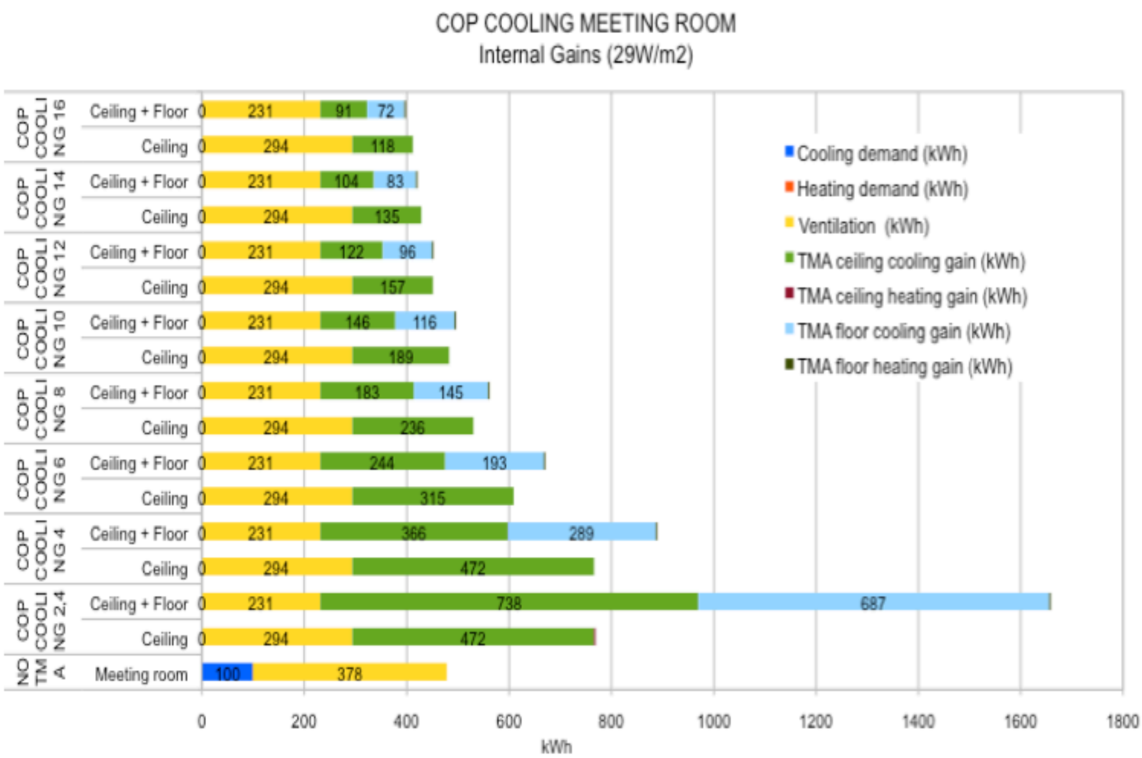
	NO TMA	COP COOLING 10		COP COOLING 12		COP COOLING 14		COP COOLING 16	
	Office 100%	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor
Cooling demand (kWh)	25.92	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating demand (kWh)	247.86	148.70	0.00	148.70	0.00	148.70	0.00	148.70	0.00
Ventilation (kWh)	33.64	27.08	32.55	27.08	32.55	27.08	32.55	27.08	32.55
TMA ceiling cooling gain (kWh)		38.28	26.23	31.90	21.86	27.34	18.73	23.92	16.39
TMA ceiling heating gain (kWh)		144.24	193.14	144.24	193.14	144.24	193.14	144.24	193.14
TMA floor cooling gain (kWh)			21.25		17.71		15.18		13.28
TMA floor heating gain (kWh)			180.48		180.48		180.48		180.48
Total (kWh)	**307.42**	**358.29**	**453.65**	**351.91**	**445.74**	**347.36**	**440.09**	**343.94**	**435.85**

Fig. 31. Shows different COPs for TMA cooling and heating in an office with 100% window

Figure 31 shows that it is not possible to obtain internal gains as low as with the office with 40% window; a cooling load always exists in the office with 100% window.

Meeting room with 40% window

The following table 32 shows the COP calculations performed on a meeting room with 40% window. The same strategy will be used as in the office with 40% window.



	Meeting Room 40% Window and TMA, 80% ExternalShading and Heat recovery 85%								
	NO TMA	COP COOLING 2,4		COP COOLING 4		COP COOLING 6		COP COOLING 8	
COP HEATING TMA 3,5	Meeting room	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor
Cooling demand (kWh)	100.10	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating demand (kWh)	0.29	10.78	0.00	10.78	0.00	10.78	0.00	10.78	0.00
Ventilation (kWh)	377.9	294	231.4	294	231.4	294	231.4	294	231.4
TMA ceiling cooling gain (kWh)		472.36	737.64	472.36	365.57	314.90	243.72	236.18	182.79
TMA ceiling heating gain (kWh)		.01	0.00	0.00	.27	0.00	0.00	0.00	0.00
TMA floor cooling gain (kWh)			686.54		289.45		192.97		144.73
TMA floor heating gain (kWh)			0.30		0.30		0.30		0.30
Total (kWh)	**478.29**	**777.15**	**1655.88**	**777.14**	**887.00**	**619.68**	**668.38**	**540.96**	**559.21**
	NO TMA	COP COOLING 10		COP COOLING 12		COP COOLING 14		COP COOLING 16	
	Meeting room	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor	Ceiling	Ceiling + Floor
Cooling demand (kWh)	100.10	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Heating demand (kWh)	0.29	10.78	0.00	10.78	0.00	10.78	0.00	10.78	0.00
Ventilation (kWh)	377.9	294	231.4	294.00	231.40	294	231.4	294	231.4
TMA ceiling cooling gain (kWh)		188.94	146.23	157.45	121.86	134.96	104.45	118.09	91.39
TMA ceiling heating gain (kWh)		0.00	0.00	0.01	0.00	0.00	0.00	0.00	0.00
TMA floor cooling gain (kWh)			115.78		96.48		82.70		72.36
TMA floor heating gain (kWh)			0.30		0.00		0.30		0.30
Total (kWh)	**478.29**	**493.72**	**493.71**	462.24	449.74	**439.74**	**418.85**	**422.87**	**395.46**

Fig. 32. Shows different COP's for TMA cooling and heating in a meeting room

These results show that from a COP of 12 for TMA cooling, the internal gains for TMA in the ceiling lower are. Cooling load is non-existent in this room while in the office with no

TMA, there is a cooling load from 100kWh. This indicates that that the indoor climate in a space with integrated thermal active elements is better.

Conclusion

COP is an important issue that has to be taken into account for TMA due to fact that the efficiency on this technology is very high. This is mainly due to the high efficiency of free cooling. With the use of free cooling, COPs can increase up to 19.6, since the COP is closely related to the amount of internal gains and the thereby related cooling load.

CHAPTER 4 : DISCUSSION

4.1. Summary

The following tables summarize the break-even results for each situation. Output data from TRNSYS16 show the two best TMA results and these results will be compared and analysed with the no TMA situation. The following colours are used in the output data:

- Red Indoor temperature °C
- Light green Direct solar gains, southwest facing surface Watt
- Light blue Diffuse solar gains, southwest facing surface Watt
- Purple Heating demand Watt

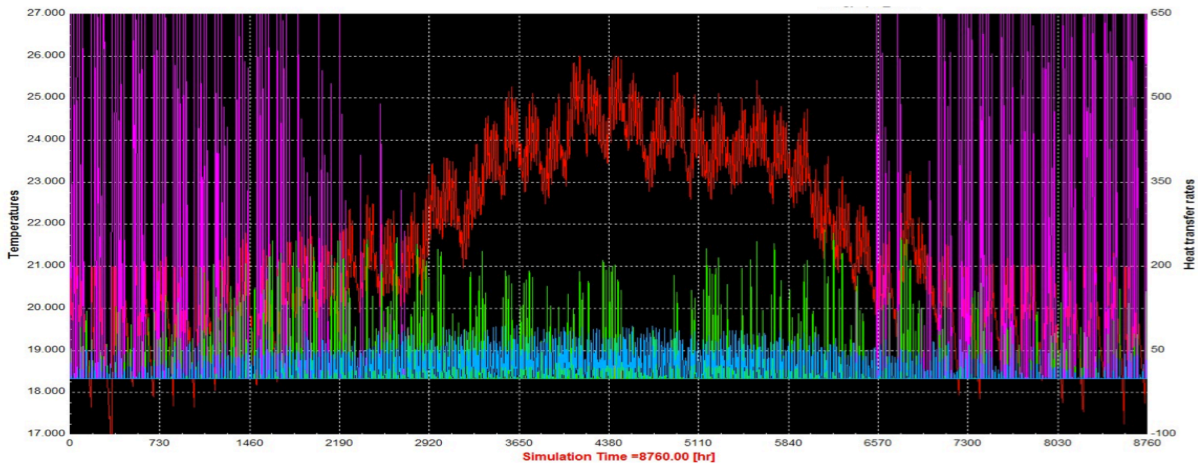
4.1.1. Office 40% window

Table 12 will show a summary from an office with 40% window.

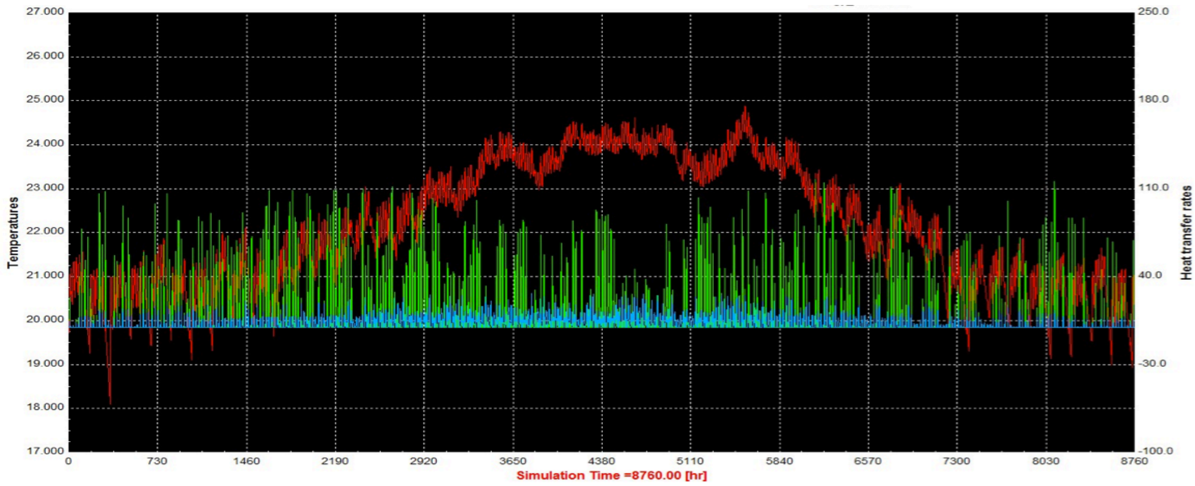
PARAMETERS	No TMA	TMA Office 40% Window		
	Office 40%	Ceiling	Floor	Ceiling + Floor
Floor surface (m2)	8.75	8.75	8.75	8.75
Orientation South – West	232	232	232	232
Occupation 12/5/52 (h)	3120	3120	3120	3120
Internal gains (W/m2)	14.42	14.42	14.42	14.42
Upper dead band shading (W/m2)	140	140	140	140
Lower dead band shading (W/m2)	120	120	120	120
Static shading %	80	80	80	80
Convective heat	60%	15%	15%	15%
Radiant heat	40%	85%	85%	85%
Heating Til/Cooling On	20C-26C	20C-26C	20C-26C	20C-26C
Tour – retour temp. Heat ventilaiton	45C – 35C	45C – 35C	45C – 35C	45C – 35C
Temperature of airflow	21	20	20	20
Airchange of ventilation (1/h)	2	2	2	2
Ach per person (m3/h)	26	26	26	26
Heat recovery (%)	85	85	85	85
Humidity	50%	50%	50%	50%
Mass Flow Rate ceiling (kg/h)		30		65
Mass Flow Rate floor (kg/h)			30	85
Inlet temperature		22	22	23
COP District heating (NS3031)	0.84			
COP Cooling TMA		8	8	10
COP Heating TMA		3.5	3.5	3.5
	NO TMA	TMA Office 40% Window		
	Office 40%	Ceiling	Floor	Ceiling + Floor
Cooling demand	1.67	0.00	0.00	0.00
Heating demand	84.58	23.70	24.10	0.00
Ventilation	41.91	33.95	32.80	28.66
TMA ceiling cooling gain		26.17		6.88
TMA ceiling heating gain		43.18		50.37
TMA floor cooling gain			30.37	15.71
TMA floor heating gain			54.50	24.61
Total (kWh)	**128.17**	**127.00**	**141.77**	**126.23**

Table 12 Shows the summary of an office with 40% window

No TMA



TMA in the ceiling



TMA in floor and ceiling

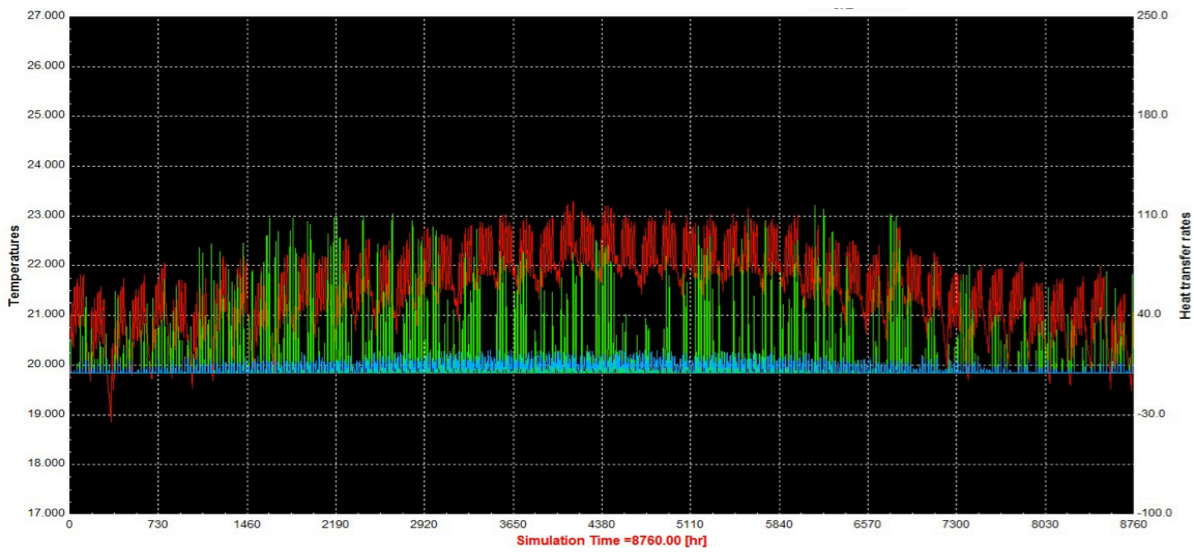


Fig.33 Shows the output data from TRNSYS for an office with 40% window

Conclusion

Table 12 and the output data from TRNSYS (fig.33) are a graphical representation of the results from table 12. Output data from no TMA show that the indoor temperatures during the summer period is mostly maintained between 25℃ and 26℃. These results were compared with the results where TMA is installed in the ceiling. In this scenario, the indoor temperatures stay between 24℃ and 25℃, indicating a better indoor climate.

The best indoor climate is realised by having active elements in both the ceiling and the floor. The indoor climate stays between 20℃ and 23℃ over the entire year. This results in the best indoor climate while concurrently maintaining sufficient quantities for daylight. This is due to the 80% external shading that is controlled by the dead band shading device as described in section 2.2.2.

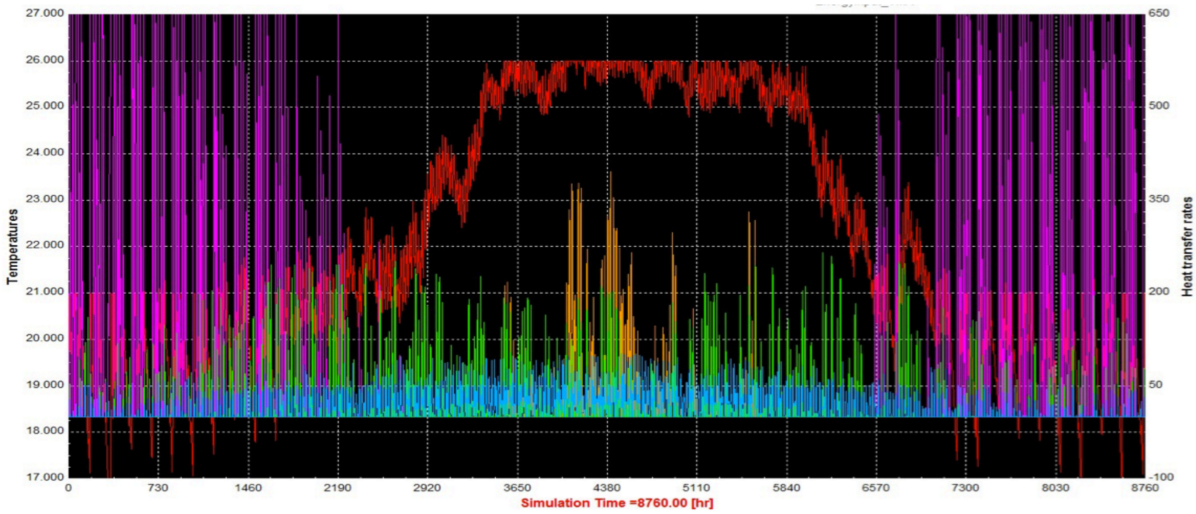
4.1.2. Office room with 100% window

Table 13 will show a summary from an office with 100% window.

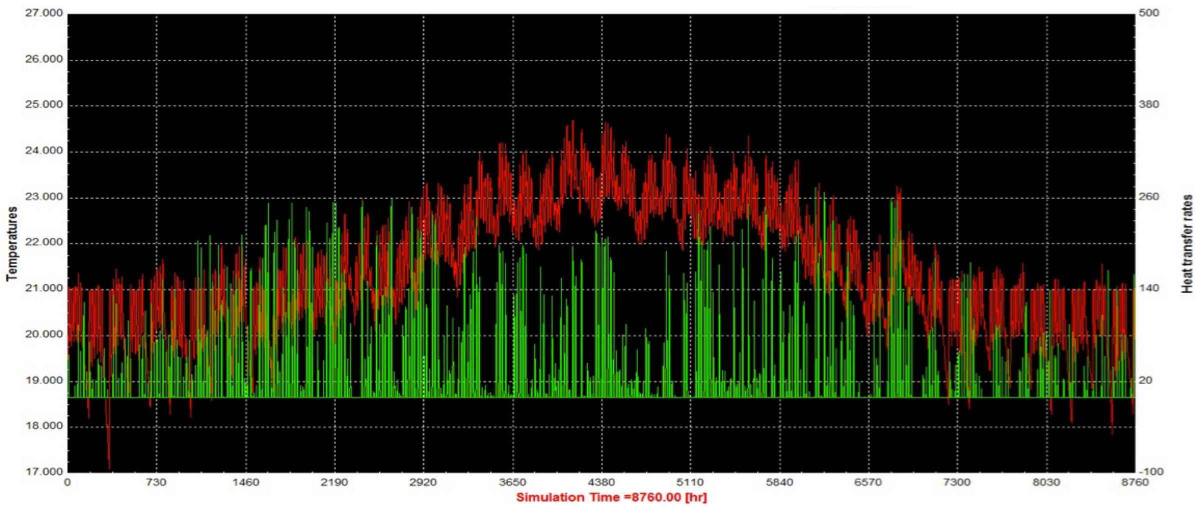
PARAMETERS	No TMA	TMA Office 100% Window		
	Office 100%	Ceiling	Floor	Ceiling + Floor
Floor surface (m2)	8.75	8.75	8.75	8.75
Orientation South – West	232	232	232	232
Occupation 12/5/52 (h)	3120	3120	3120	3120
Internal gains (W/m2)	14.42	14.42	14.42	14.42
Upper dead band shading (W/m2)	140	140	140	140
Lower dead band shading (W/m2)	120	120	120	120
Static shading %	80	80	80	80
Convective heat	60%	15%	15%	15%
Radiant heat	40%	85%	85%	85%
Heating Til/Cooling On	20C-26C	20C-26C	20C-26C	20C-26C
Tour – retour temp. Heat ventilaiton	45C – 35C	45C – 35C	45C – 35C	45C – 35C
Temperature of airflow	20	20	20	20
Airchange of ventilation (1/h)	2	2	2	2
Ach per person (m3/h)	26	26	26	26
Heat recovery (%)	85	85	85	85
Humidity	50%	50%	50%	50%
Mass Flow Rate ceiling (kg/h)		65		85
Mass Flow Rate floor (kg/h)			85	75
Inlet temperature		22	22	22
COP District heating (NS3031)	0.84			
COP Cooling TMA		8	8	8
COP Heating TMA		3.5	3.5	3.5
	No TMA	TMA Office 100% Window		
	Office 100%	Ceiling	Floor	Ceiling + Floor
Cooling demand	25.92	0.00	0.00	0.00
Heating demand	247.86	148.70	143.90	0.00
Ventilation	33.64	27.08	28.22	32.55
TMA ceiling cooling gain		47.84		32.79
TMA ceiling heating gain		144.24		193.14
TMA floor cooling gain			86.26	26.57
TMA floor heating gain			152.63	180.48
Total (kWh)	**307.42**	**367.86**	**411.01**	**465.52**

Table 13 Shows the summary of an office with 100% window

NO TMA



TMA in the ceiling



TMA in floor and ceiling

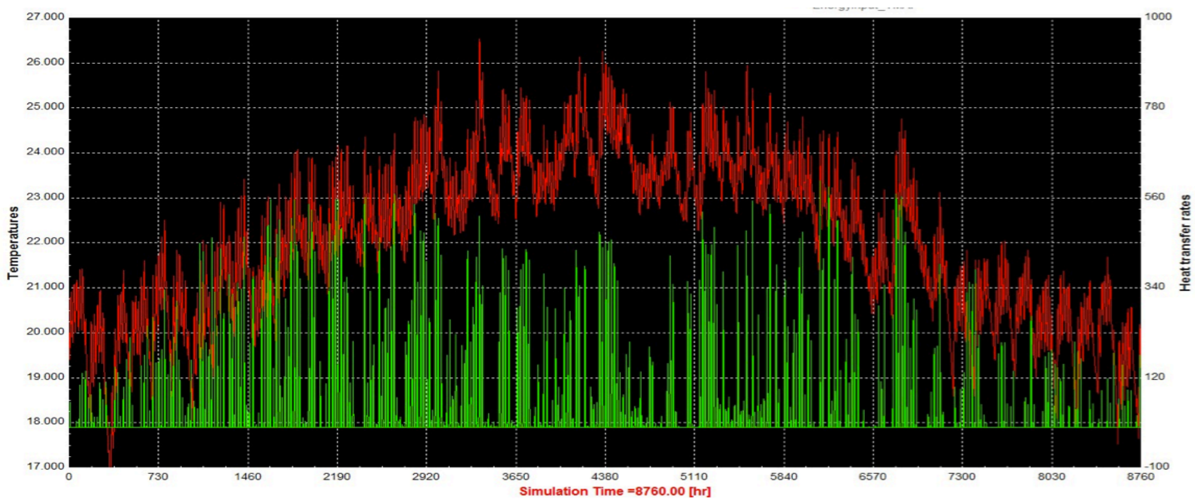


Fig.34 Shows the output data from TRNSYS for an office with 100% window

Conclusion

This table 13 and output data (fig.34) show that the ceiling is the best location for having integrated TMA. This statement can also be confirmed by comparing the results of the yearly schedule, see 2.4.1 and 2.4.2. A good yearly shading schedule for external shading will have a strong influence on maximizing the benefit of passive solar gains. There is still a high need for heating, however, since the transmission heat loss is very high. This is due to the large transparent surface area in the office room with 100% window, as shown in section 4.5.2. The transparent surface has a U-value of 0.7W/m^{2}K while the U-value of the opaque surface is 0.15 W/m^{2}K. This room therefore requires a good cooling and shading plan and TMA in the ceiling, which is the best location for optimizing a good indoor climate.

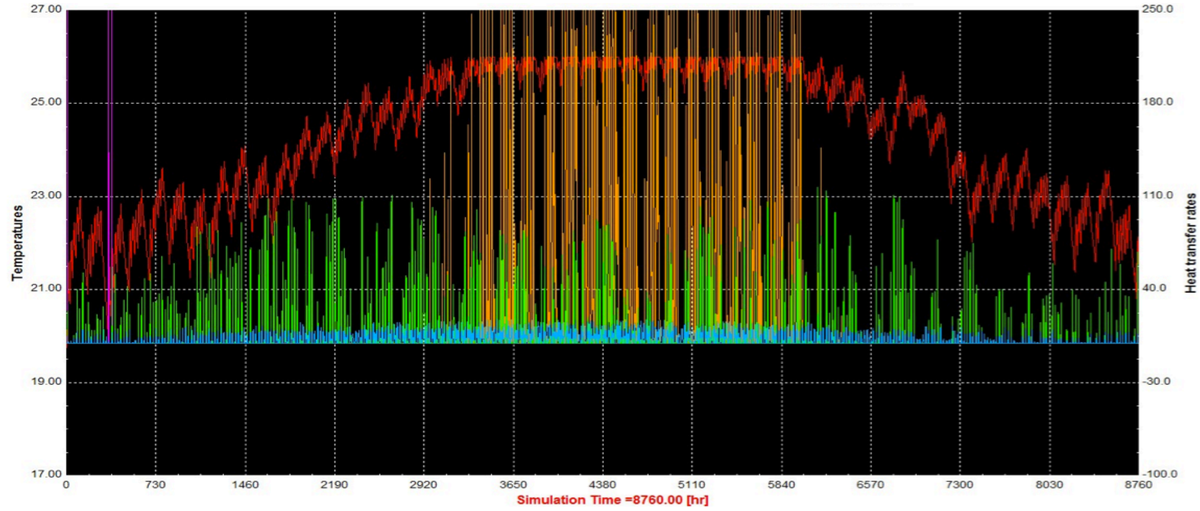
4.1.3. Meeting room with 40% window

Table 14 will show a summary from a meeting room with 40% window.

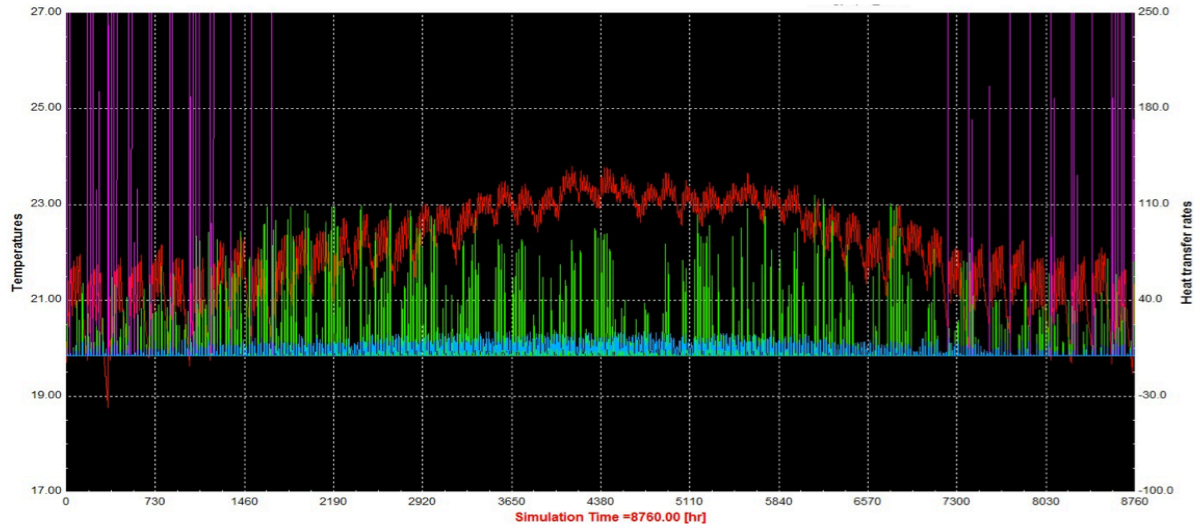
PARAMETERS	No TMA	TMA Meeting Room 40% Window		
	Meeting room	Ceiling	Floor	Ceiling + Floor
Floor surface (m2)	17.52	8.75	8.75	17.52
Orientation South – West	232	232	232	232
Occupation 12/5/52 (h)	3120	3120	3120	3120
Internal gains (W/m2)	28.21	14.42	14.42	28.21
Upper dead band shading (W/m2)	140	140	140	140
Lower dead band shading (W/m2)	120	120	120	120
Static shading %	80	80	80	80
Convective heat	60%	15%	15%	15%
Radiant heat	40%	85%	85%	85%
Heating Til/Cooling On	20C-26C	20C-26C	20C-26C	20C-26C
Tour – retour temp. Heat ventilaiton	45C – 35C	45C – 35C	45C – 35C	45C – 35C
Temperature of airflow	20	19	19	19
Airchange of ventilation (1/h)	2	3.5	3.5	3.5
Ach per person (m3/h)	26	26	26	26
Heat recovery (%)	85	85	85	85
Humidity	50%	50%	50%	50%
Mass Flow Rate ceiling (kg/h)		85	85	85
Mass Flow Rate floor (kg/h)				65
Inlet temperature		20	20	20
COP District heating (NS3031)	0.84			
COP Cooling TMA		12	12	12
COP Heating TMA		3.5	3.5	3.5
	No TMA	**TMA Meeting Room 40% Window**		
	Meeting room	**Ceiling**	**Floor**	**Ceiling + Floor**
Cooling demand	100.10	0.00	0.00	0.00
Heating demand	0.29	10.78	7.99	0.00
Ventilation	377.9	294	298.87	231.4
TMA ceiling cooling gain		157.45		121.86
TMA ceiling heating gain		0.01		0.00
TMA floor cooling gain			151.69	96.48
TMA floor heating gain			6.12	0.00
Total (kWh)	**478.29**	**462.24**	**464.67**	**449.74**

Table 14 Shows the summary of a meeting room with 40% window

NO TMA



TMA in the ceiling



TMA in floor and ceiling

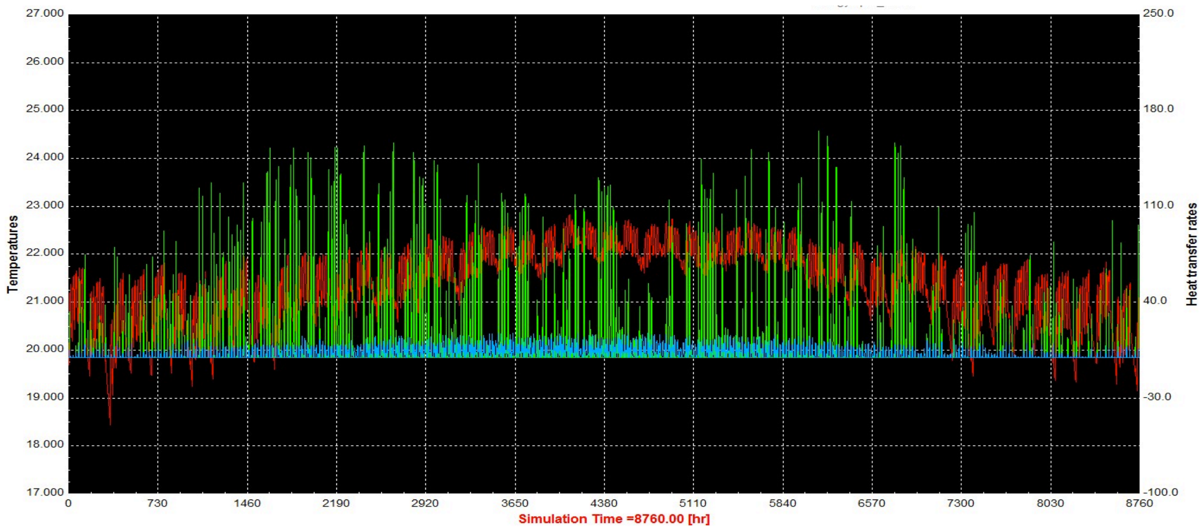


Fig 35 Shows the output data from TRNSYS for a meeting room with 40% window

Conclusion

Table 14 and the output data from TRNSYS (fig.35) shows how efficient it is to install TMA in a room with a large internal gains. TMA installed only in the ceiling removes the need for cooling. The boundary temperature for cooling can also be lowered to 24℃. When TMA is integrated in both the ceiling and the floor, the boundary temperature for cooling can be further lowered to 23℃. This creates the best indoor climate of all of the three models investigated. The ventilation strategy is in accordance to the TEK10 Norwegian building regulations.

4.2. Discussion

Three TRNSYS models were used to perform this parametric study related to the Power House One project in Trondheim. This project has the goal of being the most northern plus energy project not only in Norway, but also in the world. Thereby it was a great opportunity for investigating whether implementing thermal mass activation can contribute to reaching this goal. This has already been demonstrated in the more southerly located plus energy project, "Company Headquarters" established in Berlin, Germany. It was thus interesting to determine whether thermal mass activation technology is also suited for a project that is located so far north.

Further investigation is necessary to determine whether there could be an improvement in the energy efficiency.

- Further study on improving the solar shading, e.g. with a 150– 250 lower-upper dead band range
- Determining why ventilation is reduced by increasing the water mass flow rate for improving TMA
- What kind of ventilation system is the most compatible with this technology? Would centralised or decentralised ventilation be of most benefit?
- What is the most efficient way to operate the system, e.g. for 10min every hour so, thereby it making the most use of the inertia capacity of thermal mass.
- A more accurate parametric study on the coefficient of performance for TMA.
- It would be interesting to have more information regarding the energy efficiency of TMA, since TMA contributes greatly to creating a good indoor climate.

- Further studies on the geometry of the embedded pipe layout. This issue was neglected in this study due to the limited available research data.

CHAPTER 5 : CONCLUSION

The goal of this thesis was to determine whether thermal mass activation is an appropriate technology in the Norwegian building sector. Its effect on the indoor thermal comfort in the Norwegian climate was also studied. This technology is already well integrated in Central Europe and has shown to have great benefits in these countries. These benefits are mainly attributable to the high cooling loads that exist in office buildings in these climates. This thesis thereby also focused on the issue of cooling load.

Several parametric studies were conducted in order to determine the values of certain variables to optimize the installation and operation of a TMA system. Some of these studies also served to quantify the benefit of TMA over other technologies. The conclusions from these studies are described below.

Heat and cooling capacity in relation to ceiling and floor

The cooling and heat capacity of the ceiling and floor were simulated for determine the optimal installation location for cooling and heating using TMA systems. By calculating the heat capacity according to the NS-EN15377 standard, it can be stated that the floor has a higher heat capacity during the winter, while during the summer the ceiling has a higher cooling capacity. This is mainly due to the fact there is always a high amount of convective heat; the cooling load will always rise to highest point in the space, thereby making the ceiling the best location for integrating TMA and tackling the cooling load in the most efficient way.

Low thermal mass and high thermal mass

For this parametric study, TMA was neglected in order to isolate the effect of thermal mass. A parametric study was done on a low thermal mass building and a high thermal mass building. This study came to conclusion that high thermal mass is better for a building that has a higher occupation and therefore a high cooling load. A lightweight construction with low thermal mass needs twice the amount of heat gain to reach the same indoor climate as a heavyweight construction with high thermal mass. This leads to the conclusion that in a building with high thermal mass, it is easier to maintain a stable temperature than in a lightweight building. This is also the reason why office buildings are better suited to having a high thermal mass than a normal house. This is directly related to the occupancy of each building type. Since a lightweight building, such as a wooden house, is usually unoccupied during the day, there is no need for time delayed heat to be radiated from the construction. A lightweight building also heats up faster.

External shading

The goal of this parametric study is to determine the optimal percentage of external shading. The best performance of the external shading for the office with 100% window is seen when the percentage of the external shading varies for each season of the year. By allowing more passive solar gains during the wintertime, the heating demand will be reduced. Section 3.3.4, which discusses the yearly schedule, gives a good overview of the impact of this strategy. The room with 100% window needs special attention because of the high heat loss through the window.

A good cooling and ventilation strategy will be necessary for a room with high internal gains. The ventilation heat was lowered from 20°C to 19°C and the air change rate increased from 2 to 3.5 h^{-1}. By installing TMA, the cooling load was eliminated and the ventilation requirements were reduced. The indoor climate improved as well, as the indoor climate never exceeded 25°C (see fig.17 output data TRNSYS). Further investigation of the coefficient of performance (COP) suggests that the total energy demand will also decrease. A COP of 4 for TMA cooling was used in this diagram. See fig.17.

Shading controlled by a dead band shading controlling system

Shading has a considerable influence on the indoor climate. A good shading strategy is thus an important issue in the development of a project.

The external shading in this case will be activated when the horizontal solar gains reach 140W/m^2 and can be deactivated when they are less than 120 W/m^2. A coefficient of performance for TMA heating, 2.2 (NS 3031) and cooling 4 (NS EN 14511) were assumed.

A yearly schedule has a high potential to reduce of the total amount of heat that is needed by optimizing the use of passive solar gains. The indoor temperatures were also always kept under the boundary temperatures of 25 degrees C. This, together with the optimal use for day lighting, significantly increases the indoor climate quality.

Water mass flow rate

Controlling the water mass flow rate through the TMA pipes, can also remove the need for cooling. The boundary temperatures for cooling can also be lowered significantly by controlling this variable. The results demonstrate that the ceiling is the best location for this technology due and is the most efficient place for reducing the cooling load.

Cooling load

The room with 100% window did not have a significantly higher cooling load than the office with 40% window. The reason for this is that the increased solar gains in the internal space somewhat neutralize the heat loss. This is directly related to the 80% external shading assumed in the model; otherwise, the cooling load in the office with 100% window would be significantly higher.

Relation between ventilation and TMA

This technology is also more efficient than ventilation in this model because they do not reach the same indoor climate. Ventilation also reduces the air quality to an undesirable level. More research is necessary on this issue because these studies points out that the need for ventilation is reduced significantly with integrated TMA. The reason why this occurs should also be investigated in a future research project.

Coefficient of performance (COP)

COP is an important issue that must be considered with TMA due to fact that the efficiency on this technology is very high. This is mainly due to the high efficiency of free cooling. With the use of free cooling, the COPs can reach up to 19.6, however, the COP is closely related to the amount of internal gains and the thereby related cooling load. More research is necessary to better understand the relationship between COP and TMA.

It can be concluded that integrating thermal mass activation is a valuable component for improving the thermal comfort of an indoor space. While its performance is better than that of ventilation for activating the thermal mass, there is not thus far a clear answer as to whether TMA is more energy efficient than activating this same thermal mass by ventilation heat. Further research is necessary to investigate this issue.

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