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Rasmus Z. Høseggen

Dynamic use of the building structure - energy performance and thermal environment

NTNU
Norwegian University of
Science and Technology
Thesis for the degree of
philosophiae doctor
Faculty of Engineering Science and Technology
Department of Energy and Process Engineering



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PREFACE

The work for this thesis was carried out at the Department of Energy and Process Engineering at the Norwegian University of Science and Technology (NTNU) during the period from July 2004 to October 2007. The work is a part of a joint project between NTNU, Sintef Energy Research, The Research Council of Norway, and accompanying business partners called DKL – a Norwegian abbreviation for Laboratory for Indoor Climate.

First of all, I would like to express my gratitude to my supervisor Professor Sten Olaf Hanssen at the Department of Energy and Process Engineering at NTNU for his valuable advice, encouragement and discussions. I would also like to thank my co-supervisors Adjunct Professor Hans Martin Mathisen at Department of Energy and Process Engineering at NTNU and Dr. Ing. Bjørn Wachenfeldt at Sintef Building and Infrastructure for invaluable help, and their ability to point out important matters and asking the right questions. I am also very indebted to my friends and colleagues at the department for inspiration through this period. I am grateful for the English editing assistance from senior adviser Stewart Clark, NTNU.

I acknowledge the support from Statsbygg (The Directorate of Public Construction and Property) for financing the experimental part of this study. Also I would like to thank the always helpful and co-operative operational personnel at the case study building at Røstad.

Finally, I offer warm thanks to family and friends for encouragement, and not least, for withstanding my absence in the finishing stages of the PhD work – I have a lot of catching up to do!

Trondheim, December 2007
Rasmus Z. Høseggen

ABSTRACT

The main objectives of this thesis have been to evaluate how, under which premises, and to what extent building thermal mass can contribute to reduce the net energy demand in office buildings. The thesis also assesses the potential thermal environmental benefits of utilizing thermal mass in office buildings, i.e. reduction of temperature peaks, reduction of temperature swings, and the reduction in the number of hours with excessive operative temperatures. This has been done by literature searches, and experimental and analytical assessments. This thesis mainly concerns office buildings in the Norwegian climate. However, the methods used and the results obtained from this work are transferable to other countries with similar climates and building codes.

The total energy use for the operation of the Norwegian building stock is approximately 82 TWh in a normal year, or about 38 % of the national onshore energy use. No other sector has experienced a larger growth in the energy use in the past 30 years.

As a consequence of the Norwegian decision to join the European Economic Area, Norway was obliged to implement the EU Energy Performance of Buildings Directive (EPBD) in its national laws and regulations. The Norwegian building codes were at the time also undergoing a revision, and the EPBD became to some extent a guiding principle for the new regulations and guidelines. While the former regulations only concerned the heating energy demand in a building, the new regulations incorporate all the energy needed to operate the building. The new regulations are based on the *net energy demand* per year, i.e. the efficiencies of the energy systems are not taken into account. In addition, the self-generated renewable energy, e.g. electricity from PV-panels, use of solar collectors is not rewarded. However, passive measures that reduce the *net energy demand* contribute to satisfy the requirements. This has led to renewed interest in utilizing passive measures to satisfy the regulations and decrease the total energy use in buildings.

In this study, it is assessed how and to which extent building thermal mass can contribute to reduce the net energy demand and improve the thermal environment in office buildings. The thesis comprises a review of existing literature in the field, a parametric study employing an advanced simulation model in ESP-r, and an experimental field study on a modern office building in operation.

Within the limitations of this thesis and based on the findings from all parts and papers this thesis comprises, it is shown that utilization of thermal mass in office buildings reduces the daytime peak temperature, reduces the diurnal temperature swing, decreases the number of hours with excessive temperatures, and increases the ability of a space to handle daytime heat loads. Exposed thermal mass also contributes to decrease the net cooling demand in buildings. The quantity of the achievements is dependent on the amount of exposed thermal mass, night ventilation strategy, and airflow rates. In addition, parameters such as set-point temperatures, control ranges, occupancy patterns, daytime ventilation airflow rates, and prevailing convection regimes are influential for the achieved result. The importance of these parameters are quantified and discussed.

In contrast to some studies referred to in the literature, this study shows that thermal mass only has a minor influence on the total heating demand in office buildings. Although heavy buildings may utilize excessive heat during working hours to decrease the heating demand outside working hours, buildings with high heating capacity lose some of its energy saving potential by night temperature set-back compared to buildings with lower heating capacity.

Consequently, it can be concluded that thermal mass *contributes* to; 1) fulfil the net energy frame for office buildings, 2) eliminate the need for space cooling, and 3) improve the thermal climate, thus increase working performance in office buildings without the need for energy intensive and expensive technical installations.

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ABBREVIATIONS

A/C	Air conditioning
ACH	Air changes per hour
BEMS	Building energy management system
CAV	Constant air volume
DCV	Demand controlled ventilation
DHC	Diurnal heat capacity
DRY	Design reference year
EC	Exposed concrete
EN	European Norm
EPBD	Energy Performance of Buildings Directive
ESP-r	Energy Systems Performance (r stands for research version)
HVAC	Heating, ventilation and air-conditioning
ISO	International Organization for Standardization
LCA	Life cycle assessment
LCC	Life cycle cost
LW	Long wave (infrared radiation)
MFR	Mass-to-floor ratio
NS	Norwegian Standard
PCM	Phase changing materials
REN	Guidance for the Norwegian building regulations
RH	Relative humidity
SC	Suspended ceiling
SP	Set-point
TEK	Norwegian building regulations
TMY	Typical meteorological year
TRY	Test reference year
TTC	Thermal time constant
VAV	Variable air volume
WYEC	Weather year for energy calculations

DEFINITIONS

Control range

The interval in which the ventilation airflow rate may vary with the temperature in order to cool a space.

Energy demand

In contrast to energy use, which is actual energy use of a building, the energy demand is the calculated or estimated amount of energy to operate a building.

Free cooling

Cooling of a space by the supply of unconditioned ventilation air.

Internal heat load

Heat emitted from people, lighting, equipment, and processes inside a building.

Net cooling demand

The cooling demand of a building without consideration of the efficiencies or losses in the cooling system, i.e. the surplus heat that must be removed to achieve the desired thermal conditions.

Net energy demand

The energy demand of a building without consideration of the efficiencies or losses in the energy system.

Operative temperature

Uniform temperature of the imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual non-uniform environment (ISO 7730).

Ventilative cooling

Space cooling by the supply of conditioned or unconditioned ventilation air.

1. INTRODUCTION

Before mechanical ventilation, cooling, and artificial lighting were introduced in buildings in the beginning of 1900s, various techniques were used to climatize buildings. Examples of such techniques are temperature-driven natural ventilation, active use of window ventilation, large room volumes or large room heights, utilization of thermal mass, thought-through window and frame design for maximum utilization of daylight while minimizing solar gains.

During the last few decades, several of these techniques have been reintroduced in order to reduce or eliminate the need for mechanical ventilation and cooling. However, the requirements for comfort, indoor air quality and energy efficiency show an increasing trend. The earlier passive techniques thus need to be adapted to modern buildings.

1.1 Background

In the last few years, there has been an increased focus on CO₂ emissions and the world's increasing energy use. The building sector represents a considerable share of the total energy use, and Norwegian authorities have stated that it is a political priority to reduce the energy use in the building sector (KRD, 2005). Revision of the building regulations is one measure for improving energy efficiency, and aims at reducing the energy demand for new buildings by 20-30 % and the energy use in existing buildings by at least 6 TWh (about 7 %) within 2020 (Dagestad, 2007). However, this may also have consequences for the indoor environment.

Experiences from the measures taken during the oil embargos in the 1970s showed that there is only a short distance between *energy economizing* and uncritical *energy savings*. Reduced ventilation and sealing of houses resulted in increased dampness and indoor mould growth. The association between dampness in buildings and health is evident (Sundell et al., 2003).

The interaction between the building, the technical installations and the building users must be understood in such a way that sub-optimal solutions are avoided. At the same time, it is important to keep in mind that only a few are interested in saving energy, but many are interested in saving money.

The definition of energy economizing made by the Norwegian Ministry of Petroleum and Energy in the early 1980s is equally relevant today:

“Energy economizing is a concept which means that energy should be used in its form, quantity, and to the time that totally is most profitable when all advantages and disadvantages are weighed”

1.1.1 State of Norwegian building energy use

The total energy use for operation of the Norwegian building stock is approximately 82 TWh in a normal year, or about 38 % of the national onshore energy use. No other sector has experienced a larger growth in the energy use in the past 30 years (Enova, 2007). Figure 1-1 shows the total energy use distributed between dwellings and industrial buildings, and the share of which is for heating purposes and of which is electrical heating energy use.

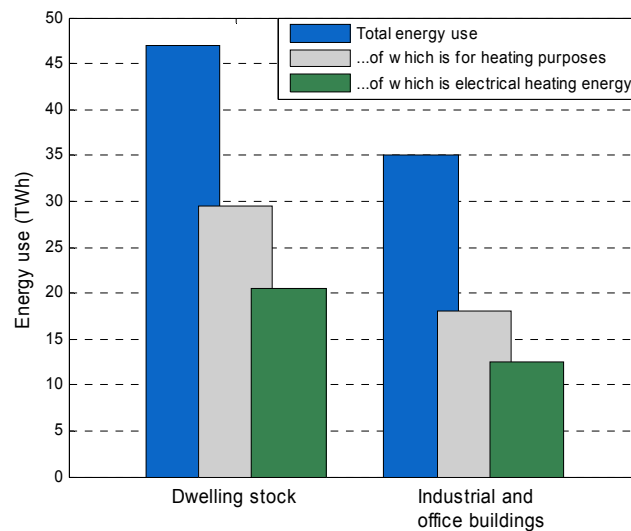


Figure 1-1 Energy use in the building sector in Norway (Enova, 2007)

The national building regulations in Norway have been revised several times since the first numerical requirements were introduced in 1949. The purpose of the recurrent upgrades has basically been to reduce the heating demand, thus reducing the overall energy use in buildings (Thyholt, 2006). However, even though the insulation thicknesses in walls, floors and roofs have been multiplied by the order of four in accordance with the Norwegian building codes in the last 38 years, the specific energy use is not reduced. In fact, according

to the yearly energy statistics published by Enova (2007), the energy use in new office buildings has increased. Other building categories do not show the same tendency. Figure 1-2 presents the climate corrected mean specific total energy use distributed by the year of construction and the number of buildings each category includes. The age groups reflect the year the building energy regulations were revised.

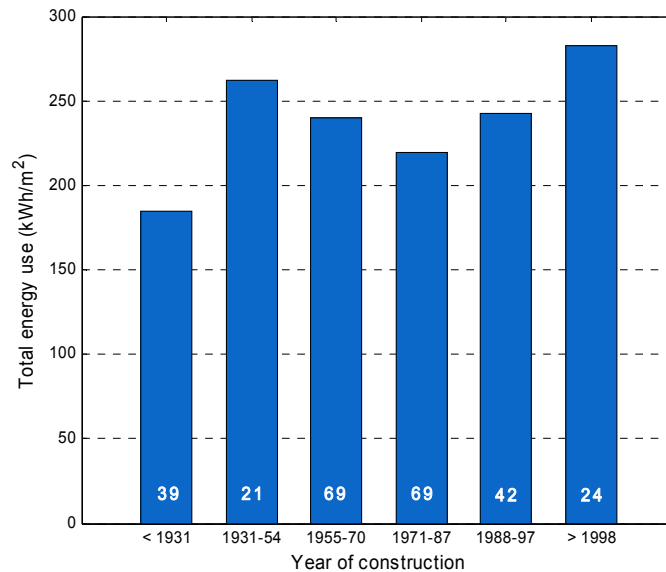


Figure 1-2 Mean total specific energy use for office buildings distributed by the year of construction, with the number of buildings each category includes inscribed in the columns (Enova, 2007)

The statistics in Figure 1-2 are based on quality ensured reports from a number of buildings, but the buildings are not stochastically selected, thus are not representative for the total national building stock. However, the tendency is evident towards increased energy use despite stricter requirements. There may be several reasons for this, but important factors are likely to be:

- Increased use of large glass façades, which generally both leads to increased heating and cooling energy demands.
- Increased demands on quality indoor climate, which implies large air volume flow rates and/or high cooling energy use.
- Due to low energy prices, generally low focus on energy efficiency and building operation costs.

- Increased energy use for lighting and equipment
- Increased area efficiency.

1.1.2 New building energy regulations

As a consequence of the Norwegian decision to join the European Economic Area, Norway is obliged to implement the EU Energy Performance of Buildings Directive (EPBD, 2002) in its national laws and regulations. The Norwegian building codes were at the time also undergoing a revision, and the EPBD became to some extent a guiding principle for the new regulations and guidelines. While the former regulations only concerned the heating energy demand in a building, the new regulations incorporate all the energy needed to operate the building. Figure 1-3 illustrates in principle the energy flow through a building.

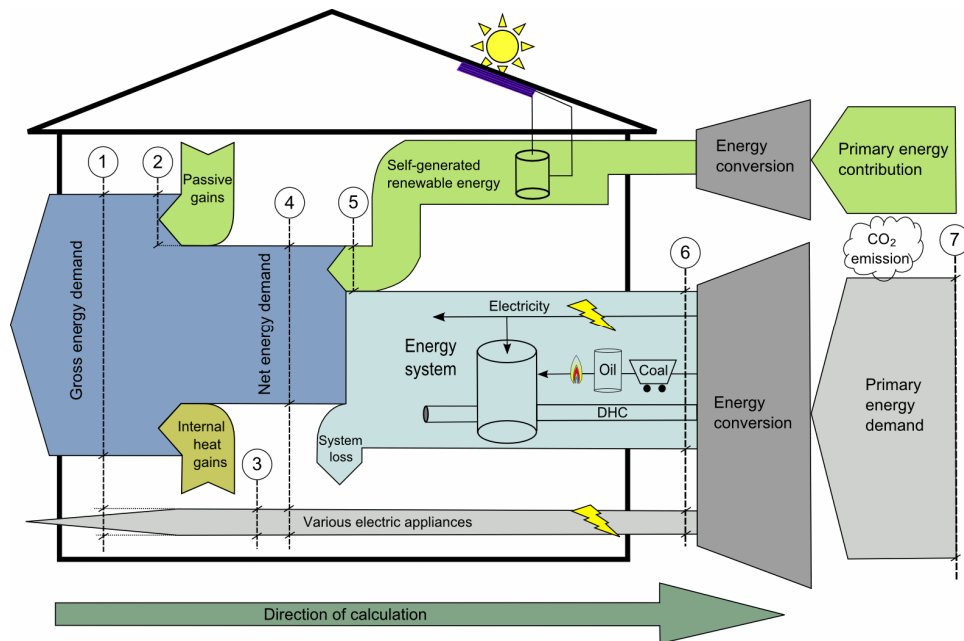


Figure 1-3 System boundaries for energy calculation (Adapted from Lexow, 2007)

Prior to the revision of the building energy regulations (TEK, 2007), the placement of the system boundary for assessment of the energy performance of buildings was broadly discussed. One of the main intentions of the EPBD is to reduce the primary energy use of buildings (point 7 in Figure 1-3). The disadvantage of putting the system boundary far out in the energy chain, is that it opens for a potential loophole for a technological shift inside the building, i.e. the requirements could be fulfilled by putting more effort into a highly

efficient energy system than enduring solutions for the building envelope. Focus in the adopted regulations was to consolidate a high quality building envelope with a low heat loss coefficient, ensuring that the choice of solution is a built-in quality. Hence, the *net energy demand* (Point 4 in Figure 1-3) was agreed to be the measure for fulfilling the building energy regulations.

There are two ways a building can fulfil the new energy regulations. One method is to employ the so-called *Energy measure method*, which sets requirements to certain building elements and installations. Alternatively, if the *net energy demand* for the building, calculated according to the methodology established in the Norwegian Standard NS3031 (2007), is within the *net energy frame* for the category of the building, the regulations are also satisfied. The new regulations apply to all new buildings and buildings undergoing major reconstructions or refurbishments, and became operative from February 2006.

Table 1-1 *Energy frame for different building categories (TEK, 2007)*

Building category	Net energy frame (kWh/m ²)
Small detached dwellings	125*
Apartment buildings	120
Kindergartens	150
Office buildings	165
Schools	135
Universities and colleges	180
Hospitals	325
Nursing homes	235
Hotels	240
Sporting halls	185
Cultural buildings	235
Industrial buildings and workshops	185
* (+ 1600 /available area)	

Since the *net energy frame* is based on *net energy demand* per year, the efficiencies of the energy systems are not taken into account. In addition, self-generated renewable energy, e.g. electricity from PV-panels, use of solar collectors etc (Point 5 in Figure 1-3), is not rewarded. However, passive gains (Point 2 in Figure 1-3) that reduce the *net energy demand* will contribute to satisfy the energy frame. This has led to renewed interest in utilizing passive measures to decrease the total energy use in buildings.

1.1.3 Thermal environment

The relation between indoor climate and health is documented in a number of reports and considerable amounts of money are lost every year due to poor indoor climate. In relation to thermal climate, the economic loss is connected to a reduction of working performance and productivity. The building regulations state that the thermal climate should give satisfactory health conditions and comfort for the intended use of the room (TEK, 2007). However, there is a difference in the interval of comfort and the interval of highest performance. The interval of comfort can be changed somewhat by adjusting the level of clothing. Few, if anyone, adjust their clothing to generate higher productivity.

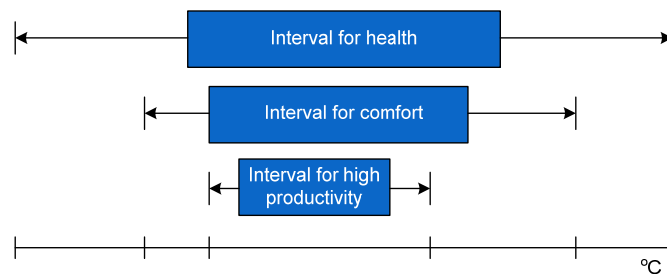


Figure 1-4 Schematically, the temperature intervals for health, comfort, and productivity

According to Guidebook No. 6 published by REHVA (2006), staff costs are about 100 to 200 times the cost of energy and 20 to 44 times the cost of the HVAC running costs in offices. Thus, a relative small increase of productivity will constitute much greater economic gain than a decrease in the energy bill. On influence of temperature on performance, the REHVA Guidebook refers to Wyon and Wargocki (2006), who state that;

“Room temperature affects the performance by several mechanisms:

- *thermal discomfort distracts attention and generates complaints that increase maintenance costs*
- *warmth lowers arousal, exacerbates SBS symptoms and has a negative effect on mental work*
- *rapid temperature swings have the same effects on office works as slightly raised room temperatures, while slow temperature swings only cause discomfort”*

Results from a number of studies on non-industrial buildings show that performance increases with temperatures up to 21-22 °C, and decreases with temperatures above 23-24 °C. Highest productivity is found to be at temperatures around 22 °C (Seppänen et al., 2006). Figure 1-5 shows the composite weighted relation between the normalized performance and temperature, based on reports from in all 24 studies with different weighting dependent on the relevance to normal office work (Seppänen et al., 2006).

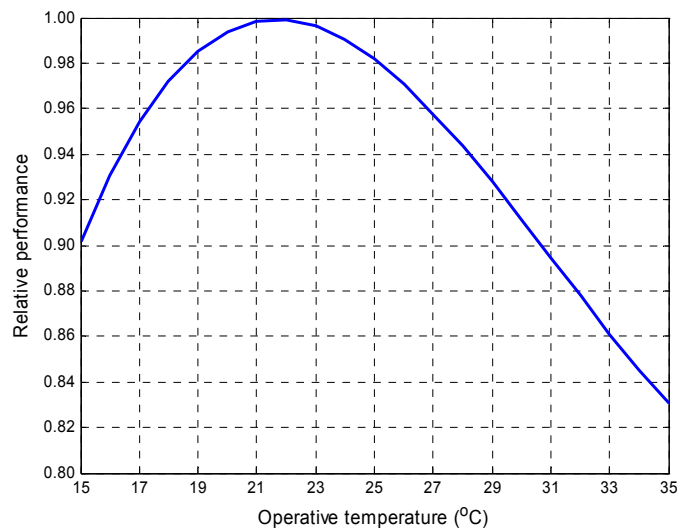


Figure 1-5 Normalized performance vs. temperature for typical office work (Seppänen et al., 2006)

Increasingly use of heat-emitting equipment in office buildings has together with a demand for highly insulated building envelopes led to a growing concern that it may lead to overheating even during wintertime. In addition, the new building regulations state that buildings should be constructed without use of local space cooling, which makes use of passive measures such as exposed thermal mass in combination with night ventilation even more relevant.

1.2 Objectives and limitations

The main objectives of this thesis are to:

- evaluate how, under which premises, and to what extent thermal mass can contribute to reduce the net energy demand in office buildings, and
- evaluate the possible thermal environmental improvements by utilize thermal mass in office buildings, i.e. reduction of temperature peaks, reduction of temperature swings, and reduction of the number of hours with excessive operative temperatures.

The underlying objectives are to:

- identify and discuss thermal mass properties,
- identify and discuss the main parameters influencing on the efficiency of thermal mass, and
- identify and discuss barriers for utilizing thermal mass.

The thesis uses searches in the literature and experimental and analytical assessment.

This thesis is mainly concerned with office buildings in the Norwegian climate. However, the methods used and the results obtained from this work are transferable to other countries with similar climates and building codes.

1.3 Outline of the thesis

The thesis constitutes four main parts, which can be read independently from each other;

Part I gives a brief introduction regarding the physical mechanisms involved and important parameters affecting the performance of thermal mass. This part also summarizes studies done on the subject found in the literature.

Part II is a parametric study employing a simulation model. In this part, the effects of several parameters are studied on the efficiency of thermal mass for a single office cell.

Part III is an extended description of a field study that formed the basis of Paper V, ‘*The effect of suspended ceilings on energy performance and thermal comfort*’, which is enclosed in Appendix 4. The chapters in this part will give some supplementary description on the instrumentation, reliability of the measurements, and the achieved results.

Part IV aims at synthesizing the different findings from all parts and papers in this thesis. It discusses them and points out the main findings and final conclusions from the work as a whole.

PART I:

LITERATURE STUDY

UNDERSTANDING THERMAL MASS

The chapters in this part will give a brief introduction regarding the physical mechanisms involved and important parameters affecting the performance of the thermal mass. This part also summarizes work on the subject found in the literature.

Chapter 2 will briefly go through the most important physical mechanisms necessary to have knowledge of when assessing thermal mass effectiveness. This chapter also describes how different factors influence on the efficiency of thermal mass.

Chapter 3 assesses some of the barriers and practical implications of using heavy constructions in buildings.

Chapter 4 summarizes in tabular form experimental and analytical studies from the literature that evaluate the effectiveness of thermal mass in different climates and building types.

2. THERMAL MASS PRINCIPLES

The air temperature in a room is dependent on the outdoor temperature and the solar radiation, together with the internal heat gains from people, lighting and technical equipment indoor. During a day this typically results in a temperature variation with a peak in the early afternoon, when the outdoor temperature is at its highest, the sun is lower in the horizon and the building has been in use for some time.

On a warm summer day, high outdoor temperatures, high noise or pollution levels may exclude the use of natural measures to dispose of excessive heat in a building. In order to keep the thermal environment at an acceptable level, heat has to be removed from the building with mechanical equipment that uses energy, in most cases electrical energy.

To reduce the indoor temperature and reduce the need for cooling, it is possible to store some of the excessive heat in the building construction. These thermal storages are the building fabric itself and will from this point on be referred to as *thermal mass*. Thermal mass constitute the building fabric in external walls, ceilings, floors, partitions and even furnishing with high thermal capacitance. Typical materials that contribute significantly to the thermal mass in a building are concrete, brick, metals and to some extent wood.

Thermal mass is often classified into two types: internal and external. The internal mass is not exposed to ambient temperature directly, while the external thermal mass such as walls and roofs are directly exposed to ambient temperature variation. In this thesis, if not explicitly underlined, thermal mass will refer to the *internal* mass of a building. The external mass is of less interest in cold climates. There are two reasons for that. Firstly, the building envelopes are very well insulated, thus changes in the external temperature will not have a great influence on the indoor temperature. Secondly, the ambient temperature rarely exceeds the indoor comfort limits. Although the sun can warm an exterior surface up to 10 to 20 degrees above the ambient temperature (Clarke, 2001), other factors, such as solar radiation through windows may constitute a greater influence on the indoor thermal conditions.

Thermal mass can give a positive contribution to the indoor environment both in summer and winter. In the wintertime, energy from the sun and internal heat gains can be absorbed in the thermal mass during the day, and released slowly to the indoor air at night, thus

partially reducing the need for heating. In the summer time, excessive heat is absorbed and reduces the need for cooling during the daytime. The absorbed heat will gradually be released when the temperature decreases during the night, thus reducing large temperature variations during the day. Buildings that are unoccupied during the evening and night may be cooled down in order to empty the thermal storages and heat may be absorbed during the following day (Balaras, 1995).

Thermal mass may also have a positive impact on thermal comfort. Firstly, buildings with a high degree of thermal mass dampen large temperature swings during the day and sustain a steadier thermal environment (Nicol, 2004). Secondly, in winter, the air temperature can be lower than the surface temperatures. A lower air temperature will, in addition to reducing the heat loss, also improve the perceived indoor air quality (Fang, 2004)

2.1 Heat storage and heat transfer mechanisms

This section will briefly consider the basic physical principles of heat storage and heat transfer related to thermal mass. At the end, different ways to approximate the laws of physics and the implementation into building simulation programs are discussed.

2.1.1 Heat storage

Heat storage takes place in all materials in the building. The specific heat capacity (c_p), density (ρ) and thickness (d) determine the amount of heat required to elevate the temperature by one degree for a material. In a multi-layer building component, the total heat capacity is calculated by the sum of heat the capacity of each layer i , i.e.:

$$C_T = \sum_n d_i \rho_i c_{p,i} \quad (\text{J/m}^2 \text{ K}) \quad (2.1)$$

However, the order of the layers as well as their thermal conductivity (λ), make an impact. For instance, if a wall with high total heat capacity is insulated on the side facing the room, just a limited amount of heat will be absorbed and conducted to the inner layers, and the wall will behave similarly to a lightweight construction. Conversely, if the insulation is on the outside, heat can be absorbed from the indoor environment, and the wall (considered from the inside) is thermally heavy, even though the total heat capacity for the two walls is the same.

Thermal diffusivity (κ) and *thermal effusivity* (β), defined in (2.2) and (2.3) respectively, are useful dynamic performance indicators (Clarke, 2001). The *diffusivity*, also called the thermometric conductivity, describes how fast a heat wave travels through a material. Materials with high *effusivity*, i.e. materials with a high heat penetration coefficient, will more readily absorb a surface heat flux than low-effusivity materials.

$$\kappa = \frac{\lambda}{\rho c_p} \quad (\text{m}^2/\text{s}) \quad (2.2)$$

$$\beta = \sqrt{\lambda \rho c_p} \quad (\text{Ws}^{1/2} / (\text{m}^2 \cdot \text{K})) \quad (2.3)$$

The diffusivity and effusivity can be applied to real constructions by reducing the multiple layers to an equivalent homogeneous layer (Clarke, 2001). Table 2-1 shows these thermal parameters for some common building materials.

Table 2-1 Properties for common building materials (Clarke, 2001; Davies, 2004)

Material	Conductivity λ (W/m K)	Density ρ (kg/m ³)	Specific heat capacity c_p (J/kg K)	Thermal diffusivity $\kappa \times 10^8$ (m ² /s)	Thermal effusivity β (W s ^{1/2} /m ² K)
Glass ¹	1.05	2500	750	56	1403
Steel ¹	45	7800	480	1202	12980
Aluminum ¹	203	2700	880	8544	21962
Heavyweight concrete ²	1.30	2000	840	77	1478
Lightweight concrete ²	0.2	620	840	38	323
Brick, inner leaf ²	0.62	1800	840	41	968
Gypsum plaster ²	0.16	800	1090	19	385
Plywood ³	0.15	700	1420	15	386
Pine, fir ³	0.12	510	1380	17	291
Rock wool ⁴	0.033	100	710	46	48
Extruded polystyrene ⁴	0.035	25	1470	95	36
Linoleum ¹	0.19	1200	1470	11	579
Synthetic carpet ¹	0.06	160	2500	15	49

¹ Impermeables, materials unaffected by water content

² Inorganic porous materials, highly dependent on the water content

³ Organic materials highly dependent on the water content

⁴ Materials dependent on the water content, but should be protected from wetting under normal circumstances

By comparing the effusivities for concrete, gypsum and extruded polystyrene from Table 2-1, it may be observed that they are in order of 40 : 10 : 1. Idealized, this means that an increase in the temperature at the surface, concrete will absorb heat 40 times more readily than rock wool, and about 4 times more than gypsum.

2.1.2 Heat transfer

Heat can be distributed within a space by convection (forced or natural) and by radiation (short-wave and long-wave).

Surface convection is heat exchange between a surface and the adjacent air layer. On internal surfaces, both natural and forced convection can occur (Clarke, 2001). The natural forces can result from heat sources, e.g. radiators, people and equipment, and from surface-to-air temperature differences. Forced air flow is generated by fans or wind entering from outside. The convective heat transfer between the air and a surface area is given by (2.4), where h_c is the convective heat transfer coefficient and ΔT is the temperature difference between the surface and the room air.

$$q_{conv}^* = h_c \Delta T \quad (\text{W/m}^2) \quad (2.4)$$

Surfaces in a room exchange heat by long-wave (infrared) radiation. The long-wave radiation exchange depends on the temperatures of the surfaces (T_i), and to which extent they are in visual contact, determined by the view-factor (f_{j-i}). The exchange is also dependent on the emissivities of the materials (ε), which determines how easily the surface emits heat. The net long-wave heat exchange between two grey (normal) surfaces is given by (2.5), where σ is the Stefan-Boltzmann constant ($5.67 \cdot 10^{-8} \text{ W/m}^2 \text{ K}^4$).

$$q_{rad}^* = \varepsilon \sigma (T_1^4 - T_2^4) f_{2 \rightarrow 1} \quad (\text{W/m}^2) \quad (2.5)$$

Heat gain from the sun constitutes a significant portion of the total heat gain in most buildings. The treatment of shortwave heat flux can therefore greatly influence the accuracy of the overall performance.

The total solar radiation striking an object is the sum of the following three contributions (Balcomb, 1992):

1. Direct normal radiation: radiation coming directly from the sun
2. Diffuse sky radiation: radiation scattered by the atmosphere (e.g. air molecules and water droplets)
3. Reflected radiation (e.g. ground reflection)

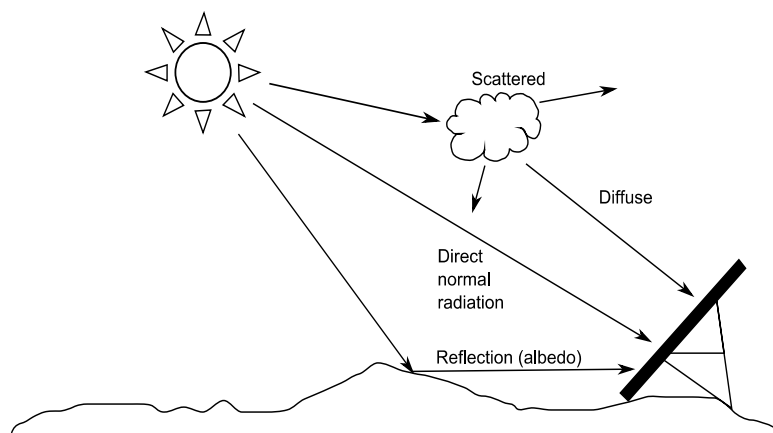


Figure 2-1 Total solar radiation striking an object

Solar radiation which strikes a transparent surface, such as a window, is partially reflected, some is absorbed in the glass layer and will cause a rise in its temperature, and some is transmitted and will strike some internal surface. At the internal surface some portion is absorbed in the material and some is reflected. If the internal surface is a transparent surface, a part will also be transmitted onward to another zone or outside. An accurate prediction of the influence of the sun requires methods for the prediction of the surface position relative to the moving pattern of the insolation of the surfaces. Information on the short-wave absorptivity for opaque elements and absorptivity, transmissivity and reflectivity for transparent elements is needed (Clarke, 2001).

2.1.3 Treatment of heat storage and heat transfer in building simulation programs

In order to predict the dynamic behaviour of a system and to calculate the resultant indoor temperature taking into account all the parameters just described, a computer program must be used. Figure 2-2 shows a cross section of a floor and the complex situation of heat transfer processes that are involved. Note that the figure shows the situation before people, lighting, equipment and airflows are brought in, which will complicate matters further.

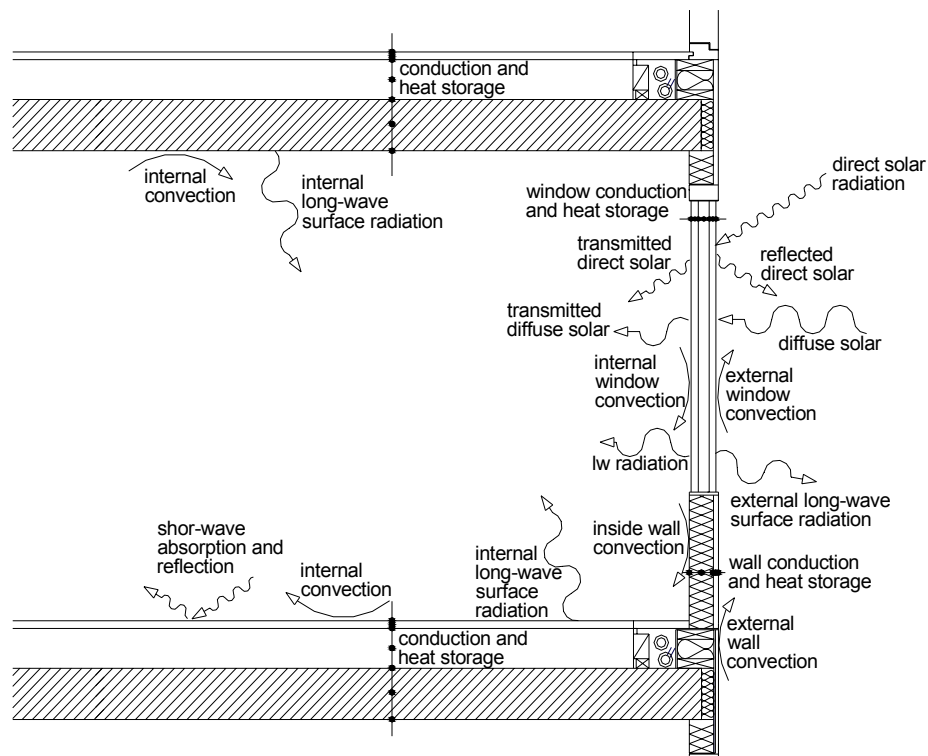


Figure 2-2 Cross section of a floor showing the heat transfer processes involved

Most building simulation programs approximate the physics of nature to some extent. Figure 2-3 shows an RC (resistance-capacitance) representation of a building zone in its simplest form. The thermal mass is lumped in one node, which represents the equivalent thermal mass for the zone. As will be discussed later, the placement of the thermal masses is not indifferent and all approximations done will affect the calculated result.

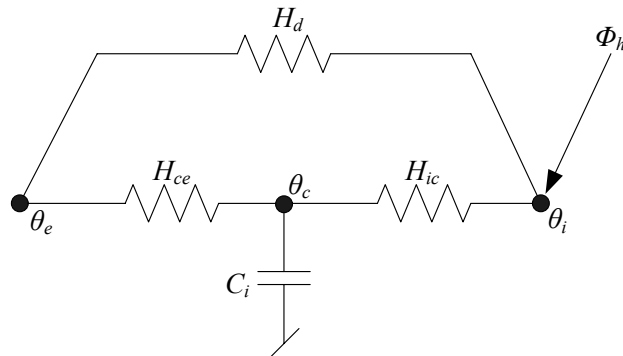


Figure 2-3 Example of a simple dynamic RC network

- θ_i = internal air temperature
- θ_c = structure temperature
- θ_e = external temperature
- H_d = direct heat loss coefficient
- H_{ce} = heat loss coefficient between the structure and the external
- H_{ic} = heat loss coefficient between the structure and the heated space
- C_i = internal heat capacity
- Φ_h = heat gains and heating power

In simplified energy modelling methods, convective heat transfer coefficients are often assumed to be time invariant and valid over an entire surface. Long-wave radiation is often simplified by the use of a linearized radiative heat transfer coefficient. Short-wave radiation in simplified methods is treated as a long-term average on the basis of design values listed in technical handbooks and standards. Since not all solar gains are usable in terms of decreasing the heating demand of a building, it is common practice to introduce a utilization factor, e.g. according to NS-EN ISO13790 (2000).

There have been several studies on the sensitivity of internal convection. The studies have shown that predicted heating and cooling energy demand may vary up to 40 % depending on the correlation used (Davies, 2005). Advanced dynamic simulation tools, such as ESP-r (ESRU, 2007) and EnergyPlus (DoE, 2007), have correlations which calculate the convective heat transfer coefficients by each simulation time-step from the temperature difference between the surfaces and the adjacent air, surface roughness, direction of the heat flow and characteristic dimensions (Zmeureanu, 1998). However, even these programs

must be treated with caution. Beausoleil-Morrison I. (2002) studied the influence of convection correlations common in building simulation programs, taking into account location of heating devices and HVAC equipment, and when they were operated. The study revealed that energy use for heating and cooling in a typical office building in Canada in reality was 19 % greater than the predicted demand when using the ESP-r with the default correlation for convective heat transfer. Implementing modified heat transfer correlations in ESP-r, and taking into account the additional information on location of heating and HVAC equipment, reduced the difference from 19 to 9 %. Moreover, Beausoleil-Morrison concludes that an improper choice of heat transfer correlation could lead to inappropriate design decisions.

There are differences in the treatment of solar radiation in simulation programs. Some tools calculate the transmitted solar radiation through glazed areas, and distribute the gain diffusely and area-weighted to all interior surfaces, while advanced modelling tools have the ability to ray-trace multiple reflections of solar beams for each time step (ESRU, 2007). In a study of a glazed space by Wall (1997), four different simulation programs were compared. The study revealed that the simpler tools tend to overestimate the heat gain from short-wave radiation and therefore are improper to be used in spaces dominated by glazed exterior surfaces.

Bellamy and Mackenzie (2003) point out that the capability of a building simulation program to accurately predict the effects of thermal mass very much depend on its capability to model solar gain. This statement is supported by their comparison of monitored and simulated energy use in a heavy-mass test building. The advanced building simulation tool BSim2000 (SBI, 2007), which has the solar ray tracing facility, estimated the heating energy demand to only 2.3 % higher than the measured value. The simpler tool Suncode (Ecotope, 2007) which uses a fixed distribution of the solar gain, underestimated the heating energy demand by 18 %.

2.2 Parameters affecting the efficiency of thermal mass

The effectiveness of thermal mass is influenced by several other parameters than the properties of the thermal mass itself. To achieve the best result, it is important to have an understanding of how the parameters affect and interact with the thermal mass. In the following sections, some of the most important factors and their impact on thermal mass efficiency are described.

2.2.1 Climate

In general, the application of thermal mass strategy is more suitable in climates with large diurnal temperature differences to take advantage of evening heat release, or nighttime cooling of the structure. Szokolay (1984) suggests that the appropriate diurnal variation should be at least 10 K in order to get the desired effect. Givoni (1998b) claims that night cooling is mainly applicable in regions with a diurnal temperature swing of more than 15 K, and where the night minimum temperature in summer is below 20°C.

Shaviv et al. (2001) investigated the influence of thermal mass and night ventilation in different Israeli climates. The study shows that the maximum indoor temperature is linearly dependent on the temperature difference between day and night. The relation can serve as a simple tool to estimate the potential of utilizing thermal mass and night ventilation given the diurnal temperature swing at different locations.

Pfafferott et al. (2003) refer to the Swiss handbook published by EMPA on passive cooling, which sets the limits on free cooling potential to 150 Wh/m² per day if the temperature difference between day and night is less than 5 K, and 250 Wh/m² per day if the difference is higher than 10 K.

A recent study by Artmann et al. (2007) evaluates the climatic potential for passive cooling by night-time ventilation in all climatic zones of Europe by analysing semi-synthetic climate data produced by Meteonorm (Meteotest, 2005). The study shows that there is a large potential for passive cooling in the whole of Northern-Europe. In Central, Eastern and even some regions in Southern Europe the potential is significant. However, a series of warmer nights can occur in some regions, making passive ventilation alone insufficient to guarantee thermal comfort all year round.

The authors point out that the method developed is applicable only during the design phase, and is not a substitute for building energy simulations, where thermal mass, solar and internal gains, and air-flow patterns is taken into account.

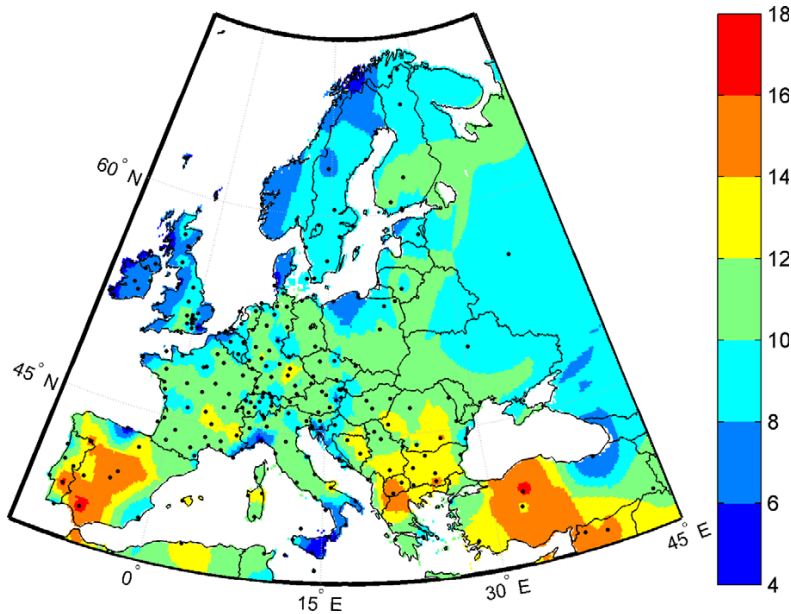


Figure 2-4 Mean differences between minimum and maximum temperatures (K) in July based on Meteonorm semi-synthetic climate data (Artmann et al., 2007)

It is claimed that there is no benefit from utilizing thermal mass in hot and humid climates, since the diurnal temperature swing is usually small and the mean relative humidity is around 80 %. This traditional point of view has been questioned, and recent studies have shown that thermal mass can also be useful in these climates (Goulart, 2004; Capeluto, 2005).

Mitchell and Beckman (1989) claim that if heating energy storage in the constructions is to have a significant effect on the energy consumption, it is important that the balance temperature is close to the average ambient temperature during the heating season. If the difference is greater than 9 K, storage will have no significant effect.

2.2.2 *Location, amount and distribution*

Location of the thermal mass is important. Thermally heavy materials exposed directly to the sun absorb heat more efficiently than materials absorbing heat indirectly by long-wave radiation and convection (Balaras, 1995). Vertical walls are better locations for thermal mass than floors and ceilings, because the heat transfer coefficient for convective exchanges to and from the surfaces of the walls are greater (Balcomb, 1992). However, if a room is equipped with supply air diffusers near the ceiling, this will have a great impact on the convective heat transfer, thus, making the ceiling a radiant cooling surface. Diaz (1994) evaluated the effect of thermal mass on the internal temperature of a single-zone space where location, amount and distribution of the thermal mass was explored. The parametric study concludes that the floor is the location where the thermal mass has least impact on the internal temperature. In zones with external boundaries on all four sides, mass in the roof is the most effective location. When only one wall is exposed to the external, it is on that wall the thermal mass has the greatest impact. Diaz' study shows that the internal walls are the least effective location, but as the zone structure becomes heavier, the location of the thermal mass loses its importance.

The total amount of heat a building component can absorb is according to Eq. (2.1) dependent on the exposed surface area and the thickness of the material. During a diurnal cycle, most of the recoverable heat is contained in the first 5 cm layer and thicknesses above 10 cm provide little additional effect, assuming the properties equal to heavyweight concrete (Balcomb, 1992). NS-EN ISO13786 (2000) approximates the effective thermal thickness as the minimum of a) half the total thickness of a component, b) the thickness of the material from the surface of interest and the first insulating layer exclusive coating layers, or c) depending on the period of the variations, 2 cm, 10 cm and 20 cm for 1 hour, 1 day and 1 week respectively. The basis for this recommendation is a material with a thermal diffusivity equivalent to $0.7 \times 10^6 \text{ m}^2/\text{s}^2$, which is near the value of heavyweight concrete (see Table 2-1).

In addition to the thickness, the thermal mass area is one of the most important parameters. The mass should be distributed to increase the efficiency of the mass, provided that all surfaces are in direct contact with the internal air (Hestnes, 2003). The mass-to-floor ratio (MFR) is convenient to evaluate the exposure of thermal mass to the internal air. According to Diaz (1994), higher MFR improves the thermal performance when the internal gains are

higher during the day than the night, and that exposure of thermal mass area is more effective than thickness to improve thermal conditions.

There have been several approaches to quantify the heat storage capability of a room or a building. Among them is the *Diurnal Heat Capacity (DHC)*, developed especially for passive solar houses (Balcomb, 1992). The *DHC* is a measure of the capacity of a building to absorb heat from the interior space, and to release the heat back to the space during the night hours. The *dhc* of a material is a function of the density, specific heat, conductivity and thickness. The total *DHC* of a building is calculated by summing the *dhc*-values of each surface exposed to the interior air, thus:

$$DHC = \sum_n dhc_i A_i \text{ (Wh/(m}^2\cdot\text{K))} \quad (2.6)$$

The expression for the *dhc* of a surface is rather complex and the equations are given in Appendix A1.1. In Figure 2-5 the diurnal heat capacities for various homogeneous building materials are plotted against the material thicknesses.

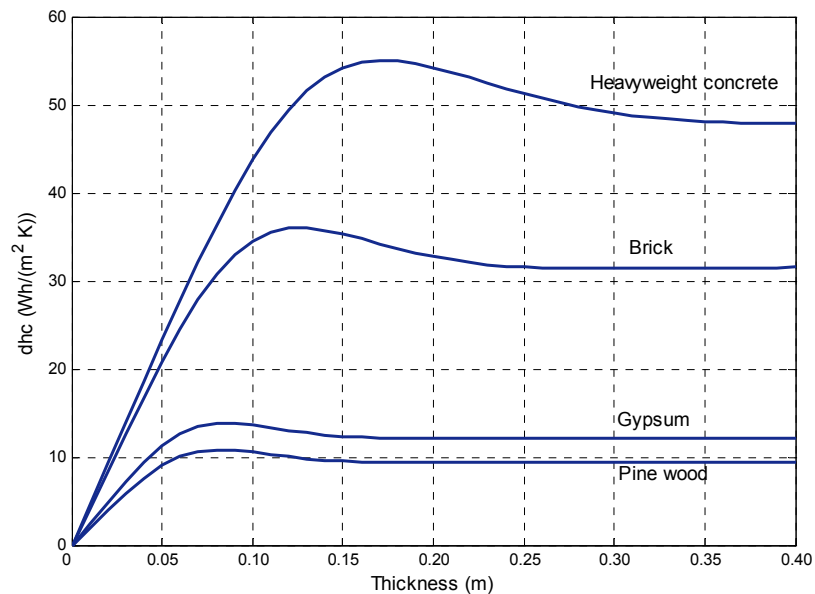


Figure 2-5 Diurnal heat capacity (*dhc*) vs. thickness for various storage media. The material properties are collected from Table 2-1

In Figure 2-5, it is worth noticing that the dhc for a material increases initially with the thickness, and then decreases at around 10-20 cm, depending on the material. This implies that there is an optimal thickness when it comes to maximum heat storage and release in a 24-hour cycle. The DHC-method can also be applied to multi-layer constructions (Balcomb, 1992).

Another measure the capability of a building to store heat is the *Thermal Time Constant* (TTC), defined as the product of the thermal resistance and heat capacity of a unit area of a building envelope element. The TTC is representative of the effective thermal capacity of a building (Givoni, 1998), or to be more precise, it characterizes the effective thermal mass of the building envelope. The total TTC_{tot} of the building envelope equals the sum of all the TTC_s of the surfaces divided by the total envelope area A_{tot} , thus

$$TTC_{tot} = \frac{\sum TTC_s}{A_{tot}} \quad (2.7)$$

Appendix A1.2 gives the expressions for the calculation of the TTC_s . A high TTC indicates a high thermal inertia, and results in a strong suppression of the interior temperature swing (Givoni, 1998). Examples of the use of the DHC and TTC are shown in the next section.

The above-mentioned study by Shaviv (2001) investigated the influence of thermal mass and night ventilation for detached houses with four different levels of thermal mass. The study shows that the maximum indoor temperature is reduced with an increase of thermal mass. However, the improvement was less significant going from a medium heavy structure to a heavy structure, than from a light to a medium heavy structure. The same observation is also made by Norèn et al. (1999), who studied the annual energy use for heating for three different buildings with equivalent U-values. The study concludes that just a small increase in the thermal mass in a building has a lowering effect on the specific heating energy requirement, but that the effect diminishes with a further increase in the thermal mass.

2.2.3 *Insulation and exposure*

The *DHC* and the *TTC*, defined in the previous section, can be used to evaluate the dynamic thermal performance of a building. The *TTC* is a measure on the effective heat capacity of a building when the heat flow across the opaque part of the building envelope is dominant. The *DHC*, on the other hand, indicates the thermal capacity for buildings where the internal and/or solar gains are considerable. Roughly, the *TTC* and *DHC* depend on the building construction as follows (Givoni, 1998b):

- Externally insulated with internal exposed mass: Both *TTC* and *DHC* are high. Heat can be absorbed during the day and, if ventilated at night, released during the night.
- Mass insulated internally: Both *TTC* and *DHC* are low. The thermal response of the building is similar to a light-weight building.
- Mass insulated externally and internally: *TTC* is high, while the *DHC* is negligible, since the internal insulation isolates the thermal mass from the interior.
- Core insulation inside two layers of mass: *TTC* is a function of the internal mass and the core insulation thickness, and the *DHC* is a function of the internal mass (*DHC* approximately similar to the first case).

The optimal order and thicknesses of thermal mass and insulation in building envelopes has been studied by several researchers. Kossecka and Kosny (2002) studied the effect of mass and insulation location for six different wall configurations by using DOE-2.1E to calculate the annual heating and cooling loads in different US climates. The results showed that the material configuration of the exterior wall significantly influence the thermal performance of the whole building. The best overall energy performance was found to be the external insulated configuration with internally exposed mass. The worst performance was the “all insulation inside” wall, which had an overall annual energy demand, depending on the climate, that was 2-11 % higher than “the all insulation outside” wall.

Asan (1999) had a slightly different approach. He investigated the optimum position of mass and insulation from a maximum *time lag* and minimum *decrement factor* point of view. The *time lag* is defined as the time it takes for a heat wave to propagate from the

outer to the inner surface and decreasing ratio of the amplitude of the heat wave during this process as the *decrement factor*.

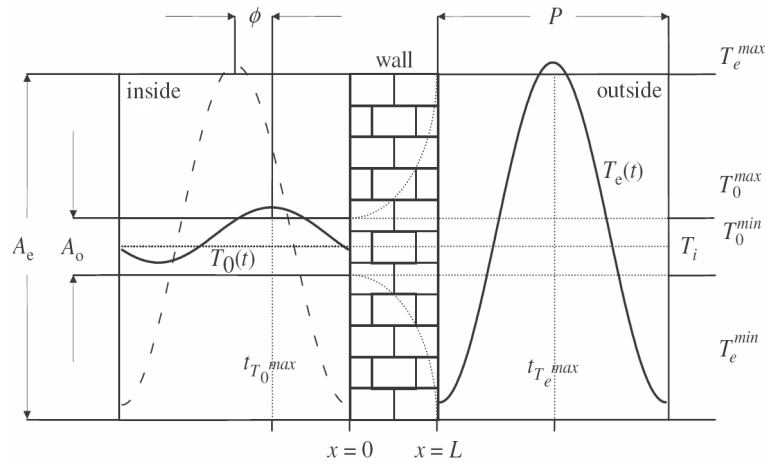


Figure 2-6 The scheme representation of time lag ϕ and decrement factor f , for a room without heat loads (Asan, 2006)

The study by Asan was done by solving the one-dimensional transient heat equation for different wall configuration with equivalent U-value. Two insulation boards were moved inside a massive wall, seeking the optimal values. Based on the results, Asan recommends that insulation should never be used as a whole in any location of the wall, except for on the outer surface. Further, by placing half of the insulation in the centre of the wall and half of the insulation at the outer surface gives high time lags and low decrement factors, and is close to the optimum value. Besides, according to Asan, this latter configuration is also practical and can easily be accomplished during construction.

However, it is not only insulation that may interfere with the performance of thermal mass. Any form of barrier between the internal air and the thermal mass will impede heat transfer thus reducing its effect. This is often called decoupling of thermal mass from the space. Surface finishes, such as wall paper, suspended ceilings, plasterboard, may decouple the thermal mass. Dry lining with an air gap on a block wall, will approximately halve the heat transfer (Orme, 2003). Thus, a building designed for being thermally heavy to control excessive heat gains may be no heavier than a wood frame construction due to careless interior design.

2.2.4 *Ventilation and HVAC-control strategies*

A thermally heavy building would require a greater output from the HVAC system to bring the thermal conditions to the desired level in the morning than a low-mass building. In theory, a building without mass would only require the time to heat or cool the air volume to the desired set-point in the morning, and would have lower overall cooling or heating loads than actual buildings (Braun, 1990). However, the key in utilizing the thermal mass is to prepare the building during the unoccupied hours, making, for instance, office buildings particularly suitable for such measures (Kolokotroni, 1999).

By combining night ventilation and a sufficient amount of thermal mass, the maximum temperature in the occupied hours can be reduced significantly and cooling energy can be saved (e.g. Braun, 2003; Kolokotroni, 1999; Shaviv, 2001; Pfafferott et al., 2003; Givoni, 1998a; Gerosa et al., 1999; Simmonds, 1991).

The efficiency of night ventilation is strongly related to three main parameters; 1) the difference between the indoor and outdoor temperature, 2) the air flow rate applied during the night period and 3) the thermal capacity of the building. The lower the outdoor temperature during night and the higher the outdoor air supply, the higher is the efficiency of night ventilation (Gerosa et al., 1999). The efficiency of night ventilation increases also with the amount of exposed thermal mass (Shaviv, 2001; Givoni, 1998a). The peak temperature reduction and energy savings are practically none for low mass buildings, but increases as the buildings get heavier. Additionally, the interior planning of the building plays a very important role, as it determines the airflow paths through the building (Gerosa et al., 1999).

Controlling the night cooling is of great importance. High ventilation rates combined with low external temperatures may under-cool the building structure and lead to an early morning heating demand to achieve comfort conditions (Orme, 2003; Kolokotroni, 1999).

Natural night ventilation is considered to be an energy efficient way to reduce the cooling demand in buildings. On the other hand, there are many obstacles that may prevent natural ventilation in buildings. The main barriers are safety, noise, air pollution, shading devices, draught, and occupant behaviour. If natural ventilation is not applicable, running the mechanical ventilation in free cooling mode at night can be a good alternative. In some

countries there are different utility rates at various times of the day and thermal mass can be utilized to delay the peak demand to low-rate periods. Reduced plant capacity may also be a significant investment benefit (Kolokotroni, 1999). There have been several studies on peak load-shifting, especially in the US, e.g. (Braun, 2003). The strategy is to pre-cool the building when the utility rates are low, and let the temperature float during high-rate periods, and turn on the air-conditioning when the rates become low again.

In the heating season, excessive heat stored during the day can be released during the night to lower the heating energy needed when the internal gains are small. Although a heavy structure cools more slowly than a light one, a heavy structure requires a larger output from the heating system than a light structure with the same level of insulation (Hitchin, 1979). The energy savings potential by night setback is also less for heavy buildings than light buildings. Burch et al. (1984a) studied the difference in night temperature setback savings for buildings with different amount of thermal mass. Their conclusion was that thermal mass decreased the setback energy savings, but only by 0.5-3 % on annual basis. If the night setback is large, the peak heating required to heat the house to the day set-point in the morning may be excessive. In heavy buildings, smaller setbacks and somewhat earlier reset of the thermostat should be used (Hestnes, 2003; Simmonds, 1991).

Weather forecast control is a promising technique to reduce the energy use of the building and to avoid indoor temperature fluctuations. The weather forecast control utilizes the building thermal mass based on knowledge of the forthcoming weather, including temperature, wind and solar radiation (Uppström et al., 2004). For instance, if it is known that a cold night will be followed by a warm and sunny day, heating can be reduced several hours in advance, without a drop in the indoor temperature. Likewise, if cold and windy weather is on the way, heating can be increased in advance to meet the demand.

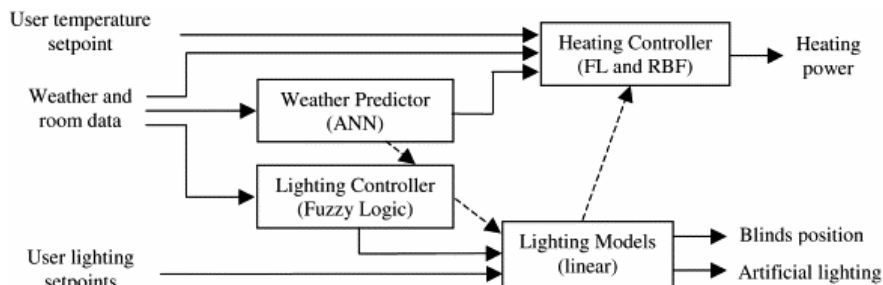


Figure 2-7 Example of a self-adaptive integrated building control system with weather predictor (Guillemin and Morel, 2002)

2.2.5 *Occupancy and internal gains*

As mentioned, office buildings are well suited for utilizing thermal mass. These and other buildings that are unoccupied during the night, have the possibility to use the night hours to cool the building structure in order to prepare the building for the next day. It is also advantageous to store excessive heat from the day to lower the heating energy needed during the night time. Hence, buildings with 24-hour occupancy, for instance hospitals, or buildings occupied during the night, for instance residential buildings, have less potential for saving energy by utilizing thermal mass (Braun, 1990; Diaz, 1994).

Tenorio (2002) used an ESP-r simulation model to assess the effect of combining air conditioning with passive means (so called dual-mode operation) for different room types, thermal mass amount and occupancy patterns. Dual-mode operation and thermal mass showed promising results in reducing cooling energy demand, but not in every case. The author points out that proper use of thermal mass was dependent on the user profile and room type.

A sensitivity study on the impact of different input parameters in simulations of a cross night ventilated office building concludes that internal heat gains have the largest important impact on thermal comfort. According to the study, internal gains are over 3 times more influential than the second-most influential parameter (Breesch and Janssens, 2005). A good estimate of the internal gains is therefore important.

Studies have shown that single office rooms may be unoccupied for long periods during the working day (Mathisen, 2007). The more scattered the occupation is, the greater is the potential to save energy by using demand control ventilation (DCV). However, a responsive control system also requires a responsive system to control. Heavy mass buildings have slow thermal response and intended savings might be considerably reduced if the system as a whole is not taken into consideration.

Building design strategies are tested during the design stage, often with the aid of computer modelling or simpler tools. However, the input parameters used in the calculations of assumed occupant use and behaviour for example, are often not correct or the design presumptions diverge considerably from reality. Although some strategies are dependent on

occupant control, designers employing these strategies assume that the technical installations and design strategies will be used as intended by the designers (Foster, 2000).

The design solutions are often not robust enough to cope with human “inventiveness”. Simple examples of such things are furnishing placed in front of displacement ventilation diffusers that obstructs the supply air or air paths crucial for a passive strategy to work as designed. More comprehensive examples are unconditioned atria, designed and modelled as unheated spaces, being used by the occupants as fully conditioned spaces.

“It is important that designers keep in mind that users represent the final determining factor on the effectiveness of any building system – including thermal mass.”

(Goulart, 2004)

3. THERMAL MASS IMPLICATIONS

In the previous chapters it is shown that utilization of thermal mass may have several advantages. However, there are some issues that must be considered before thermal mass is used as a building design measure. Some of them are discussed in the following sections.

3.1 Acoustics

Unfortunately, materials with great heat storage abilities are the worst possible surfaces for absorption of sound and hence may create some acoustical challenges. The acoustical property of a material is determined by its sound absorption coefficient (α), which is the fraction of incident sound which is not reflected (Smith et al., 1996). For instance, concrete has an absorption coefficient of about 0.02, while in comparison, some porous perforated materials commonly used in suspended ceilings may be as large as 0.80 dependent on the sound frequency (see Table A1.1).

The reverberation time (t_r), defined as the time it takes for a sound to decay by 60 dB, is a well known metric to assess the acoustical properties for a room. In Appendix A1.3 it is shown how the reverberation time can be calculated approximately. Table A1.1 gives the sound absorption coefficients for common building materials.

In Norway, guiding reverberation time limits for different buildings categories and room types are set by the Norwegian Standard NS 8175 (2004). The standard categorizes the acoustical qualities in classes A to D, where A is the strictest and D is the poorest standard. Class C is the minimum level to fulfil the building regulations (TEK, 2007), based on making 80 % of the occupants satisfied with the acoustical conditions. Table 3-1 lists reverberation time limits for offices. The time limits must be fulfilled for every octave band from 125 to 2000 Hz.

Table 3-1 Maximum reverberation times in seconds for office buildings (NS8175, 2004)

Room type	Class A	Class B	Class C	Class D
Offices, meeting rooms	0.6	0.6	0.8	0.9
Common areas, hallways	0.8	0.8	1.0	1.3
Stairways	1.0	1.0	1.3	1.3

Obviously, the acoustical challenges are in conflict with the desire to make a room thermally heavy, and may restrict the area of exposed thermal mass. However, if surfaces are acoustically unfortunate, acoustic tiles can be mounted without impeding the heat transfer significantly (Andersson et al., 1987). There are also other ways to improve the acoustics without decoupling the thermal mass from the indoor air. Hanging acoustic ceiling panels have proved to be very useful in some cases (Voss, 2000). However, some of these panels may collect dust, thus demand extra cleaning. Another drawback with this solution is that hanging panels may interfere with the ventilation airflow patterns.

In any case, conflicts with the acoustic situation require careful planning and balanced solutions.

3.2 Building adaptability

In the Norwegian literature, adaptability is often defined as a function of generality, elasticity and flexibility. The use of these terms has been somewhat inconsistent. However, Arge (2003) defines the terms as:

- *Generality*: The ability of a building to fulfil changing demands without changing its qualities, i.e. no structural or technical modifications is needed.
- *Elasticity*: The ability of a building to meet varying area demand, i.e. its possibility to split the floor in separate units or add new units to increase the floor area.
- *Flexibility*: The ability of a building to make changes within the main structure (e.g. change from single room offices to office landscape).

In the scope of this thesis, flexibility is the term which is most central. High flexibility is a question of: 1) time to make changes, 2) how much the changes interfere with the core activity in the building and 3) the need for specialists to carry out the changes.

During the last hundred years, major changes have taken place in office layouts and workplace design, and the changes are expected to accelerate (Blakstad, 2001). Trends and office solutions may change quickly, but an office building will last for 50 or even 100 years, and will contain different types of offices during its lifetime. An office building must also be able to meet rapid changes in organization structures which often require changing the floor plan (Blakstad, 2001).

A survey among 750 corporate managers in the Nordic countries reveals that every third manager plan to reorganize or rebuild the office structure within the next two years (Bjerrum, 2004). Clearly, use of heavy materials in partition walls may come in severe conflict with desire of having a flexible building.

3.3 LCC / LCA

Life cycle costing (LCC) is a technique which enables comparative cost assessments to be made over a specified period of time, taking into account all relevant economic factors, i.e. initial capital costs and future operational costs (ISO 15686-1, 2000).

Despite an increasing interest in the LCC approach, the adoption and application of LCC in the building sector remains limited. A survey made among clients in the building sector in Sweden (Sternier, 2000), shows that more than 2/3 indicated that they use a life-cycle perspective when making decisions. However, this does not necessarily mean that they use LCC calculations. The study also reveals that if LCC calculations are used, they are primarily related to the technical installations, such as the HVAC system, and not the building project as a whole.

A holistic objective analysis must be carried out in order to determine the impact a particular building has on the environment. Life Cycle Assessment (LCA) is, in terms of building construction, an assessment of the total environmental impact associated the use, disposal and with all actions in relation to the construction of buildings. LCA, in contrast to LCC, does not address economic or societal aspects. Cole (1996) states that LCA has been accepted in the environmental community as the only legitimate method to compare alternative materials, components, and services. Nevertheless, LCA has not been particularly successful in practice in the construction sector, because of problems concerning the availability of input data and the complexity of the LCA analysis itself (Sternier, 2002).

Life-cycle energy is a commonly used assessment indicator, and includes all energy incurred in the production, use and removal of a building. Cole and Kernan (1999) define four categories of the energy use of a building during its life-cycle:

- Energy to initially produce the building (initial embodied energy)
- The recurring embodied energy for refurbishment and maintenance over the lifetime of a building
- Energy to operate the building, i.e. the total energy required for heating, cooling, lighting and equipment
- Energy to demolish the building and dispose of the building materials

Some studies have compared light and heavy mass buildings in the life-cycle energy perspective. Ståhl (2002) compared heavy and light building constructions for a residential building, a school building and an office building. The results from the study showed that the light structures demanded least energy during the phases of production and demolition in comparison to the equivalent heavy buildings. The heavy buildings used least energy during the operational phase, and in sum there was in fact no significant difference between the two cases in life-cycle energy use.

A similar study was done for residential houses in New Zealand (Mithraratne, 2004), where a light, a heavy and a super insulated house were compared in both an LCA and LCC perspective. The conclusion from this study was that the super insulated house had substantially less negative impact on the environment than the other two alternatives. The light and heavy structure demanded 45 % and 58 % more energy respectively during the life-cycle. However, the additional insulation was not found to be cost beneficial, i.e. the initial cost of construction connected to the additional insulation increased and remained higher throughout the useful life time of the building.

A study in Denmark compared three different layouts of typical lightweight timber-based and heavyweight concrete-based low-energy residential buildings. Even though the heavyweight houses performed best during the operational phase, they did not show a significant environmental advantage over the lightweight houses in terms of total CO₂ emissions, even when solar strategies were used.

The energy used to operate buildings is by far the largest component of life-cycle energy use, about 80-90 % (Ståhl, 2002; Cole, 1999; Mithraratne, 2004). However, as buildings become more energy efficient, the amount of energy to produce the building (the embodied energy) will represent an increasing component of the life-cycle energy. As operating energy is reduced and halved compared to current standards, embodied energy will be a dominant factor. However, for the present, design strategies to significantly reduce the operating energy should be emphasized. When the operating energy has been reduced, more should be ascribed to reducing the embodied energy.

Reducing the embodied energy is more a question of comprehensive design, than the embodied energy in the materials themselves. Since the recurring embodied energy associated with the replacement and repair of building materials is significant (in order of

the initial embodied energy over the lifetime (Cole, 1999)), it is important to focus on the longevity of the materials and their ability to replace elements within a total building assembly. Thus, making buildings adaptable is also a question of making buildings sustainable.

4. PERFORMANCE OF THERMAL MASS

This chapter presents an overview of some experimental and analytical work that evaluates the effectiveness of thermal mass for different building types in various climates. Table 4-2 summarizes 24 studies on the subject. The table includes studies on reducing cooling and/or heating demand, and studies on thermal comfort. The table presents the scope and results together with annotations of climate and building type used in the studies, and also whether simulations (sim) or measurements (meas) have been carried out.

The simulation studies include a wide range of simulation programs, and those mentioned are listed in Table 4-1. For further information, the capabilities of a number of simulation programs are compared in the comprehensive study by Crawly et al. (2005).

Table 4-1 Simulation programs used for thermal mass assessment

Program	Origin	Users*	Website
APACHE	UK	?	http://www.iesve.com
BSim2000	DEN	~125	http://www.bsim.dk/
DEROB-LTH	SWE	~150	http://www.derob.se/
DOE-2.1E	USA	>1000	http://simulationresearch.lbl.gov/
ENERGY10	USA	>3200	http://www.sbicouncil.org/
ESP-r	UK	?	http://www.esru.strath.ac.uk/
HTB2	UK	?	http://www.cardiff.ac.uk/archi/school
Suncode (Sunrel)	US	>100	http://www.ecotope.com/
TAS	UK	~250	http://ourworld.compuserve.com/homepages/edsl
TRNSYS	US	>500	http://sel.me.wisc.edu/trnsys
TSBI3	DEN	~200	http://vbn.aau.dk/research/tsbi3(10050074)/

* Approximate estimation by the US DoE (2007)

Table 4-2 Summary of studies done on thermal mass

Author(s)	Building	Climate	Method	Scope	Results
Bellamy and Mackenzie (2003)	Single zone test buildings	Moderate (NZL)	Meas. and sim.	A timber and a concrete test building were monitored for a year. Simulations with BSim2000 and Suncode were compared against the measurements.	The concrete building used 7.5 kWh/m ² (15.7 %) less heating energy than the timber building. The simulated savings were 16.2 % and 23.2 % for BSim2000 and Suncode, respectively.
Birtles et al. (1996)	Office	Moderate (ENG)	Sim.	A bioclimatic chart was used to estimate the cooling potential of thermal mass and night ventilation. APACHE simulation model was used to assess the estimates.	The bioclimatic analyses indicate that thermal mass and night ventilation should be sufficient to cover cooling demand in typical UK offices. This was confirmed by the simulations in most cases.
Blondeau et al. (1997)	Classroom	La Rochelle (FRA)	Meas. and sim.	An experimental field study and TRNSYS simulations were carried out to assess the comfort advantages and cooling potential of night ventilation.	Night ventilation decreased indoor air temperatures by 2 K. The simulations indicated that cooling energy reductions were 12 %, 25 % and 54 % for set-points at 22°C, 24°C and 26°C, respectively.
Bojic (2005)	Residential	Hot humid (CHN)	Sim.	A HTB2 simulation model was used to predict the annual cooling demand for twelve alternative wall constructions.	The simulations indicated that if the thermal capacity of the walls was reduced, this would lead up to a 60 % increase of the cooling energy demand.
Braun (1990)	Office	Not specified	Sim.	Investigation of building thermal mass and dynamic building control to reduce the peak electricity demand and to offset peak cooling loads.	Cooling energy cost and peak electricity demand was significantly reduced through optimal control of the building thermal mass. Cost savings were most significant in the presence of time-of-day rates.
Burch et al. (1984b)	Single zone test buildings	Moderate (MD, US)	Meas.	Six similar one-room test buildings with equivalent U-value were monitored. The effect of wall mass on the cooling energy requirements was studied.	Heavy buildings observed to have significant lower cooling energy use. Exposed mass facing the inside was most effective. Night ventilation and thermal mass reduced the cooling energy use up to 36 %.

1) This study is also referred to in chapter 2.1.3

2) This study is also referred to in chapter 2.2.4

Table 4-2 continued...

Author(s)	Building	Climate	Method	Scope	Results
Burch et al. (1984c)	Single zone test buildings	Moderate (MD, US)	Meas.	Six similar one-room buildings with equivalent U-value monitored. The effect of wall mass on the heating energy demand and comfort studied.	Thermal mass observed to have little effect on the heating energy use during the cold periods. In the intermediate seasons, the heavy buildings used significantly less heating energy.
Capeluto et al. (2004)	Residential	Hot humid (ISR)	Sim.	ENERGY simulation model was used to find necessary amount of thermal mass and sufficient night ventilation flow rate to avoid air-conditioning	Thermal mass combined with night ventilation was ³⁾ found to be sufficient in order to achieve thermal comfort for residential buildings, provided that the diurnal temperature swing was greater than 5 K.
Florides et al. (2002)	Residential	Hot humid (CYP)	Sim.	TRNSYS simulation model was used to evaluate low energy measures and their respective cost effectiveness.	Thermal mass and night ventilation lowers the maximum temperature, but not sufficiently to get within acceptable limits during the summer time.
Givoni (1998a)	Residential	Hot arid (CA, US)	Meas.	Three buildings with equivalent heat loss coefficient but different mass levels were monitored under various ventilation and shading conditions.	Night ventilation showed only little effect in the ⁴⁾ low-mass building, but was very effective in lowering the maximum indoor air temperature in the high-mass buildings.
Gratia and De Herde (2003)	Office	Moderate (BEL)	Sim.	TAS simulation model was used in a parametric study of an office building, where thermal mass is one energy reducing measure.	The simulations showed that thermal mass combined with night ventilation reduced overheating, and if the internal heat gains were not too excessive, mechanical cooling could be omitted.
Kolokotroni et al. (1998)	Office	Moderate (ENG)	Meas. and sim.	Assessment of summer night cooling suitability by using bioclimatic charts, measurements and APACHE simulations.	Night cooling was shown to be a viable method for addressing the issue of summer overheating in typical UK office buildings.

³⁾ The study is also referred to in chapter 2.2.1

⁴⁾ This study is also referred to in chapter 2.2.4

Table 4-2 continued...

Author(s)	Building	Climate	Method	Scope	Results
Kolokotroni and Aronis (1999)	Office	Moderate (ENG)	Meas.	A 3TC simulation model was used to estimate the energy saving potential by night ventilation in an A/C office.	Natural night ventilation was found to be beneficial. 5) Energy savings increased from 5 % to 12 % by increasing the area of exposed thermal mass.
Kosny et al. (2001)	Residential	Ten climates in the US	Sim.	A DOE-2.1E whole-building simulation model was used in analyses of sixteen different wall constructions. Historic and current field studies were also presented.	Walls with thermal mass in good contact with the indoor air were found to be most effective. Dependent on the climate, energy use was reduced 6-18 % by switching from light to heavy walls.
Kossecka and Kosny (2002)	Residential	Six climates in the US	Sim.	DOE-2.1E whole-building simulation model was used for energy analyses of different wall configurations.	Insulation on the outside and exposed mass facing 6) inside showed best energy performance. Depending on the climate, the total energy savings were 2-11 %.
La Roche and Milne (2004)	Single zone test buildings (CA, US)	Hot humid (CA, US)	Meas.	A test cell with varying mass amount and a smart ventilation controller compared to a cell with fixed ventilation rate.	The study confirms that more mass performed better than less mass and that higher volumes of controlled ventilation outperformed fixed ventilation rates.
Lerum and Lathey (2002)	School	Five climates (AZ, US)	Sim.	Energy-10 building simulation model was used to study the temperature and cooling energy reduction by utilizing thermal mass and night ventilation.	High mass and night ventilation reduced the peak temperature and could, depending on the climate, replace mechanical cooling for 3-5 months. Hence, cooling energy demand was significantly reduced.
Norén et al. (1999)	Residential	Cold (SWE)	Sim.	Three buildings with different mass levels were simulated in TSB13, EN832 and BRIS, during the heating season.	All three simulation tools showed the same tendency: 7) The heaviest building required 82-86 % of the heating energy demand of the lightweight building.

5) This study is also referred to in chapter 2.2.4

6) This study is also referred to in chapter 2.2.3

7) This study is also referred to in chapter 2.2.2

Table 4-2 continued...

Author(s)	Building	Climate	Method	Scope	Results
Ogoli (2002)	Single zone test buildings	Hot arid (KEN)	Meas.	Four test chambers with different thermal mass amount and location were monitored during a warm period.	From the measurements, a formula for predicting maximum indoor air temperature for closed high mass buildings was derived.
Ruud et al. (1990)	Office	Hot humid (FL, US)	Meas.	Two stories in an office building were monitored and compared for cooling energy use; one storey was pre-cooled and one free floated, during the night hours.	The daytime cooling load was reduced by 18 % in the pre-cooled storey, but there was no reduction in peak demand. Weekend pre-cooling was not effective in reducing the week time cooling loads.
Shaviv et al. (2001)	Residential	Hot humid (ISR)	Sim.	An ENERGY simulation model was used to calculate the influence of thermal mass and night ventilation on the maximum indoor temperature in summer.	A peak reduction of 3-6 K in heavy buildings 8) achieved. A simple design tool to predict the indoor temperature reduction given mass amount, night ventilation rate and temperature swing was derived.
Simmonds (1991)	Office	Moderate (NLD)	Sim.	Computer simulations were used to find optimal building design for both summer and winter conditions by utilizing building thermal mass.	Night ventilation improved thermal comfort significantly and a cooling plant could be omitted. Night setback found to be unsuitable in cold periods due to an increased demand for heating capacity.
Ståhl (2002)	Office, school and residential	Cold / moderate (SWE)	Sim.	DEROB-LTH simulation models of light and heavy versions of three different building types were compared for energy demand over life cycle of the building.	The heavy buildings demanded less energy during the operational phase than the light buildings. However, over the life cycle there was no significant difference in total energy demand.
Tenorio (2002)	Residential	Hot humid (AUS)	Sim.	ESP-r simulation model was used to assess A/C and passive means ran in dual-mode for different room types, thermal mass amount and occupancy patterns.	Dual-mode operation and thermal mass showed 10) promising results in reducing cooling energy demand, but not in any case. Proper use of mass was very dependent on the user profile and room type.

8) This study is also referred to in chapter 2.2.1, 2.2.2 and 2.2.4

9) This study is also referred to in chapter 3.3

10) This study is also referred to in chapter 2.2.5

4.1 Closing remarks

The general conclusion that can be derived from the literature review is that thermal mass together with night ventilation reduce the indoor maximum temperature and the cooling energy demand, and also offset the peak cooling demand. All studies, both experimental and analytical, support this. Thermal mass combined with night ventilation may reduce the maximum indoor temperature by 2-6 K (e.g. Givoni, 1998a; Shaviv et al., 2001), provided that the diurnal outdoor temperature swing is adequate (at least 5-10 K). Depending on the climate and building type, the cooling energy savings found in the literature span from 5 % to 36 % (e.g. Burch et al., 1984b; Ruud et al., 1990). Moreover, some studies (e.g. Gratia and De Herde, 2003; Kolokotroni et al., 1998) conclude that if the heat gains are not too excessive in office buildings, thermal mass and night ventilation should be sufficient to cover the cooling demand alone in moderate climates.

Studies also conclude that thermally heavy buildings have lower space heating energy demand than light buildings. Heavy residential buildings demand about 15 % less heating energy compared to equivalent light buildings (e.g. Bellamy and Mackenzie, 2003; Norén et al., 1999), and about 20 % less for offices (e.g. Ståhl, 2002). Heating energy savings are most significant in the intermediate seasons in cold climates and in climates where the balance temperature of a building is close to the mean outdoor temperature (e.g. Burch et al., 1984c).

PART II:

PARAMETRIC STUDY

ASSESSING THERMAL MASS

As discussed in Part I, various parameters affect the efficiency of thermal mass. The aim of this part is to assess the efficiency of thermal mass in a Norwegian context. A simulation model is employed to evaluate the efficiency of thermal mass by varying several parameters such as climate, occupancy profiles, internal heat loads, control ranges and convection correlations. All tested parameters are given separate sections and each is closed with a brief discussion. The most important findings from the parametric study are summarized in a separate section at the end.

5. THERMAL MASS ASSESSMENT

The main objective of this part is to assess the efficiency and potential benefit of utilizing thermal mass for a single office room which has high standards for insulation and air tightness, and fulfil the new requirements in the revised building code in Norway (TEK, 2007). The goal of this part is to answer following questions:

- Is it possible to avoid mechanical space cooling?
- Will thermal mass improve the thermal environment?
- Will thermal mass decrease heating energy demand?

The following sections will investigate the thermal mass influences on the net energy demand and the thermal environment, and study whether the influence of thermal mass depends on;

- climate,
- occupancy profiles,
- internal heat loads,
- set-point for cooling and control range, and
- heat transfer correlations

5.1 Choice of simulation program

As discussed in Part I, several approximations commonly made in simulation programs are not appropriate if accurate assessment of the influence of thermal mass is required. Capabilities, such as taking into account the placement of thermal mass (geometry), advanced treatment of solar gains (ray-trace), options to choose convenient convection correlations and possibility to invoke advanced building control can be advantageous when analysing different aspects of thermal mass utilization. Today, only a handful of simulation programs possess all these capabilities, and ESP-r was the one that finally was chosen for the study.

5.1.1 ESP-r

ESP-r attempts to simulate the real world as rigorously as possible and to a level which is consistent with current best practice. By addressing all aspects simultaneously, ESP-r allows the designer to explore the complex relationships between the form, fabric, air flow, plant and control of a building (ESRU, 2007).

ESP-r is based on a finite volume, conservation approach in which a problem (specified in terms of geometry, construction, operation, leakage distribution, etc.) is transformed into a set of conservation equations (for energy, mass, momentum, etc.) which are then integrated at successive time-steps in response to climate, occupant and control system influences (ESRU 2007).

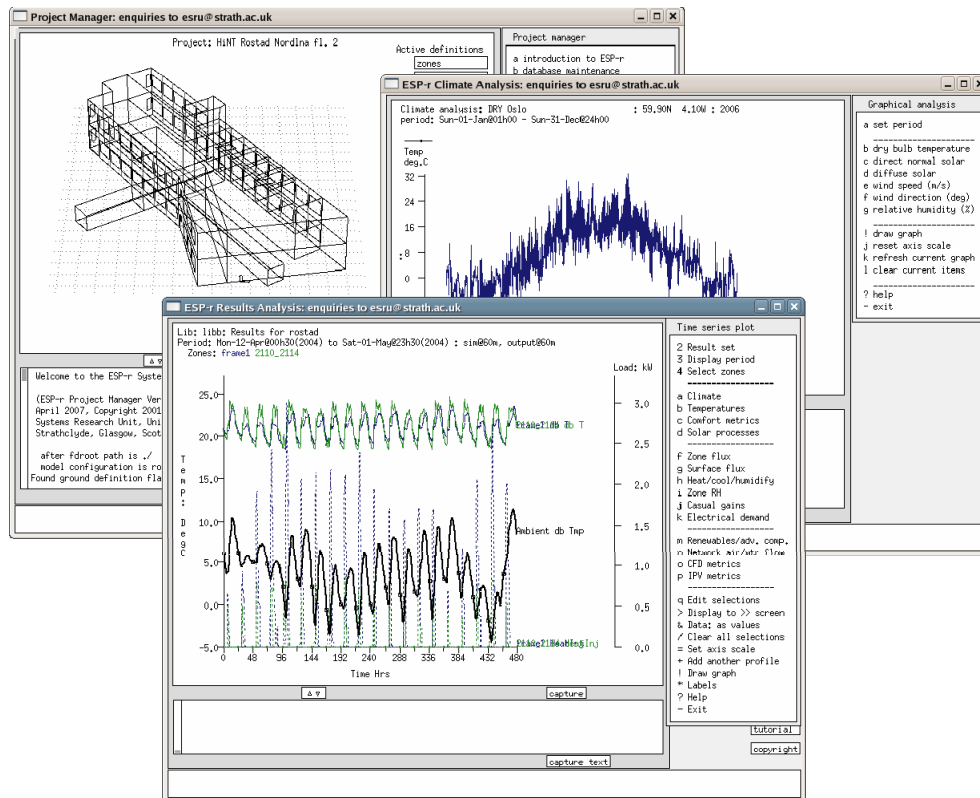


Figure 5-1 Screenshot from ESP-r System version 11.3 showing the Project Manager (back), the Climate Analysis module and the Result Analysis module (front)

ESP-r has been under development for more than 25 years, and has been undergoing numerous validation tests. A summary of all validation tests can be found in Strachan (2000).

Support modules included in ESP-r are among others:

- Climate display and analysis
- Integrated simulation engine
- Environmental impacts assessment
- 2D-3D conduction grid definition
- Shading/insolation calculation
- View factor calculations
- Convection calculations
- Detailed airflow analysis (CFD)

ESP-r was also chosen for its responsive simulation community through a mailing list, and due to its free distribution under the GPL license through the ESRU website. The website also includes an extensive publications list, example models, cross-referenced source code, tutorials and resources for developers.

However, ESP-r also has its weaknesses. Specialist features require knowledge of the particular subject. Although robust and increasingly used for consulting, ESP-r retains much of the look and feel of a research tool and lacks the extensive databases associated with commercial tools. The current Windows implementation does not conform to the standard look and feel of most Windows applications and lacks a few features available on other platforms. It is estimated that to obtain the level of expertise, 1-2 years of frequent use is required (Haugaard, 2003).

5.2 Simulation model

To study the potential benefits of thermal mass utilization, three single office rooms with different amounts of thermal mass are modelled in the building simulation program ESP-r. All versions of the office room have the same U-values and fulfil the requirements according to the so-called *energy measure model* defined in the revised building code (TEK, 2007) given in Table 5-1. The models should give answers to indoor thermal climate and power demand issues, but also, according to Høseggen (2005), a single office room model is applicable to estimate the heating energy demand in a building dominated by cell offices. However, as pointed out in the paper, such an approach tends to overestimate the ventilation heating energy demand. In any case, in the following comparative studies, the relative differences are of greater interest than the absolute level of energy demands.

Table 5-1 The new building regulations for commercial buildings compared to the old regulations (TEK, 1997; TEK, 2007)

	TEK '97	TEK '07
Glass and door area ^a	20 %	20 %
U-value external wall	0.22 W/(m ² ·K)	0.18 W/(m ² ·K)
U-value roof	0.15 W/(m ² ·K)	0.13 W/(m ² ·K)
U-value floor on ground	0.15 W/(m ² ·K)	0.15 W/(m ² ·K)
U-value windows and doors (frame included)	1.60 W/(m ² ·K)	1.20 W/(m ² ·K)
U-value glazed walls and roofs	2.00 W/(m ² ·K)	same as for windows
Air tightness at 50 Pa	3.0/1.5 ^b ach	1.5 ach
Heat recovery ^c	60 %	70 %
Specific fan power (SFP)	none	2.0/1.0 ^d kW/(m ³ /s)
Local space cooling	minimized	should be avoided ^e
Temperature control	none	night set-back to 19 °C

^a maximum percentage of the available area of the building

^b buildings up to two floors / buildings with more than two floors

^c annual mean temperature efficiency

^d SFP day/night

^e automatic sun shading devices or other measures should be used to fulfil the thermal comfort requirements without using local cooling equipment

The three rooms, which are 2.4 m wide and 4.2 m deep, with a room height of 2.75 m, have the following construction properties with regard to thermal mass;

- *Light*: External framework wall, gypsum internal partition walls, suspended ceiling, and suspended floor with parquet.
- *Medium*: External framework wall, gypsum internal partition walls, exposed concrete ceiling, and concrete floor with linoleum finish.
- *Heavy*: External concrete wall, concrete partition walls, exposed concrete ceiling, and concrete floor with linoleum finish.

For comparison, Table 5-2 shows the thermal heat capacities and the thermal effusivities (as defined in Section 2.1.1) per floor area for the office rooms. The calculations of the parameters are shown in Appendix A2.1.

Table 5-2 Heat capacities and thermal effusivities per floor area of the office rooms

Room	Heat capacity (kJ/(m ² ·K))	Thermal effusivity (W·s ^{1/2} /(m ² ·K))
<i>Light</i>	1895	1522
<i>Medium</i>	4880	4001
<i>Heavy</i>	10884	5688

Examining the numbers from Table 5-2, it can be observed that the heat capacities for the rooms are coarsely in order of 2:5:11 for the *light*, *medium* and *heavy* rooms, respectively, while the thermal effusivities are in order of 3:8:11. As shown later, the thermal effusivity is the parameter that best describes the thermal mass properties.

The façade of the office rooms is facing south-west in order to evaluate the extreme summer conditions. Appendix A2.4 shows the evaluation of different façade orientations and why south-west is considered to be the most exposed orientation.

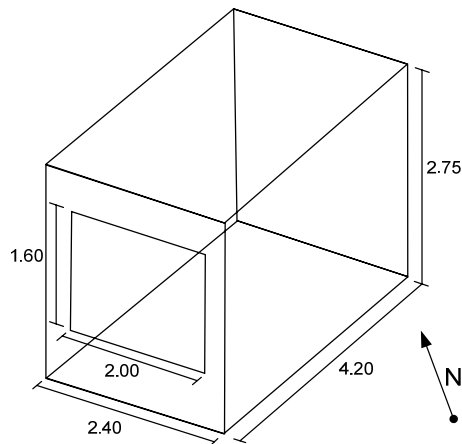


Figure 5-2 The office room model with measures in metres

The offices are equipped with presence detectors, temperature sensors and controller units that control heating, ventilation and lighting.

Internal heat gains from lighting, equipment, and people will vary dependent on the room occupancy. In the basic model, recorded occupancy data from about thirty office rooms in *Statens Hus* (Mathisen, 2007), a typical modern office building in Trondheim is used. In Figure 5-3, the mean recorded occupancy is shown together with the hourly averaged occupancy.

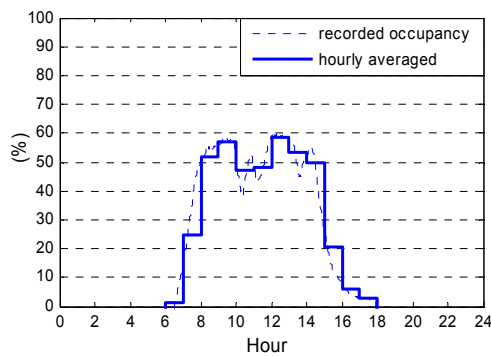


Figure 5-3 Mean occupancy from *Statens Hus*. The hourly averaged curve represents the occupancy of the office rooms

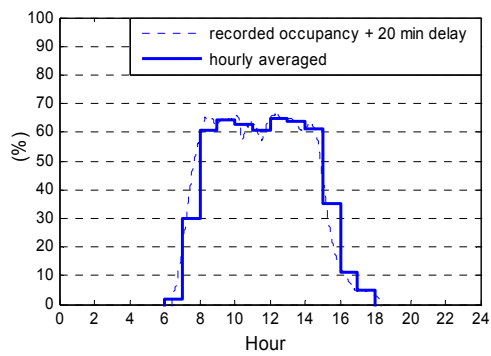


Figure 5-4 Mean occupancy added 20 min delay. This curve controls the ventilation and lighting in the office rooms

Normally, to avoid the ventilation and lighting to turn on and off frequently, a time delay between registered room absence and control action is implemented. In this case, a 20 minute time delay is set for the lighting and ventilation controls. The curve which controls the lighting and ventilation is shown in Figure 5-4. The internal heat gains and air flows rates are listed in Table 5-3.

Table 5-3 Heat gains and air flows for the office rooms used in the basic model (ASHRAE 2005, REN 2007)

	Amount	Time of operation
People	75 W sensible + 45 W latent	6-18 working days *
Lighting	3x35 W	6-18 working days **
Equipment	12 W/m ²	6-18 working days *
Ventilation ***	2.0 l/s per m ² (high) or 0.7 l/s per m ² (low) + 7.0 l/s per person	6-18 working days 6-18 working days **
Infiltration	0.1 h ⁻¹	0-24 all days

* Occupancy dependent. The value is multiplied by the hourly averaged room occupancy (Figure 5-3)

** Occupancy dependent. The value is multiplied by the hourly averaged room occupancy with time delay (Figure 5-4)

*** 2 l/s is the common design flow rate in Norway. 0.7 l/s is the minimum air flow rate provided well known and documented low emitting materials (REN, 2007).

When the room temperature reaches the set-point for cooling (23 °C), the ventilation air flow rate increases proportionally with the temperature and reaches full air flow rate at 25 °C, regardless of whether the room is occupied or not. The office rooms are each equipped with a 500 W radiator, which have a set-point for heating at 21 °C all year around and a night temperature set-back of 2 K outside working hours and weekends. The radiators are placed under the windows, thus affecting the convection regime when they are turned on (see Appendix A2.7).

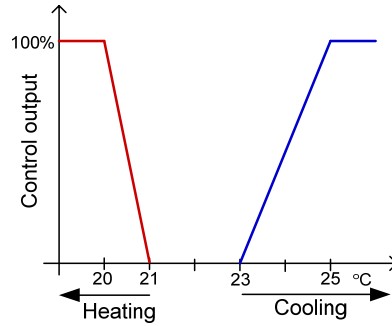


Figure 5-5 Heating and comfort ventilation control strategy

The ventilation air flow rate is dependent on time, occupancy and temperature, thus the total ventilation rate, q_{vent} , is given by

$$q_{vent}(t, \theta) = q_{basic}(t) + O(t) \cdot q_{pres} + q_{temp}(\theta) \quad (\text{m}^3/\text{s}) \quad (5.1)$$

where

q_{vent} = total ventilation rate

q_{basic} = basic ventilation from 6 am to 6 pm

O = occupancy according to Figure 5-4

q_{pres} = occupancy dependent ventilation rate

q_{temp} = temperature dependent ventilation rate according to Figure 5-5

t = time

θ = temperature

The rooms are equipped with external Venetian blinds, which are controlled by the incident solar radiation (closed when $I > 200 \text{ W/m}^2$) during working hours and by the indoor temperature (closed when $\theta > 21 \text{ }^\circ\text{C}$) outside working hours to benefit from the solar gains when there is a heating demand.

5.3 Climate data

Accurate estimation of building energy performance requires knowledge of the weather to which the building is subject to. Modern design methods by computer simulations require one year of proper hourly meteorological data. These data files normally contain dry bulb temperature, solar radiation, humidity, wind velocity, and wind direction (Forejt et al., 2005). The most important representations of yearly hourly weather data are Test Reference Year (TRY), Typical Meteorological Year (TMY), Design Reference Year (DRY), and Weather Year for Energy Calculations (WYEC).

- The “European” TRY is known as the TMY in the US. Unlike the “US” TRY, which is a selected historical year closest to the average (typically a 30-year period), the “European” TRY may contain months from a number of different years. The months selected are the ones that deviate the least from a) the mean b) the frequency distribution of the individual parameters, and c) the implied correlation between variables within each month of the long-term data set. Where months from different years are linked together, the parameters are “smoothed” over 5 hours to avoid sudden jumps (Lund, 1985). The TMY has been revised and improved solar models and extended solar parameters have been added.
- DRY is a further attempt to modify the TRY to be even more like the yearly average months by adjusting the selected months. The parameters such as dry bulb temperature, solar radiation, and humidity (but not wind) are adjusted by replacing certain days with days from other years. In addition, some new parameters are added to the data sets, and also 5 minute values for direct normal radiation and forecast information to be used with advanced building energy management systems (Skartveit et al., 1994).
- WYEC is constructed by determining for each month of the year, the single, real month of hourly data whose mean dry bulb temperature is closest to the average dry-bulb temperature for that month during the 30-year period of record. Then the rest of the month is constructed by substituting days from the months in the other years to bring the mean for the prevailing month closer to the 30-year average. Like the TMY, the WYEC has been revised, called WYEC2 (Stoffel, 1998).

An extensive study by Crawley (1998) compares simulation results using different reference years (“US” TRY, TMY, TMY2, WYEC, WYEC2) to the results based on actual

hourly weather data for the 30-year period from 1961 to 1990. One of the conclusions from the study is that users of energy simulation programs should avoid using single years (i.e. “US” TRY-type weather data), because no single year can represent the typical weather patterns. The TMY2 and the WYEC2 will provide the energy simulation results that most closely represent typical weather patterns. Crawley also recommends that future weather sets should include a cold/cloudy and a hot/sunny year to capture more than the average conditions and provide simulation results that identify the uncertainty variability inherent in weather.

At this moment, only DRYs for Oslo (59.9° N), Bergen (60.2° N) and Andøya (69.3° N) are available for Norway. Although Norway is a country where many parts are north of the Arctic Circle, the Gulf Stream makes the climate in Norway relatively coastal and mild, and has a climate more similar to the British Isles and Northern Europe than other countries at these latitudes. Table 5-4 shows the monthly maximum, minimum, and mean temperatures for the DRY climate files.



Figure 5-6 Map of Norway (modified from SSB, 2006)

Table 5-4 Minimum, maximum and mean temperatures (°C) for the DRY files

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual
Oslo	Min	-22.0	-24.7	-17.7	-7.6	-1.0	3.5	8.0	5.2	-1.2	-6.8	-14.7	-20.9	-24.7
	Max	10.7	10.2	14.1	19.0	26.4	30.8	29.8	32.6	24.2	19.6	12.9	11.2	32.6
	Mean	-3.7	-4.8	-0.5	4.8	11.7	16.5	17.5	16.9	11.5	6.4	0.5	-2.5	6.3
Bergen	Min	-14.7	-12.0	-8.6	-4.3	-1.7	3.8	5.5	4.0	1.3	-4.3	-9.6	-13.2	-14.7
	Max	10.1	10.1	13.8	16.7	23.3	26.6	27.1	27.3	23.4	18.3	12.1	9.7	27.3
	Mean	1.0	0.5	2.6	4.7	9.4	12.1	13.6	13.2	10.5	7.3	3.9	1.1	6.7
Andøya	Min	-18.8	-16.7	-18.5	-12.7	-8.5	0.4	2.7	0.8	-2.4	-5.8	-11.5	-14.5	-18.8
	Max	7.2	7.0	6.9	11.7	19.6	22.7	22.1	22.6	18.1	13.7	8.9	8.1	22.7
	Mean	-2.0	-1.6	-1.2	1.2	5.1	8.1	10.7	10.7	7.6	4.6	0.8	-1.3	3.6

In the DRY files, the humidity is expressed as dew point temperatures. Appendix A2.3 shows how the relative humidity (RH), which is the humidity input parameter in ESP-r, can be found from the dry bulb temperature, pressure and dew point temperature.

5.4 Climatic considerations

In this section, the influence of the climate on thermal mass will be evaluated. At first, a degree-hour approach will be used as a coarse evaluation of the cooling potential. Further, the cooling potential and potential heating energy savings related to thermal mass will be evaluated by dynamic simulations.

5.4.1 Climatic cooling potential

In Section 2.2.1, the influence of climate on thermal mass efficiency is discussed. In this section the degree-hour approach by Artmann et al. (2007) is employed for different Norwegian climates.

The climatic cooling potential (*CCP*) is defined as the summation of products between building and external air temperature difference. Thus,

$$CCP = \frac{1}{N} \sum_{n=1}^N \sum_{h=h_s}^{h_f} m_{n,h} (\theta_{b,n,h} - \theta_{e,n,h}) \text{ (Kh)} \quad \begin{cases} m = 1 \text{ h if } \theta_b - \theta_e \geq \Delta\theta_{crit} \\ m = 0 \text{ h if } \theta_b - \theta_e < \Delta\theta_{crit} \end{cases} \quad (5.2)$$

where

- N = nights in period
- θ = temperature
- $\Delta\theta_{crit}$ = threshold value of the temperature difference

with subscripts

- f = finishing hour of night ventilation, (6 am)
- s = starting hour for night ventilation (6 pm)
- n = night (date)
- h = hour {0, ..., 24 h}
- b = building
- e = external

The building temperature is assumed to oscillate harmonically, thus

$$\theta_{b,h} = 24 + 2 \cos\left(2\pi \frac{h-18}{24}\right) \text{ (}^\circ\text{C)} \quad (5.3)$$

Applying this definition, the indoor temperature reaches the maximum of 26 °C at 6 pm and the minimum of 22 °C at 6 am. During the day, this definition makes the air temperature vary within an interval of 4 K and to stay below 26 °C.

A certain temperature difference between the indoor air temperature and the ambient temperature is needed for effective night purge between the room air and the room surfaces. Hence, it is assumed that night ventilation is only applicable when the difference is greater than $\Delta\theta_{crit} = 3 \text{ K}$.

Figure 5-7 shows the monthly mean climatic cooling potential per night using the DRY files for Oslo, Bergen, and Andøya. The CCP for Trondheim is also included, using data produced by Meteonorm (Meteotest, 2003).

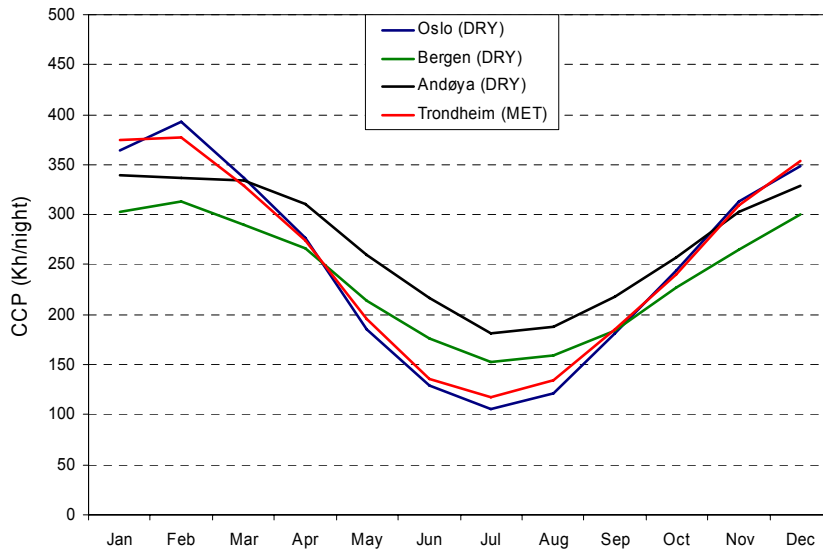


Figure 5-7 Monthly mean climatic cooling potential per night

A rough calculation for an office room is shown in Appendix A2.2. For the assumptions done in the example, the heat flux that may be absorbed per degree hour of the cooling potential is 0.34 W/m^2 per Kh. Assuming that the total mean internal heat load and the solar gains are 34 W/m^2 during an eight hour working day, a *CCP* of 100 Kh per night is needed to discharge the heat stored in the room.

However, individual climatic cooling potential per night can fall far below the monthly mean values. Therefore, in Figure 5-8, the nightly *CCPs* are presented as cumulative frequencies. For example, the *CCP* for Bergen is below 100 Kh only 5 nights during the year, and for 90 % of the nights the *CCP* is above 130 Kh.

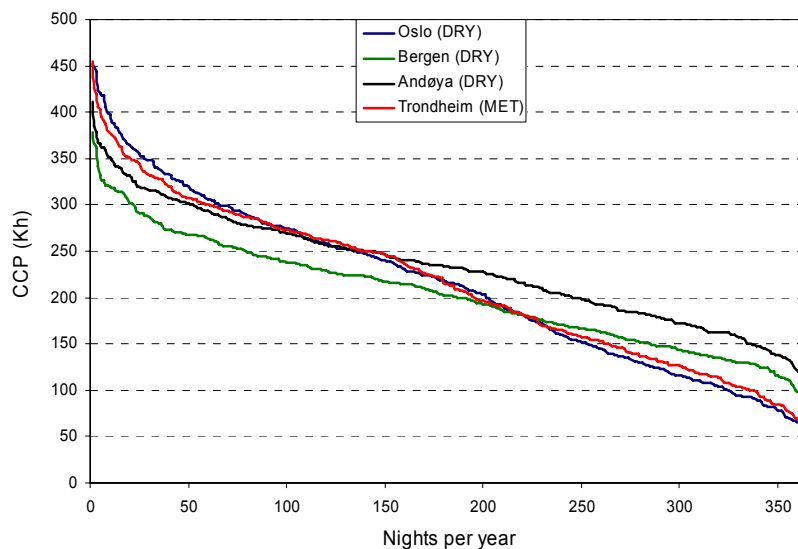


Figure 5-8 Cumulative frequency distribution of climatic cooling potential

The simple degree-hour approach presented can give valuable information in the design phase of a building given the local climate. However, the method is not a substitution for building simulations where building specific parameters, time dependent internal and solar heat loads, thermal mass and airflow patterns are taken into account.

5.4.2 Dynamic performance

The previous section indicates that climates similar to the Norwegian one have a great potential for passive cooling during night time. In this section, hourly climate conditions and building parameters will be taken into consideration. The model described in Section 5.2 is used employing the DRY climate files. Daytime, the rooms are supplied with conditioned air at 17 °C. If necessary, the rooms are also free cooled according to Figure 5-5 outside working hours. Simulations are done for the cooling season from May 1st to September 30th, for both high (10 m³/h m²) and low (5 m³/h m²) ventilation air flow rates, according to Table 5-3.

Figure 5-9 shows the number of hours the operative temperature is in the respective intervals during working hours (6 am to 6 pm).

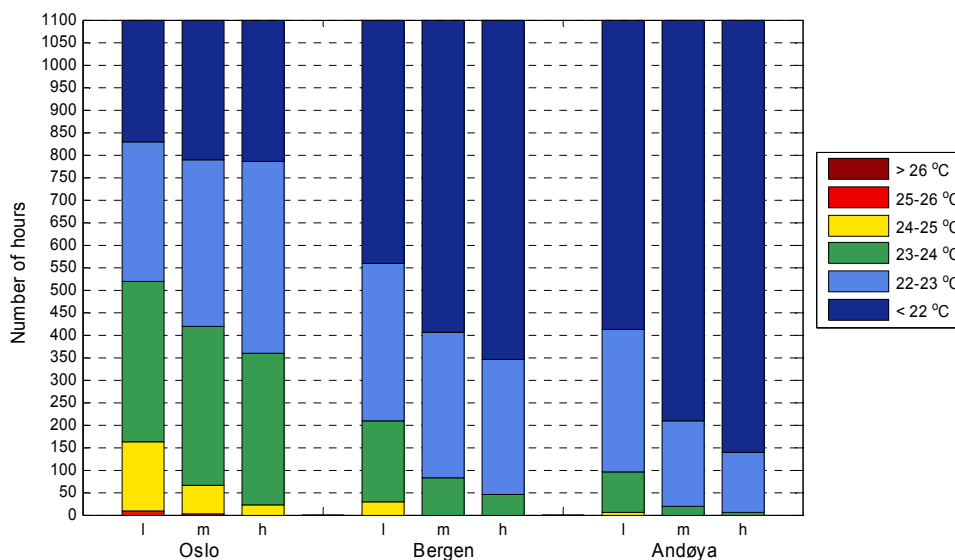


Figure 5-9 Number of hours the operative temperature occurs in different intervals during the working hours with high airflow rate for the light (l), medium (m) and heavy (h) office room, respectively

The simulation of the office rooms ventilated with 10 m³/h m² indicates that the operative temperature will stay well below 26 °C during the entire cooling season for all simulated climates.

Figure 5-10 shows the results from the simulation with a supply airflow rate of 5 m³/h per m².

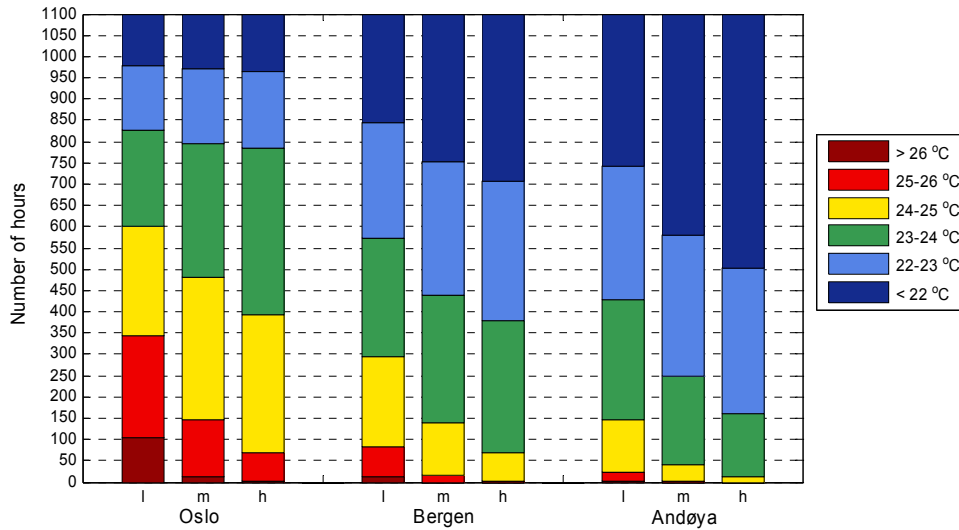


Figure 5-10 Number of hours the operative temperature occurs in different intervals during the working hours with low daytime airflow rate for the light (l), medium (m) and heavy (h) office room, respectively

The simulations indicate that utilization of outdoor air to cool the office rooms outside working hours keep the operative temperature well within acceptable limits for most cases during working hours. The exception is the *light* office room, which in the case of low airflow rate in Oslo experiences about 100 hours of temperatures exceeding 26 °C.

5.4.3 Heating energy demand

In the literature studied in Part I, some authors claim that the utilization of thermal mass reduce the heating energy demand. For the same office cells, the space heating and the ventilation preheating demands are here investigated for Oslo, Bergen and Andøya. The annual heating demands are calculated for both the high (10 m³/h per m²) and low (5 m³/h per m²) air flow rate.

In the first part, the night temperature set-back requirement from the building regulations is implemented for the heating control, i.e. the set-point for heating is 21 °C during working hours and 19 °C outside working hours and weekends. The supply air temperature is 19 °C during the winter (Nov to Mar), 18 °C during the transitional months (Apr and May/Sep and Oct), and 17 °C during the summer (Jun to Aug). Figure 5-11 and Figure 5-12 show the annual space heating and ventilation preheating energy demand for high and low airflow rate, respectively.

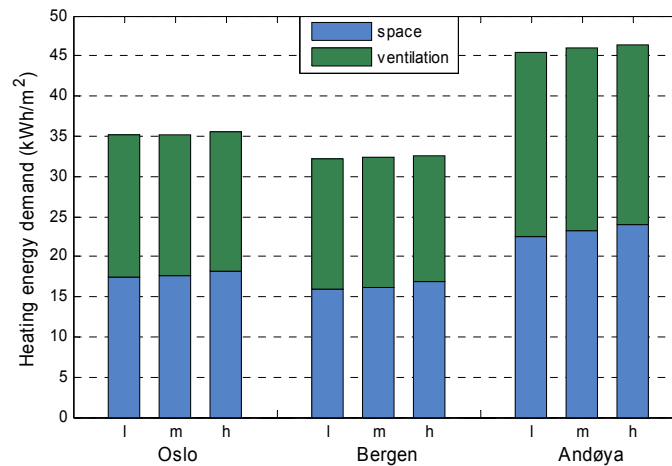


Figure 5-11 Annual space heating and ventilation preheating energy demand for the light (l), medium (m), and heavy (h) office room supplied with high airflow rate

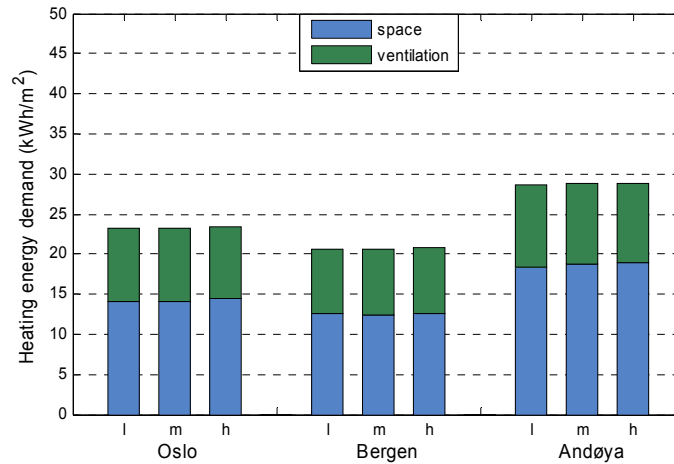


Figure 5-12 Annual space heating and ventilation preheating energy demand for the light (l), medium (m), and heavy (h) office room supplied with low air flow rate

The simulation results indicate that there are only minor heating energy demand differences with respect to thermal mass. The *heavy* room has the largest heating demand, however just at most 0.7 kWh/m² or 1.5 % more than the *light* room.

As pointed out in Section 2.4.4., Burch et al. (1984a) claim that the potential energy savings by night set-back savings decrease with an increase of thermal mass. To investigate this, simulations are run similarly to the simulations above, but without temperature set-back. Figure 5-13 and Figure 5-14 show the results from the simulations without night set-back.

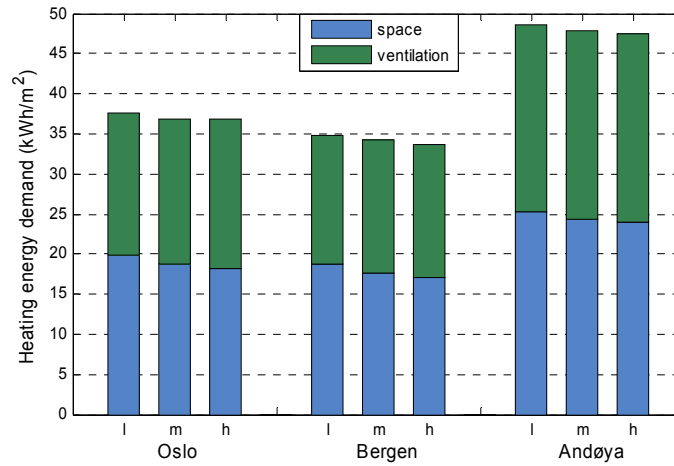


Figure 5-13 Annual heating energy demand without night and weekend temperature set-back and high air flow rate

In contrast to the simulations with night set-back, the simulations without temperature set-back indicate that thermal mass is favourable. However, the heating energy differences are still relatively small. The greatest difference is for Bergen in the case of low air flow rate, where the *heavy* mass room annually demands 1.1 kWh/m² or 4.7 % less total heating energy compared to the *light* room.

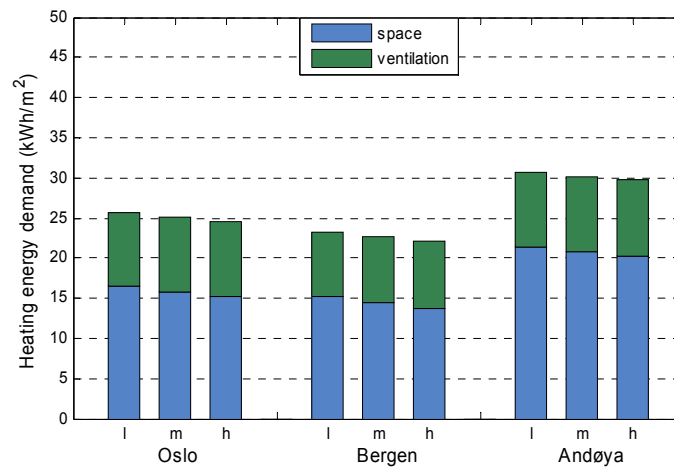


Figure 5-14 Annual heating energy demand without night and weekend temperature set-back and low air flow rate

A comparison of the simulations with night temperature set-back to those without set-back, confirms the observation made by Burch. The potential for saving heating energy with night and weekend temperature set-back is less for thermally *heavy* rooms than for *light* rooms. Figure 5-15 shows that, for example, night set-back for a *heavy* room at Andøya only saves just over 2 % heating energy annually. However, night temperature set-back is still beneficial for all cases considered. Tables with absolute differences are given in Appendix A2.5.

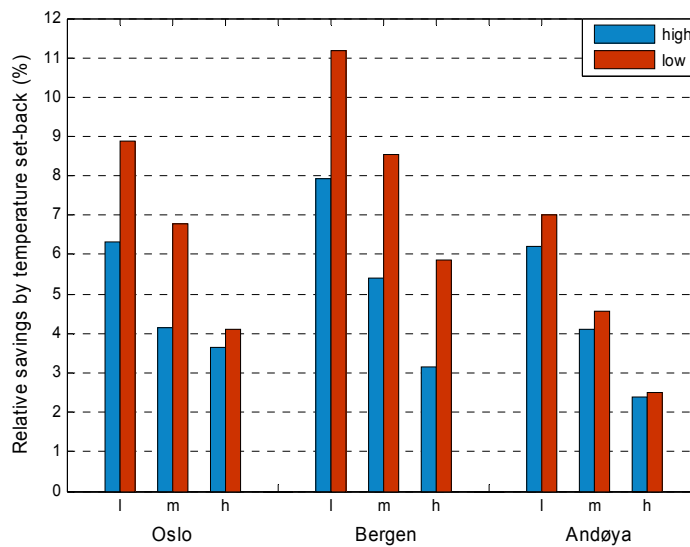


Figure 5-15 Relative annual heating energy savings by night and weekend temperature set-back for high and low ventilation air flow rates

5.4.4 *Closing remarks: Climate*

The simplified degree-hour approach indicates that the Norwegian climate should have a substantial potential to cool buildings outside working hours. The dynamic simulations support these results. The simulations indicate that the utilization of outdoor air to cool the office rooms outside working hours, keep the operative temperature well within acceptable limits for most cases during working hours. The Oslo climate is the most extreme of the climates evaluated here, thus further analyses on summer conditions will be done employing the Oslo DRY file.

The annual simulations done in this section indicate that there are only minor differences in heating energy demand with respect to thermal mass, and the claimed savings from the literature could not be confirmed. The minor differences reported here also agree with the findings in Høseggen et al. (2007), (Paper V in Appendix 4), where the savings were estimated to be less than 3-7 %, dependent on the ventilation system and internal heat gains for a medium mass room compared to a lighter room. Consequently, no further examination of the potential heating energy savings related to thermal mass will be done.

5.5 Influence of occupancy patterns

In the previous two sections, it was assumed that the room occupancy is equivalent to the averaged occupancy for several rooms in a specific office building. It may be questioned whether hourly averaging of the room occupancy is an adequate approach to approximate typical room use. In open plan offices or in rooms where several people work, such an approach should be applicable. However, in single offices, people are either present or absent. In theory, a mass-less building would only require the time to heat or cool the room air mass, and practically no time for heating or cooling to the desired set-point, thus demanding lower overall cooling or heating loads than actual buildings. The question is whether short time absences from the room can be used to cool or heat the room quickly, and make them fully “recovered” when the person is back.

To investigate whether the approach of averaged values is appropriate, the following section deals with different occupancy patterns. Figure 5-16 shows four different patterns, which all have an averaged occupancy of 50 %, assumed working day duration from 8 am to 4 pm.

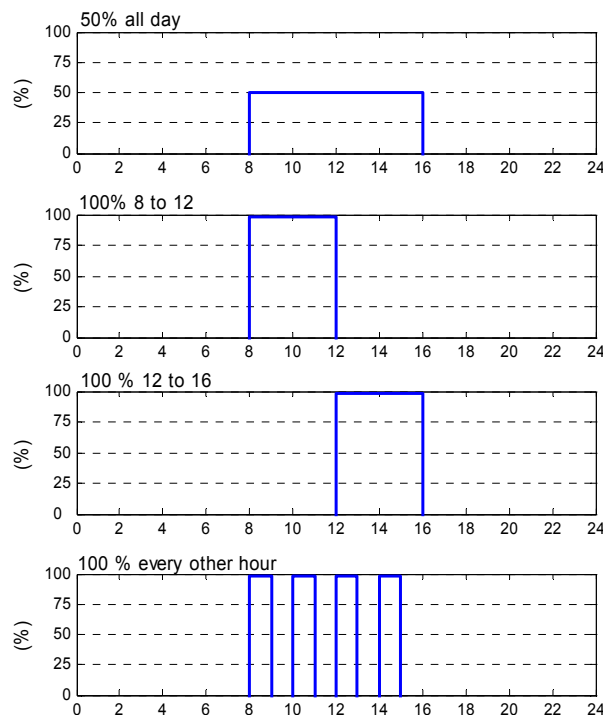


Figure 5-16 Four different fictitious occupancy patterns with an averaged occupancy of 50 % during the working day

5.5.1 Peak operative temperature

This section evaluates the influence of occupancy patterns on the peak operative temperature in the office rooms. The DRY climate file from Oslo is used in the simulations. Figure 5-17 shows the most influential climatic parameters for a warm and sunny week in August.

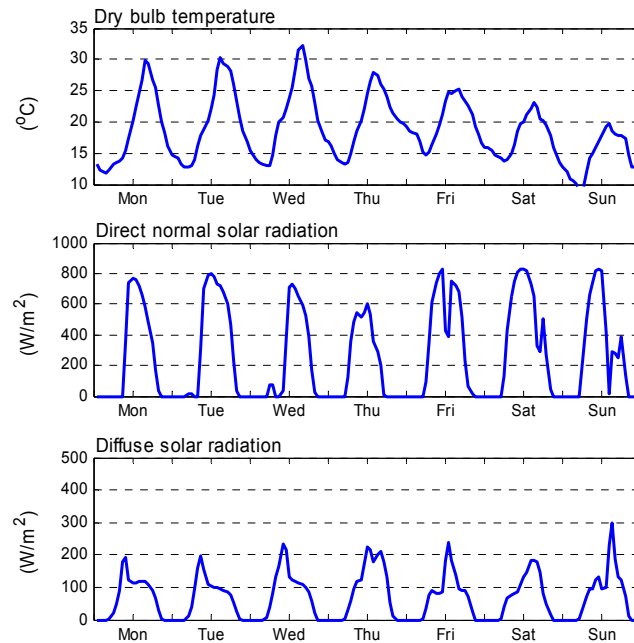


Figure 5-17 The most influential climatic parameters in the DRY file for Oslo for August 14th to August 20th

In the simulations, the offices are mechanically ventilated with conditioned air at supply temperature 17 °C during working hours, and outside working hours if needed, according to Eq. (5.1) and Figure 5-5. To simplify the approach to the problem somewhat, in this section it is assumed that there is no time delay added to the lighting and ventilation controls.

Figure 5-18 to Figure 5-25 show the operative temperatures for the single office rooms on Wednesday in the warm week using the four different occupancy profiles. In Figure 5-18 to Figure 5-21 the high supply airflow is used, and in Figure 5-22 to Figure 5-25 the low supply air flow is used. Appendix A2.6 gives the simulation results from the entire week.

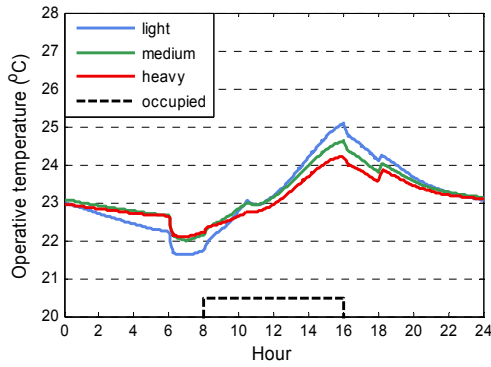


Figure 5-18 Operative temperatures using the “50 % all day” occupancy profile and high airflow rate

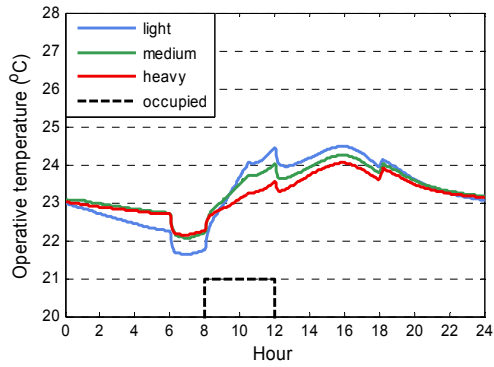


Figure 5-19 Operative temperatures using the “8 to 12” occupancy profile and high airflow rate

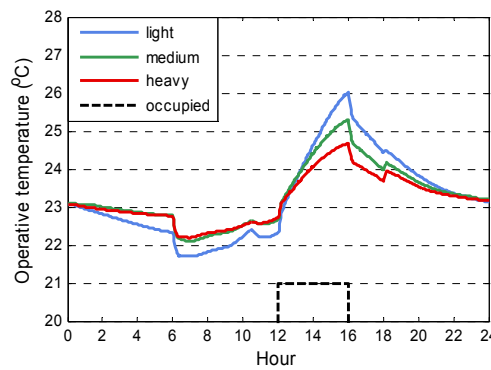


Figure 5-20 Operative temperatures using the “12 to 16” occupancy profile and high airflow rate

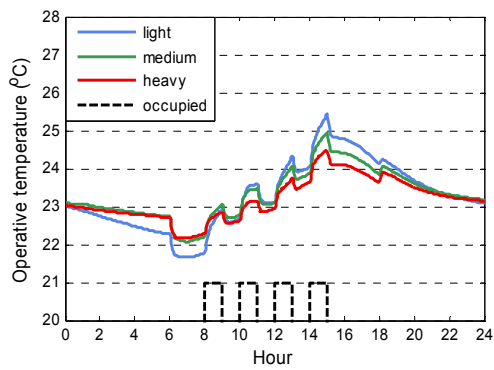


Figure 5-21 Operative temperatures using the “every other hour” occupancy profile and high airflow rate

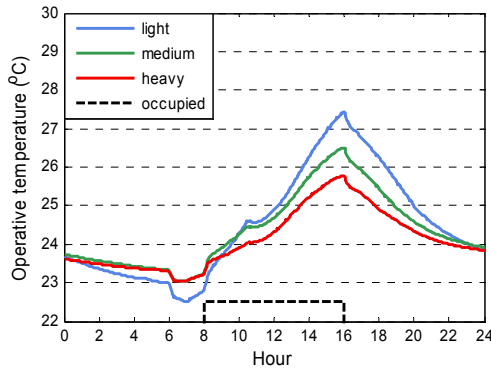


Figure 5-22 Operative temperatures using the “50 % all day” occupancy profile and low airflow rate

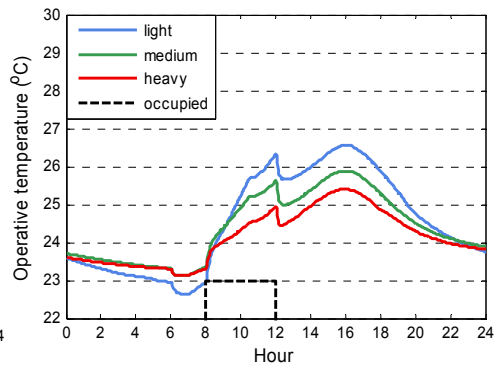


Figure 5-23 Operative temperatures using the “8 to 12” occupancy profile and low airflow rate

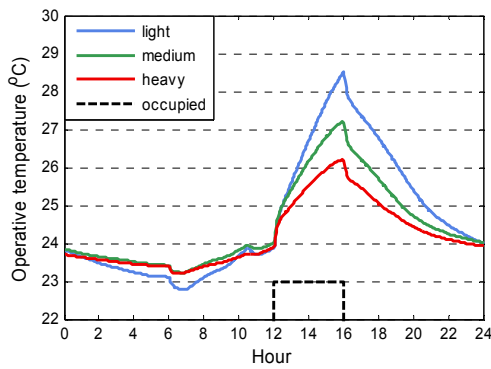


Figure 5-24 Operative temperatures using the “12 to 16” occupancy profile and low airflow rate

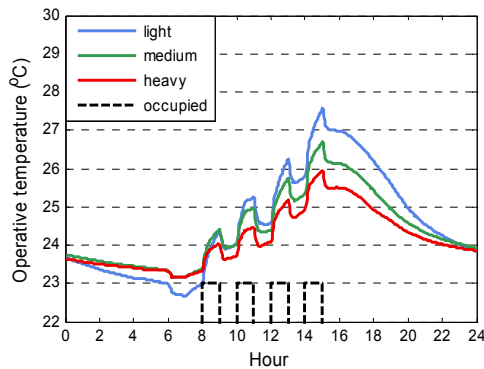


Figure 5-25 Operative temperatures using the “every other hour” occupancy profile and low airflow rate

5.5.2 Operative temperature distribution

This section studies the influence of the occupancy patterns on the seasonal temperature distribution. Simulations are done for both high and low ventilation airflow rate for the cooling season from May 1st to August 31st.

Figure 5-26 shows the percentage distribution of the operative temperature during working hours when the high ventilation airflow rate is supplied to the office rooms.

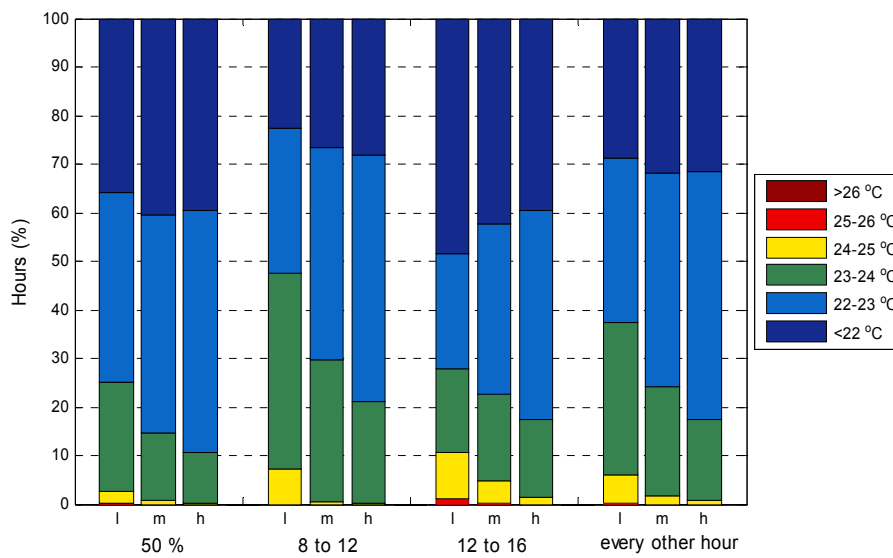


Figure 5-26 Share of the working hours which operative temperatures are in the respective temperature intervals for high airflow rate

Figure 5-27 shows the percentage distribution of the operative temperature during working hours when the low ventilation airflow rate is supplied to the office rooms.

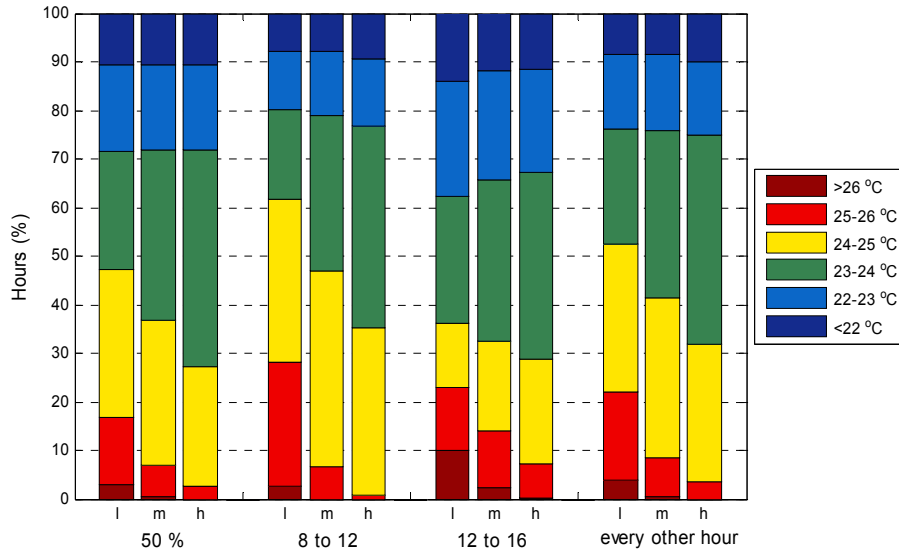


Figure 5-27 Share of the working hours which operative temperatures are in the respective temperature intervals for low airflow rate

In Figure 5-26 and Figure 5-27, it is worth noticing that the influence of the occupancy pattern on the temperature distribution decreases as the rooms are thermally heavier.

5.5.3 Influence of occupancy patterns on the energy demand

To estimate the fan power energy demand, the simplified method employed in Høseggen (2006, Paper III in Appendix 4) is used. In this method it is assumed that the efficiency of the ventilation system η_{tot} is constant for all air flow rates. Provided the ventilation airflow is fully turbulent, the relation $P \sim Q^3$ can be used, where P is the total of all fan power measured as power input to the fan engine (kW) and Q is the total mechanical airflow rate (m^3/s). Thus, the following relation can be obtained:

$$\frac{P}{Q^3} = \frac{P_{nom}}{Q_{nom}^3} \Rightarrow P = P_{nom} \cdot \left(\frac{Q}{Q_{nom}} \right)^3, \quad (5.4)$$

where the subscript *nom* indicates nominal air flow rate. By definition, the specific fan power (*SFP*) is (Mysen, 1999):

$$SFP = \frac{P_{nom}}{Q_{nom}} \quad (5.5)$$

Combining (5.4) and (5.5), following approximation for the fan power can be obtained:

$$P = Q_{nom} \cdot SFP \cdot \left(\frac{Q}{Q_{nom}} \right)^3 \quad (5.6)$$

Figure 5-28 and Figure 5-29 show the fan energy demand for the different profiles together with the net cooling demand for the different occupancy profiles for high and low air flow rate, respectively. The SFP is assumed to be $2 \text{ kW}/(\text{m}^3/\text{s})$.

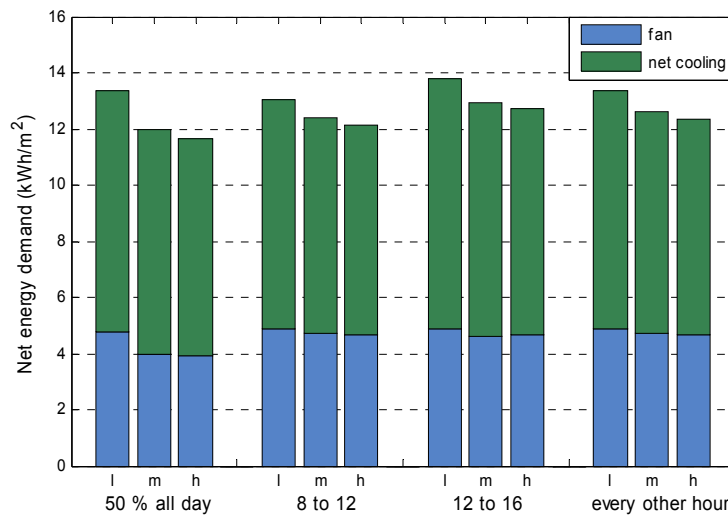


Figure 5-28 Fan energy demand and net cooling demand for the offices employing different occupancy patterns and high air flow rate

The simulations indicate that different occupancy profiles only have minor influence on the energy demand. The greatest difference found is between the “12 to 16” profile and the “8 to 12” profile for high air flow rates and *light* office, where the energy demand of the former is about 0.7 kWh/m² (5.4 %) higher.

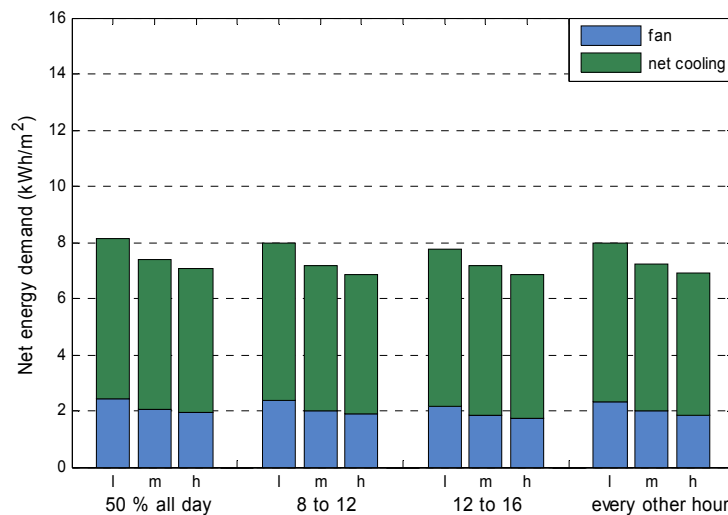


Figure 5-29 Fan energy demand and net cooling demand for the offices employing different occupancy patterns and low air flow rate

5.5.4 Closing remarks: Occupancy patterns

The use of occupancy profiles is not indifferent when it comes to estimation of the thermal environment. Not surprisingly, the “12 to 16” profile is the one that causes both the highest peak operative temperatures and most hours with excessive temperatures during working hours. One of the reasons is that the internal loads and the maximum solar loads occur at the same time, since the offices are facing south-west. In addition, the maximum outdoor temperature also occurs in the early afternoon. As seen in Figure 5-20 and Figure 5-24, the “12 to 16” profile also has the greatest thermal mass effect. The maximum difference between the *light* and the *heavy* room is 1.4 K and 2.3 K for high and low ventilation rates, respectively. In comparison, for the “8 to 12” profile the differences are 0.9 K and 1.4 K.

Also worth noticing, with an increase of thermal mass, the differences between the profiles decrease.

The occupancy profiles are of less importance when it comes to fan power and cooling energy. The energy demand is typically 3-4 % lower for heavy rooms than light rooms. However, the total integrated air volume supplied to the *heavy* room is actually larger than what is supplied to the *light* room over the period. The reason for this is that the *heavy* room distributes the temperature dependent air demand (caused by temperatures exceeding the set-point for cooling) over a broader period of time, while the *light* room demands a full airflow rate more often but in shorter periods of time. Part load air supply is well rewarded according to Eq. (5.6), however, it may be questioned whether the exponent of power should be less than 3. Investigations of VAV systems by Maripuu (2006) showed that the exponent of power can vary from 1.3 to 3.0. The exponent is very dependent on the system characteristics and therefore difficult to assess on a general basis.

As for the number of hours with excessive temperatures, the differences decrease as the thermal mass increases. All things considered, using the averaged occupancy pattern seems to be an adequate approach to approximate the real use of a room.

5.6 Internal heat loads

This section considers the capability of the rooms to handle internal heat loads. As shown in the previous sections, most of the cases with the averaged room occupancy with normal heat gains result in acceptable thermal conditions. According to REN (2007), the operative temperature should basically not exceed 26 °C during the cooling season. However, on warm days it is difficult to keep the temperature within the recommended limits without the use of local cooling equipment. Therefore, exceeding the maximum limit is accepted in periods with high outdoor temperatures. As somewhat cryptically stated in REN; “*Exceeding the highest temperature limit should be accepted during warm summer periods with outdoor temperatures higher than the ones being exceeded with 50 hours in a normal year*”. For the DRY file from Oslo used in these simulations, this should mean by 50 hours.

5.6.1 Mechanical night ventilation

In this section it is assumed that the rooms are occupied the entire day from 8 am to 4 pm. The ventilation air flow is still set according to Eq. (5.1) and Figure 5-5 during day time. Three scenarios are considered during night time for high and low air flow rates;

- no night time ventilation,
- night time ventilation with conditioned air, and
- night time free cooling

The supply air temperature rise due to fan power, friction and heat exchange in the ducts can be significant. Depending on the length of the ducts, airflow rate, and whether the ducts are insulated or not, the temperature rise is in extreme cases for VAV systems measured up to 10 K for low airflow rates (Maripuu, 2006). In the case of night time ventilation with conditioned air, the diffuser air temperature is 17 °C at all times. In the case of mechanical night free cooling, the temperature rise of the outdoor air is taken into consideration approximately. A heat source added before the air inlet rises the temperature in the interval 1 K to 5 K depending on the airflow rate, and is constricted not to exceed the room temperature. Figure 5-30 shows the relation between the free cooling airflow rate and temperature rise that is implemented in the model in the case of free cooling.

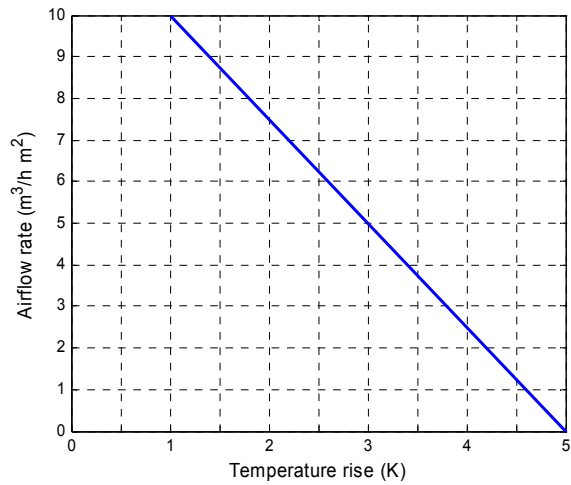


Figure 5-30 Temperature rise for different airflow rates in the case of free cooling

To find the maximum internal heat load the rooms can handle with the given restriction, simulations are run iteratively until the load is found that makes the number of hours with excessive temperatures equal to 50 hours. Figure 5-31 and Figure 5-32 show the maximum internal load that the *light* (l), *medium* (m) and *heavy* (h) rooms can handle employing the different night cooling strategies for high and low air flow rates, respectively.

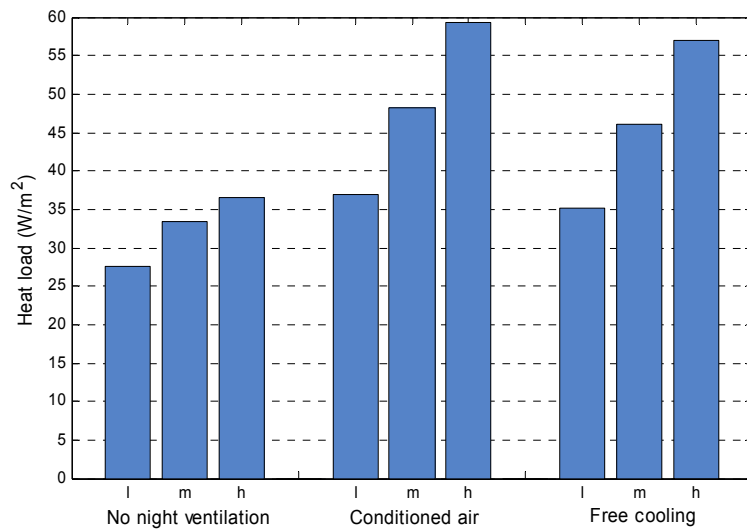


Figure 5-31 Maximum internal heat load the office rooms can handle daytime employing different night ventilation strategies - high airflow rate

The simulation results indicate that the rooms can handle a considerably greater internal heat load if night ventilation is employed. Moreover, the *heavy* room can handle 22.4 W/m² (61 %) higher internal heat loads than the *light* room in case of night ventilation with conditioned air. In comparison, if no night ventilation is provided, the difference is only 9 W/m² (33 %). As shown in Figure 5-32, the tendency is the same for low airflow rates, but the absolute differences are smaller.

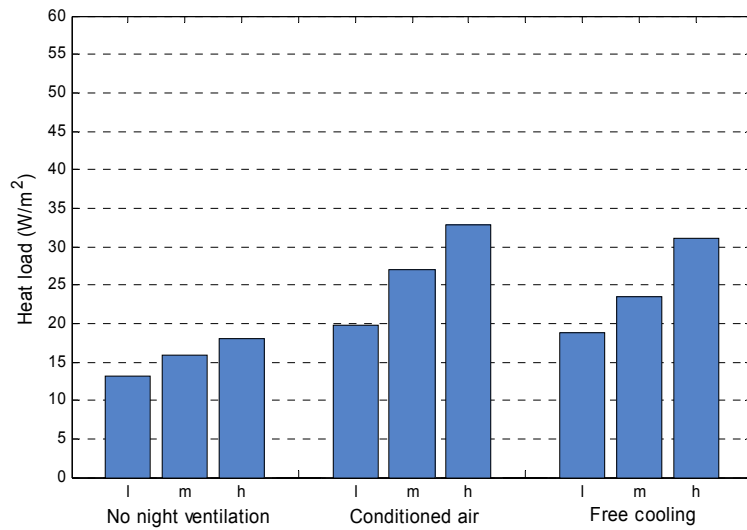


Figure 5-32 Maximum internal heat load the office rooms can handle daytime employing different night ventilation strategies - low air flow rate

5.6.2 Night free cooling

The simulations clearly show that the utilization of night ventilation is advantageous, and that the ability to handle heat loads increases significantly with thermal mass. However, the potential of utilizing night ventilation is limited by the capacity of the ventilation system. To investigate the full potential of utilizing night ventilation, it is therefore assumed that the rooms are free cooled with outdoor air (for instance supplied through operable windows or hatches in the façade) during night time and mechanically ventilated during day time. Both the night time and the daytime ventilation are controlled according to Figure 5-5, which implies that the night ventilation is active as long as the room temperature is above 23 °C.

Figure 5-33 and Figure 5-34 show the maximum internal heat load the rooms can handle as a function of night time air changes for high and low daytime ventilation airflow rates, respectively.

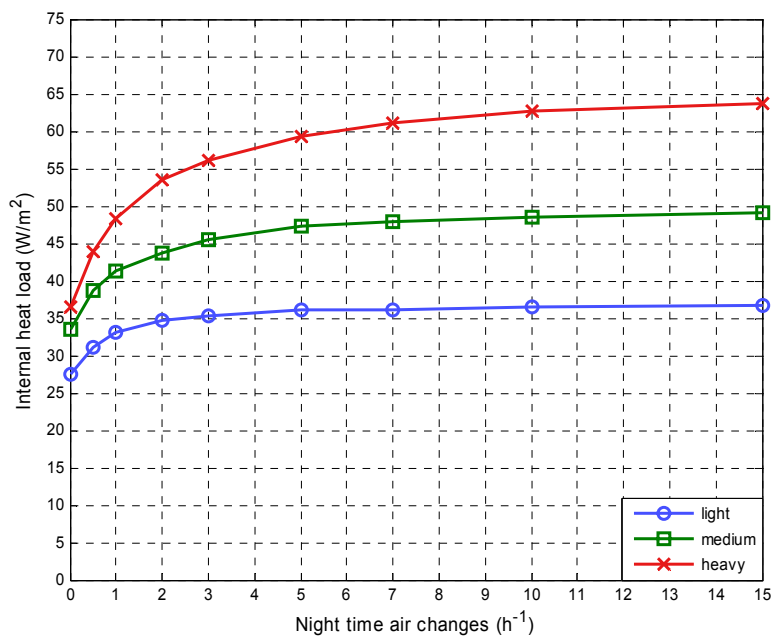


Figure 5-33 Maximum internal heat load the rooms can handle as function of night time air changes. High daytime airflow rate

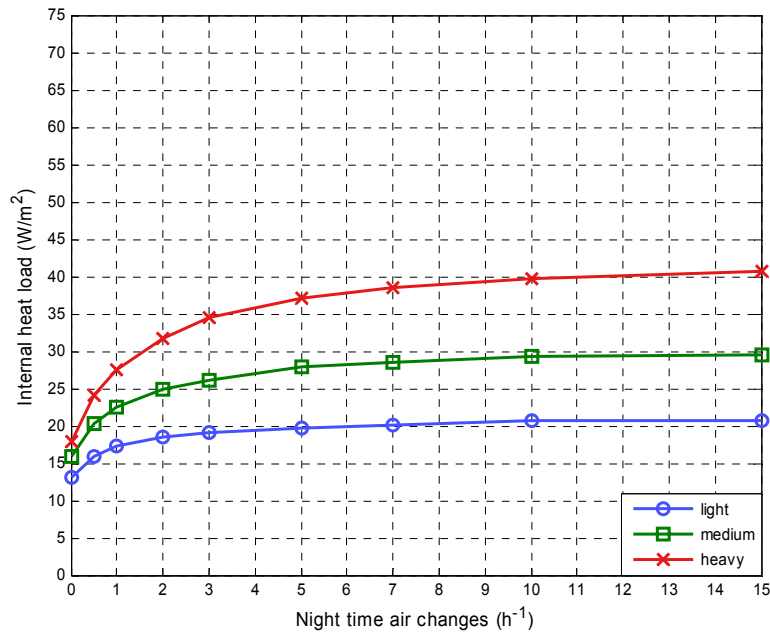


Figure 5-34 Maximum internal heat load the rooms can handle as function of night time air changes. Low daytime airflow rate

In addition to define an upper limit for the number of hours the operative temperature can exceed 26 °C, REN (2007) also recommends a maximum temperature swing of 4 K during the day. In order to assess the full potential within the limits of REN, the set-point temperature for ventilative cooling is in the following simulations set to 22 °C.

Figure 5-35 and Figure 5-36 show the simulation results, which indicate that by lowering the set-point 1 K the rooms can handle an extra internal heat load of about 10 %.

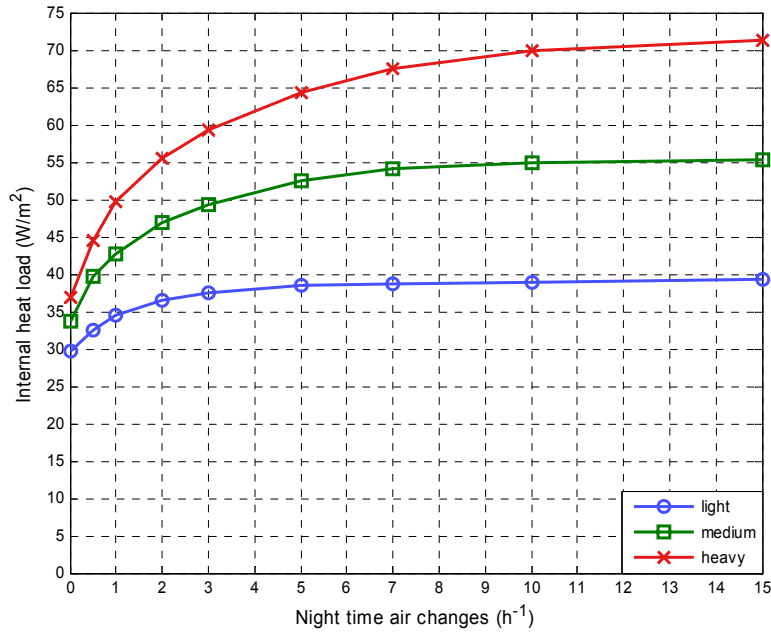


Figure 5-35 Maximum internal heat load the rooms can handle as function of night time air changes. High daytime airflow rate and set-point for ventilative cooling at 22 °C

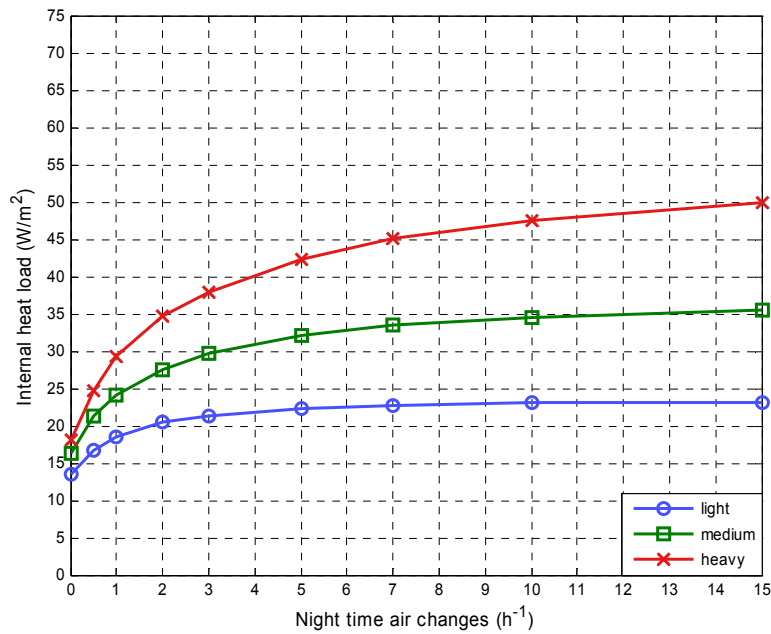


Figure 5-36 Maximum internal heat load the rooms can handle as function of night time air changes. Low daytime airflow rate and set-point for ventilative cooling at 22 °C

5.6.3 Closing remarks: Heat loads

The simulations carried out in this section, clearly show the benefits of utilizing night ventilation in combination with thermal mass. By supplying the offices with air outside working hours, the ability of the heavier rooms to handle daytime internal heat loads is increased significantly compared to leaving the ventilation system turned off outside working hours.

By increasing the airflow rate outside working hours, the potential of handling large daytime internal heat gains increase further for the heavier rooms. The *light* room experiences no greater improvement by increasing the night time airflow rate beyond 1-2 ach. Although the ability to handle heat loads still increase beyond the limits of the figures, most of the potential for the *medium* and *heavy* rooms is reached at about 7 and 10 ach respectively.

5.7 Influence of set-points and control ranges for ventilative cooling

This section investigates the influence of set-point for ventilative cooling and temperature control intervals on the energy demand. In Mathisen et al. (2005), (Paper I in Appendix 4), it was found for a building with a hybrid ventilation system that the set-point and allowed temperature swing had significant influence on the energy demand.

In the previous sections, the heating and cooling have been controlled according to Figure 5-5, which has a set-point for cooling at 23 °C and a control range of 2 K. In this section other control ranges are evaluated for set-points for cooling at 22 °C and 23 °C. Both high and low ventilation airflow rates are studied.

5.7.1 Set-point for ventilative cooling at 22 °C

The simulation model is similar to the basic model described in Section 5.2, except that Figure 5-37 replaces Figure 5-5 for the temperature dependent HVAC control strategy.

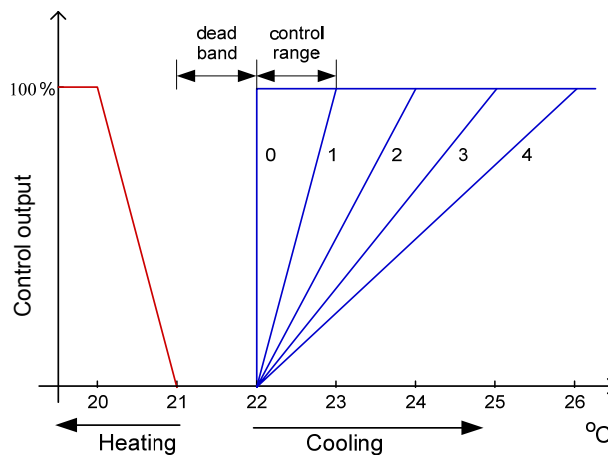


Figure 5-37 Heating and cooling control strategy with set-point for ventilative cooling at 22 °C

Figure 5-38 and Figure 5-39 show the fan energy and net cooling demands for different control ranges for high and low airflow rates, respectively. The fan and cooling energy demand is calculated approximately as shown in Section 5.5.

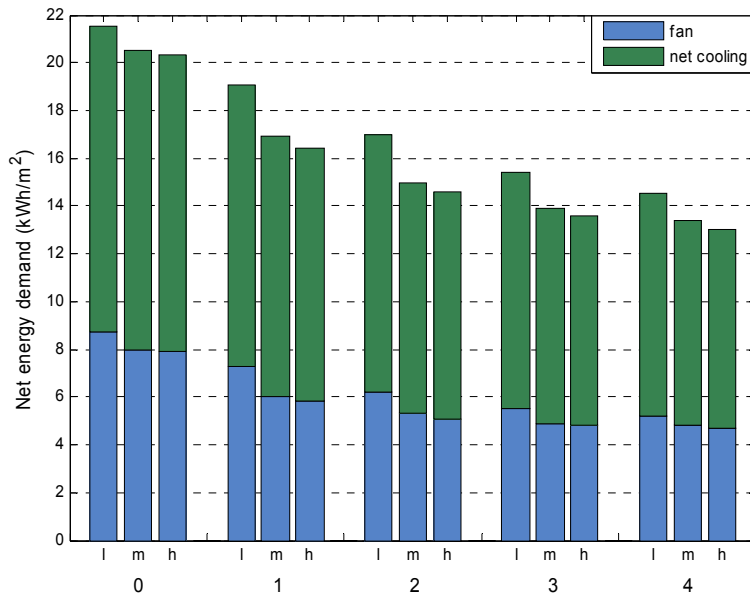


Figure 5-38 Fan power and net cooling energy demand for different control ranges and high ventilation airflow rate. Set-point for ventilative cooling is 22 °C

For high ventilation airflow rates, all control ranges are applicable, in the sense that none cause the number of hours with excessive temperatures to exceed 50.

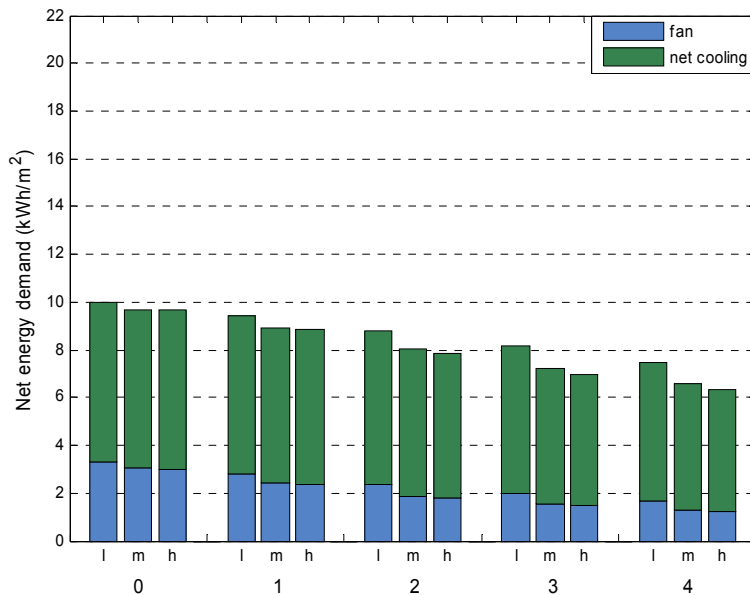


Figure 5-39 Fan power and net cooling energy demand for different control ranges and low ventilation airflow rate. Set-point for ventilative cooling is 22 °C

For low ventilation airflow rates, the number of hours with excessive temperatures in the *light* office room is approaching the critical value. Table 5-5 summarizes the number of hours the operative temperature exceeds 26 °C during working hours for the office rooms.

Table 5-5 Number of hours the operative temperature exceed 26 °C during working hours for set-point at 22 °C and low ventilation airflow rate

	Control range (K)				
	0 K	1 K	2 K	3 K	4 K
<i>Light</i>	41	52	83	101	132
<i>Medium</i>	0	2	3	8	13
<i>Heavy</i>	0	0	0	0	2

5.7.2 Set-point for ventilative cooling at 23 °C

The fan and net cooling energy demands are also evaluated for a set-point for cooling at 23 °C. Figure 5-40 shows the temperature dependent HVAC control strategy.

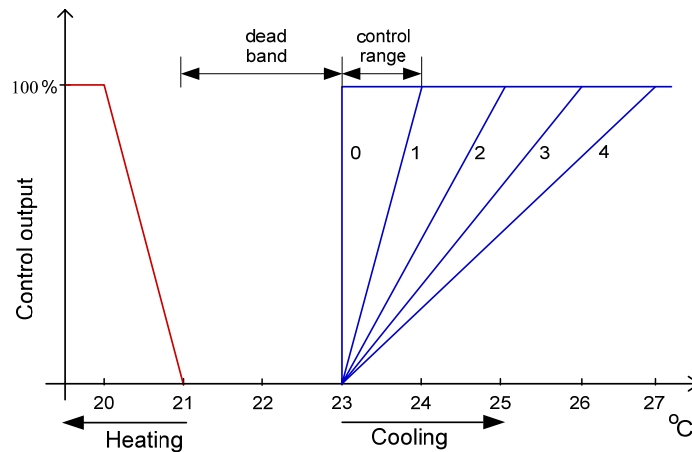


Figure 5-40 Heating and cooling control strategy with set-point at 23 °C

Figure 5-41 and Figure 5-42 show the fan energy and net cooling demands for different control ranges for high and low airflow rates, respectively.

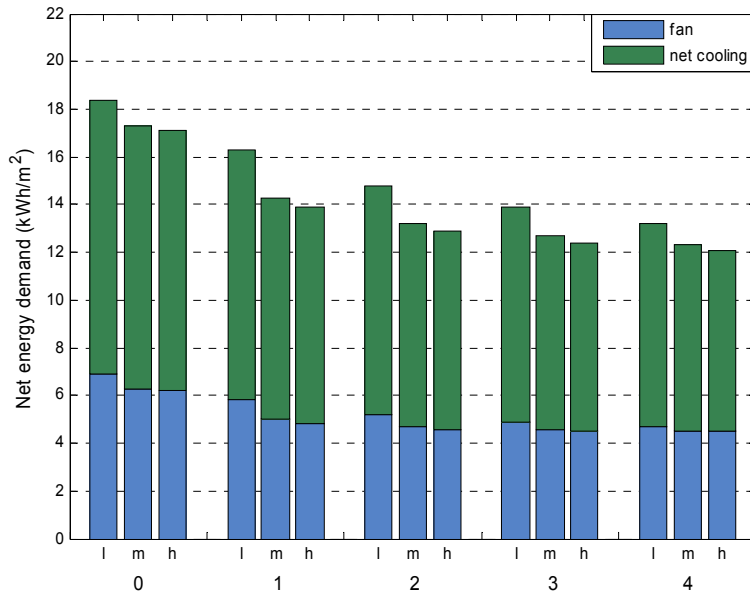


Figure 5-41 Fan power and net cooling energy demand for different control ranges at high ventilation airflow rate. Set-point for ventilative cooling is 23 °C

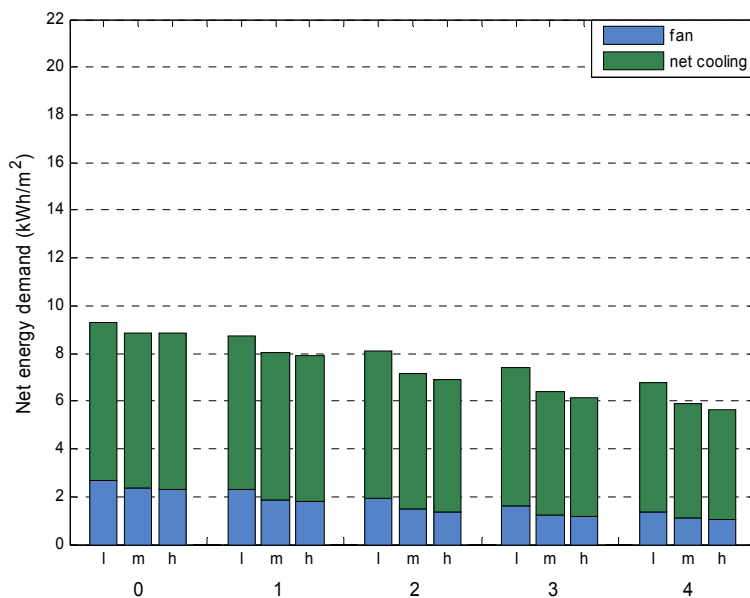


Figure 5-42 Fan power and net cooling energy demand for different control ranges at low ventilation airflow rate. Set-point for ventilative cooling is 23 °C

By increasing the set-point for cooling by 1 K to 23 °C, the net energy demand in the case of high airflow rate decrease about 7-15 % for the control ranges, respectively. For low airflow rate, the demand decrease about 7-12 %.

As for a set-point at 22 °C, the number of hours with excessive temperatures is acceptable for all rooms and control ranges for a high airflow rate. However, as seen in Table 5-6, for a low airflow rate applied to the light office room, the limit is exceeded for any control range. Even for the *medium* room excessive temperatures are critical when the control interval is 4 K. On the other hand, as seen in Figure 5-41 and Figure 5-42, the energy savings gained by increasing the control range beyond 3 K are sparse when the set-point for cooling is 23 °C.

Table 5-6 *Number of hours the operative temperature exceed 26 °C during working hours for set-point at 23 °C and low ventilation airflow rate*

	Control range (K)				
	0 K	1 K	2 K	3 K	4 K
<i>Light</i>	95	110	147	178	218
<i>Medium</i>	5	10	16	32	59
<i>Heavy</i>	0	0	2	6	15

5.7.3 *Closing remarks: Set-points and control ranges*

For small control ranges and low set-point, thermal mass is only sparsely utilized, and only minor energy demand differences between the rooms is observed. By increasing the set-point for cooling and allowing the temperature to float somewhat before full airflow is supplied, the *medium* and *heavy* rooms perform significantly better than the light room.

An increase of the control range beyond 3 K to 4 K does not improve the performance significantly, and will also for the *medium* room at low airflow rate, cause the number of hours with excessive temperatures to approach the critical value.

It is worth noticing that the heavy room performs only slightly better than the medium heavy room concerning fan power and net cooling energy demand.

5.8 Influence of convective heat transfer correlations

By default, ESP-r uses natural convection correlations for building surfaces. However, as discussed in Section 2.1.3, the choice of heat transfer coefficients may greatly influence the simulation results. In this section, the influence of the air supply diffusers on the convection regimes and their impact on the operative temperature is evaluated. For the office rooms, two cases are studied;

- Buoyant forces are assumed to dominate the airflow regime and the default Alamdari and Hammond (1983) correlations are applied to the room surfaces at all times.
- The Fisher and Pedersen (1997) correlations for ceiling radial jets are applied when the ventilation system is running. Outside the time of operation, the Alamdari and Hammond correlations are applied.

The heat transfer correlations by Fisher and Pedersen are applicable only for supply temperatures between 10 °C and 25 °C and for enclosure air change rates from 3 to 100 ach. Hence, only the case of high airflow rate (equivalent to 3.6 ach) is considered. The convection correlations used in this study can be found in Appendix A2.7.

In this section, it is assumed that the rooms are occupied the entire day from 8 am to 4 pm. The internal gains from Table 5-3 are used, i.e. in total 34.5 W/m², and the ventilation air flow is still set according to Eq. (5.1) and Figure 5-5.

5.8.1 Influence on the peak operative temperature

Figure 5-43, Figure 5-44, and Figure 5-45 show the operative temperature on Wednesday from the warm week in August presented in Section 5.5, for the *light*, *medium*, and *heavy* office rooms, respectively.

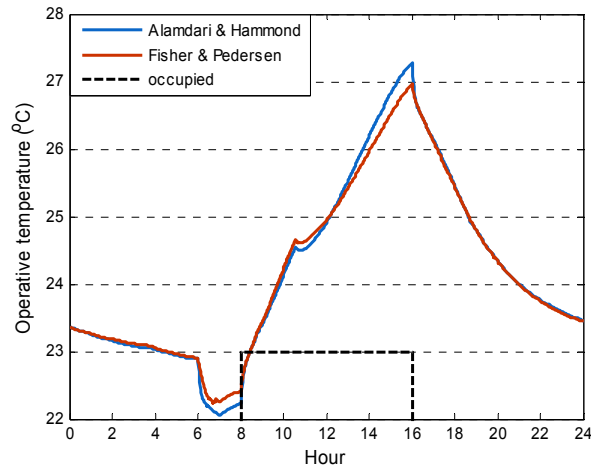


Figure 5-43 Operative temperature in the light office room assumed when natural forces dominate at all times (Alamdari & Hammond) and when forced convection is taken into account (Fisher & Pedersen)

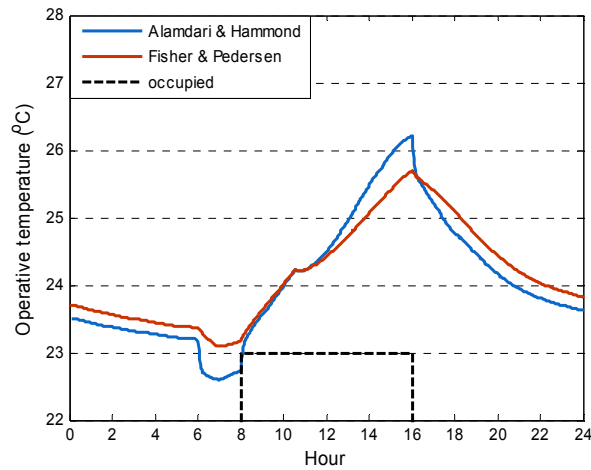


Figure 5-44 Operative temperature in the medium office room when natural forces dominate at all times (Alamdari & Hammond) and when forced convection is taken into account (Fisher & Pedersen)

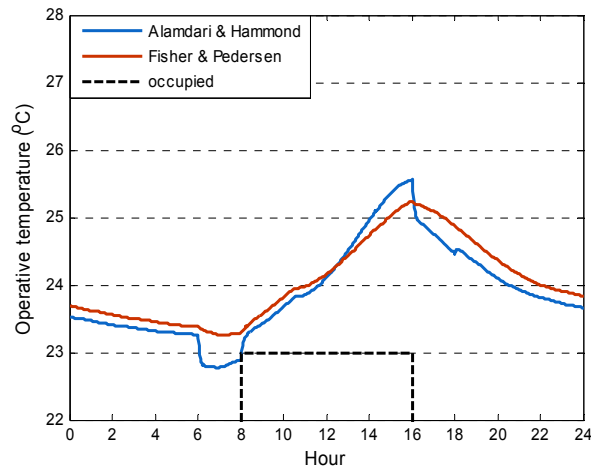


Figure 5-45 Operative temperature in the heavy office room when natural forces dominate at all times (Alamdari & Hammond) and when forced convection is taken into account (Fisher & Pedersen)

The simulation results indicate that the choice of convection correlations significantly influence the room operative temperatures. Moreover, the greatest effect is for the *medium* room, which experiences a decrease of 0.6 °C of the peak operative temperature. Table 5-7 summarizes the peak operative temperatures for the rooms.

Table 5-7 Peak operative temperatures and peak temperature differences

	Peak temperature (°C)		Difference (K)
	Alamdari & Hammond	Fisher & Pedersen	A&H - F&P
<i>Light</i>	27.3	27.0	0.3
<i>Medium</i>	26.2	25.6	0.6
<i>Heavy</i>	25.6	25.2	0.4

5.8.2 Capability to handle internal heat loads

Since the choice of convection correlations improves the ability of the room to decrease the peak operative temperatures, it should be interesting to see whether the forced convection regimes also increases the internal heat load the room can handle, as studied in Section 5.6.

Figure 5-46 shows the maximum internal heat load the *light* (l), *medium* (m) and *heavy* (h) rooms can handle, when the Alamdari and Hammond correlations (similar to Figure 5-31 in Section 5.6) and the Fisher and Pedersen convection correlations are applied.

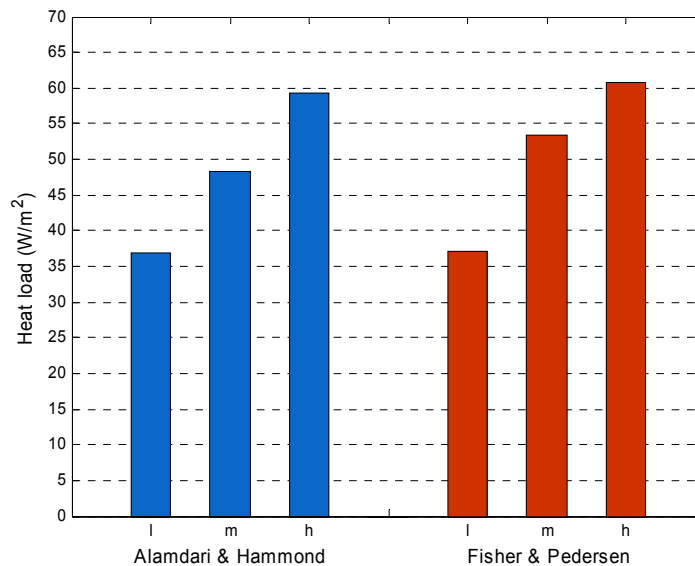


Figure 5-46 Maximum internal heat load the office rooms can handle daytime with the different correlations applied

The *medium* room experiences the greatest improvement in handling internal heat loads by applying the Fisher and Pedersen correlations. When taking into account the forced convection regime when the ventilation system operated, the *medium* room can handle an internal heat load of about 10 % higher than if only natural convection forces are assumed. In comparison, the *light* and *heavy* rooms only experience an improvement of less than 1 % and 3 %, respectively.

5.8.3 *Closing remarks: Heat transfer coefficients*

Improved heat transfer as a consequence of the forced convection regimes will in practice make the rooms heavier because the effective thermal mass increases. This is evident in the morning, as the temperature drops more rapidly when only natural convection correlations are used when the ventilation system switches on at 6 am. The peak operative temperatures are also decreased compared to when only buoyancy forces are taken into consideration.

The simulations done in this section indicate that if the supply air is forced along the ceiling surface, the performance of the *medium* room improves relatively more than the *heavy* and *light* rooms. The explanation is that in case of the *heavy* room, relative increase of the effective thermal mass is smaller than the *medium* room. For the light room, the forced convection does not increase the effective thermal mass considerably, because the ceiling construction cannot absorb much heat due to the light construction of the suspended ceiling.

5.9 Summary

A summary now outlines the most important findings and results from the parametric study.

5.9.1 Operative temperature

Both the simplified analytical method and dynamic simulations indicate that the climates evaluated in this study should have a large potential to cool buildings outside working hours. The simulations indicate that the utilization of outdoor air to cool the office rooms outside working hours, keeps the operative temperature well within the limits of comfort for most cases during working hours.

The peak operative temperature during a warm and sunny day is significantly decreased for the *medium* and *heavy* rooms compared to the *light* room. This is proven to be the case for the occupancy patterns, internal heat loads and climates similar to those evaluated here.

If the air is supplied to the rooms along the ceiling surface, the peak operative temperatures are decreased further. The most important decrease is observed for the *medium* room, which experiences 0.6 K reduction of the peak operative temperature compared to cases where the forced convection regime is not taken into account.

5.9.2 Capability to handle internal heat loads

Thermal mass combined with night time ventilation make the rooms capable of handling considerable higher daytime internal heat loads than light rooms. Provided thermal capacity is equivalent to the *medium* room or more, high daytime ventilation rates, and sufficient night ventilation, a single office room with normal use will stay well within the limits of comfort without space cooling. Even though the *medium* room supplied with a low airflow rate can handle only the internal heat load of about half of the load with a high airflow rate, one must keep in mind that the simulations on internal heat loads are done with the assumption that the room is occupied constantly.

Night cooling with outdoor air can substitute air-conditioned ventilation without a significant decline in performance. If outdoor air is supplied at rates beyond the capacity of the ventilation system, the ability of the rooms to handle daytime internal heat loads

increases. While the capacity of the *light* room is reached already at 2 ach, the *medium* and *heavy* rooms increase their respective performance up to 7 ach and 10 ach and beyond.

Concerning peak operative temperature, the *medium* room makes best use of the forced convection regime that a radial ceiling diffuser provides. While the light and heavy rooms barely increase the ability to handle internal heat loads, the *medium* room increases its ability by about 10 % compared to the case with only natural convection forces involved.

Based on the simulations, the ambition of TEK, that local cooling should be avoided, is applicable in any office building, provided that there is sufficient night ventilation and thermal mass. However, and this cannot be emphasized enough – this is valid only when the external heat loads are controlled. This means that effective external sun shading is crucial for the results from this study to be valid.

5.9.3 Energy savings

The energy assessments done in this study have been related to the cooling of ventilation air, fan power, space heating, and preheating of ventilation air. The simulations suggest that thermal mass improves the energy efficiency. In this study it is found that the *heavy* room demands about 10-20 % less total fan power and net cooling energy than the *light* room. Moreover, the *medium* room has about the same potential for saving fan power and cooling energy as the *heavy* room.

The simulations with focus on the set-point for cooling and control ranges showed that the temperature should be allowed to float somewhat before full air supply to utilize the thermal mass. A control range of 2-3 K should be allowed, however, with an increase of the control band beyond 3 K only minor savings are gained and the risk of excessive temperatures is increased.

However, the simulations done could not confirm the claimed benefits from the literature of utilizing thermal mass to reduce the annual heating demand. Only minor differences in heating energy demand with respect to thermal mass were found. The minor differences reported here (about 1 % - 5 %) are in accordance with the findings in Høseggen (2007, Paper V in Appendix 4), where the savings were estimated to be less than 3-7 %, depending on the ventilation system and room occupancy for a medium mass room compared to a light mass room.

5.9.4 *Energy savings vs. thermal environment*

The simulations done in this parametric study show that thermal mass improves the energy efficiency in buildings, thus decreases the building operation costs. However, the fact that the number of hours with high operative temperature is decreased and that the temperature is more stable during the working hours may constitute a greater economic savings potential due to increased productivity.

For example, if the working performance loss curve introduced in Figure 1-5 is employed on the simulation results with Oslo climate from Figure 5-9, where all rooms are within comfort limits, the light room has a loss of productivity of 6 hours, the medium room 3.5 hours, and the heavy room 3 hours. Although the loss of productivities seem modest, applied to a building where several hundred people work, productivity loss due to high temperatures may be significant. And again, note the small difference between the *medium* room and the *heavy* room.

PART III:

EXPERIMENTAL STUDY

FIELD STUDY

Part III is an extended description of the field study that formed the basis of the paper '*The effect of suspended ceilings on energy performance and thermal comfort*', which is found in Appendix 4.

The sections in this part will give some supplementary description on the instrumentation, reliability of the measurements, and the achieved results.

6. FIELD STUDY

In the literature reviewed in Part I, only a few studies were found that both assessed the potential heating and cooling load benefits of exposed thermal mass in office buildings. Moreover, the experimental studies were solely on test cells, and they were not taking user behaviour and advanced building control systems into account. The objective of the field study presented here was to use measurements to determine the potential energy savings and thermal environmental benefits of exposing the concrete ceiling to the indoor air as an alternative to the original suspended ceiling in a real building in operation.

The field study was carried out during the period from October 1st 2005 to October 15th 2006. The case used in this study is an office building on the Nord-Trøndelag University College (HiNT) campus in Levanger (63.75°N), located 80 km north of Trondheim, Norway (see map in Section 5.3). The building, which will be referred to as *Røstad*, is located in rural surroundings close to the fjord. The climate can be considered coastal.



Figure 6-1 The case building viewed from the south

6.1 Building description

The building, which was ready for occupation in August 2002, has one wing with common and educational areas, and two office wings. Here, only one of the office wings (the TKS wing) was studied. The TKS wing has a separate ventilation system and can be controlled individually from the rest of the building.

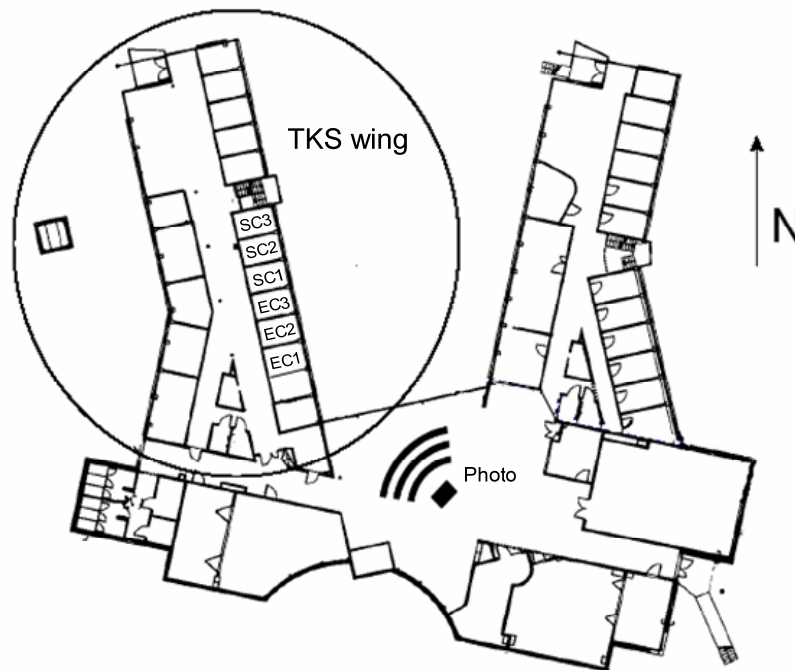


Figure 6-2 Plan view of the second floor showing the common area and the two office wings. The point labelled “Photo” is from where the photo in Figure 6-1 is taken. (Modified from Statsbygg, 2002)

The TKS office wing has mostly single office rooms and a few meeting rooms. In all, the wing has 38 rooms and a total available area of about 670 m².

6.1.1 Ventilation system

The hybrid ventilation is of so-called culvert type. Air is brought into a concrete duct embedded in the ground via an air intake tower and flows into to the distribution culvert, which is located on the central axes of the building. In principle, the system at *Røstad* is constructed as shown in Figure 6-3 and Figure 6-4.

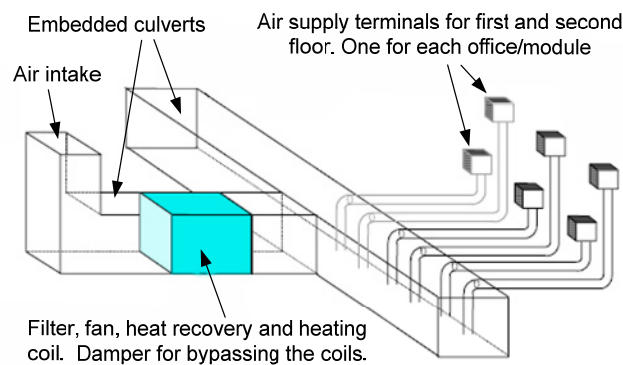


Figure 6-3 The principle of the culvert (Mathisen, 2004)

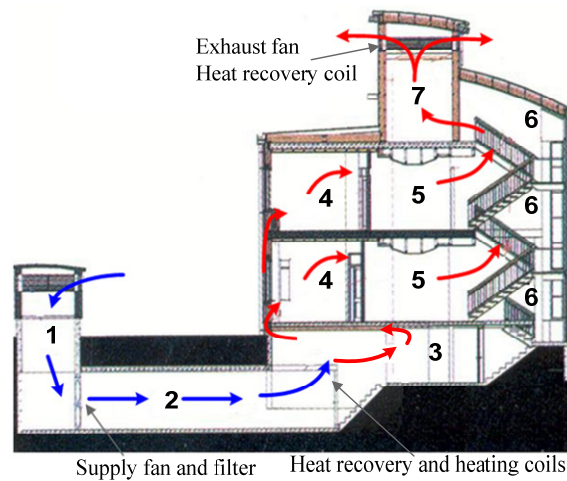


Figure 6-4 Cross section of the building with culverts. 1: Air intake tower. 2: Air intake culvert. 3: Air distribution culvert. 4: Offices. 5: Corridors. 6: Stairway. 7: Exhaust air tower. (Modified from Statsbygg, 2002)

The use of concrete air supply ducts has some advantages. During winter time, when the outdoor temperatures are low, the culvert contributes to preheat the ventilation air due to the relatively warmer wall and ground temperature. Conversely, on hot summer days, the

walls provide a cooling effect since the temperature in the ground is relatively cooler than the outdoor air (Wachenfeldt, 2003).

The ducts from the culvert to the rooms are buried in the ground beneath the floor. At the façade the ducts turn 90° upwards. The ducts end in dampers placed inside the supply air terminal device. The air diffusers are placed at the floor beneath the windows. From the offices the air flows through grilles placed close to the ceiling and into the corridor. Exhaust of air takes place through corridors and stairway up to the tower on the roof. The exhaust air tower contains a heat recovery coil and a fan.



Figure 6-5 Inside of the air distribution culvert, showing the air supply duct stubs. Each duct supplies an office room (Mathisen, 2004)

6.1.2 Heating and ventilation control

Figure 6-6 shows the principle of the ventilation system. The supply and exhaust fans are controlled by the pressure difference between the culvert and corridor on the second floor, i.e. the fans keep the pressure difference constant. When there is no heating demand for the ventilation air and the outdoor temperature exceeds 15 °C, the bypass dampers open to reduce the pressure drop.

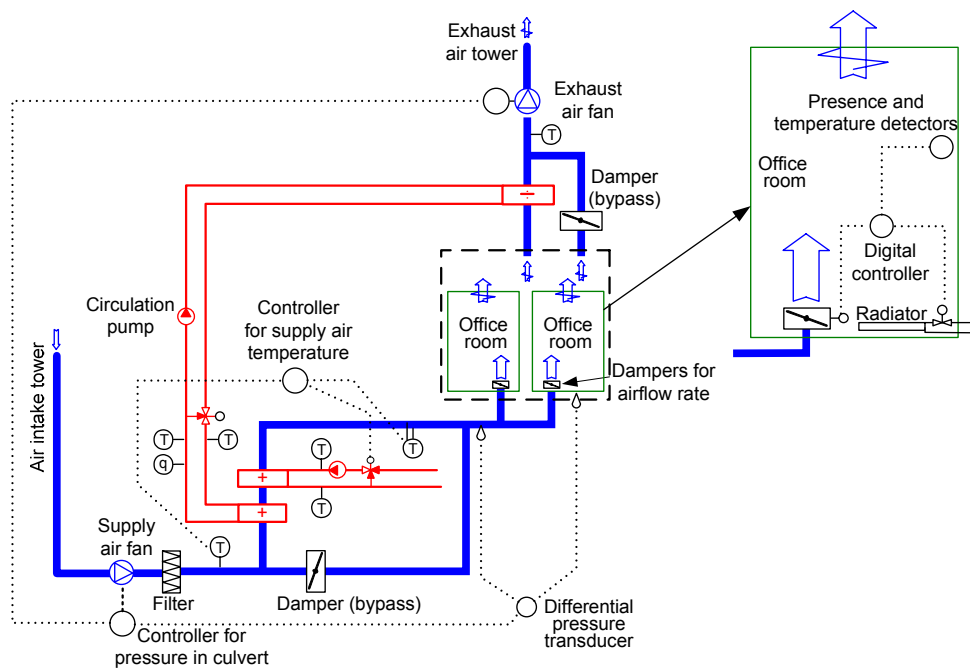


Figure 6-6 Principle of the ventilation system

The dampers in the supply air terminal devices control the airflow rate to each room. They operate as follows: Normally they close at 4 pm. At 6 am the dampers open to give approximately 25 m³/h (3.3 m³/h per m²) ~ 1.2 ach) of air. When a person enters the room they open to supply 43 m³/h (5.7 m³/h per m²) ~ 2 ach). If the room air temperature exceeds the set-point for ventilative cooling they open further. The dampers will continue to open proportionally with the temperature until they reach full opening at 25°C, supplying the rooms with about 200 m³/h (26.7 m³/h per m²) ~ 9 ach). If the room air temperature is above the set-point for ventilative cooling, the dampers also open at night, provided that the outdoor temperature is above 15 °C. Figure 6-7 shows the temperature dependent heating and ventilation control.

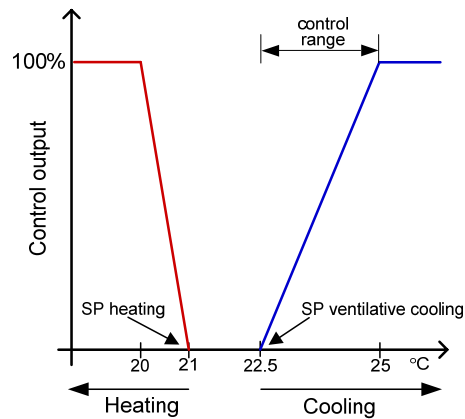


Figure 6-7 Heating and ventilation control strategy

Originally the set-point for ventilative cooling was 21.3 °C and had a control range of 1 K before full air supply rate. After the findings in the study by Mathisen and Høseggen (2005, Paper I in Appendix 4), which evaluated the influence of the set-point and control range on the heating energy demand, the set-point for ventilative cooling and control range were increased to the present.

6.1.3 Office rooms

Each office has a presence detector, a temperature sensor and a digital controller unit that controls heating, ventilation and lighting. A room is registered as empty if the detector has not detected any movement for five minutes. To avoid the ventilation and lighting to turn on and off for short room absences, a 20 minute time delay is implemented for these controls. This means that a person must be absent more than 20 minutes before the lights switch off, the set-point changes and the ventilation is turned down.

The six rooms of special interest in this study (Room *EC1* to *EC3* and *SC1* to *SC3*, see Figure 6-2) are located on the second floor at the east-north-east side of the building. From this point on, the rooms with exposed concrete in the ceiling will be referred to as *EC*, and the rooms with the original suspended ceiling as *SC*.



Figure 6-8 Room with the suspended ceiling removed (EC)



Figure 6-9 Room with suspended ceiling (SC)

The rooms are about 7.5 m² and are equally equipped with a laptop, an external LCD-screen, are similarly furnished and have the same heating and lighting equipment. The office rooms have painted plaster internal walls and glazed wall to the hallway. The floors are concrete with a linoleum covering. All rooms originally had suspended ceilings of painted plasterboard with 50 mm insulation above, which are removed in Room EC1 to EC3.

The reason why six offices were involved in the study was to make the rooms adjacent to EC2 and SC2 similar, thus minimizing boundary differences.

6.2 Instrumentation

6.2.1 Climate parameters

Outdoor dry bulb temperature and wind data were monitored by the building energy management system (BEMS). The solar data were collected using the BF3 Sunshine Sensor (Delta-T, 2006), which measures both global and diffuse radiation. Appendix A3.1 shows how the direct normal radiation is found from the global and diffuse radiation. The relative humidity was recorded using the Tinytag Plus 2 logger (Intab, 2006)

The climatic parameters were used for calibrating a simulation model, as shown in Paper V.

6.2.2 Room and building parameters

Room air temperature, air supply damper positions, radiator valve positions and room set-points were registered every 15 minutes by the building energy management system (BEMS). The occupancy of every room in the building was also registered. In addition, *Room EC2* and *Room SC2* were equipped with six thermocouples connected to a logger to get a more detailed picture of the air temperature stratification, air supply temperature and surface temperatures.

6.2.3 Measurement uncertainties

The thermocouples measuring the room temperatures were calibrated using an ice/water mix as reference before they were assembled. Although the thermocouples of the type used in this experiment (T-type) has an uncertainty of less than 0.1 K in the temperature interval in question, the loggers used raised the uncertainty to 0.3 K. The error ranges of the BEMS room temperature sensors are reported by the manufacturer to be 0.5 K.

Since both heating and ventilation control outputs are dependent on the room temperature, it was crucial that the room temperature sensors were accurate. In Figure 6-10 and Figure 6-11 the temperatures measured by the thermocouples are compared to the temperature recorded by the BEMS for *Room EC2* and *Room SC2*, respectively. The periods from each season chosen for further analysis are also marked in the figures.

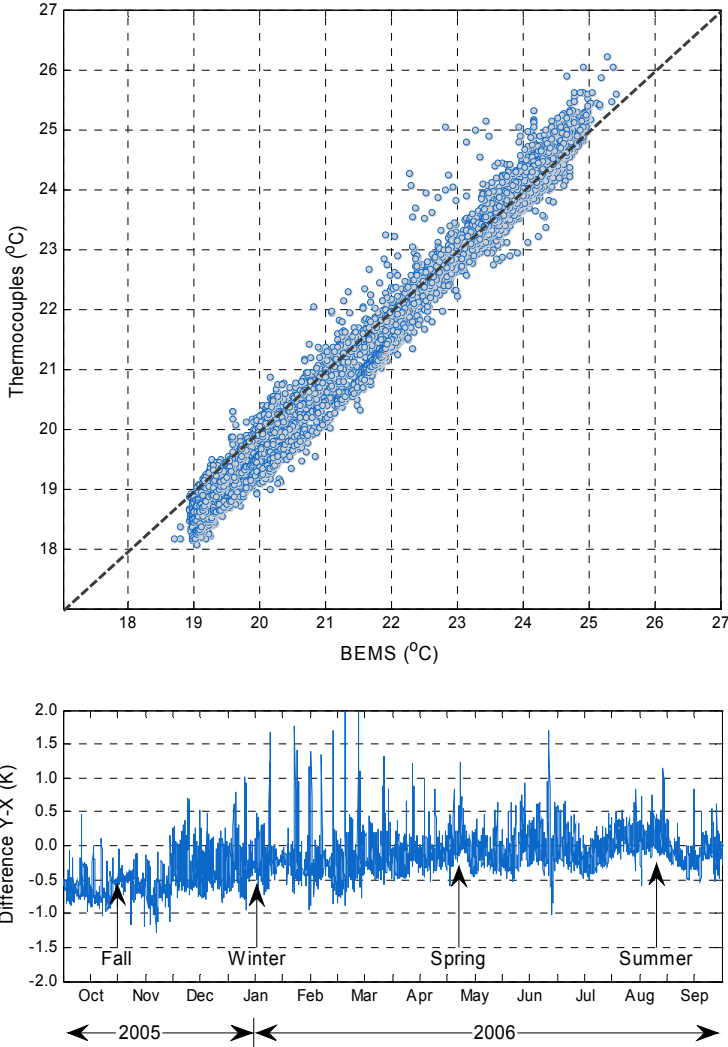


Figure 6-10 Room EC2: At the top, the air temperature measured by the thermocouples (Y-axis) plotted against the temperature registered by the BEMS (X-axis). Below, the temperature difference between the two values (Y-X). The periods chosen for further analysis are also marked

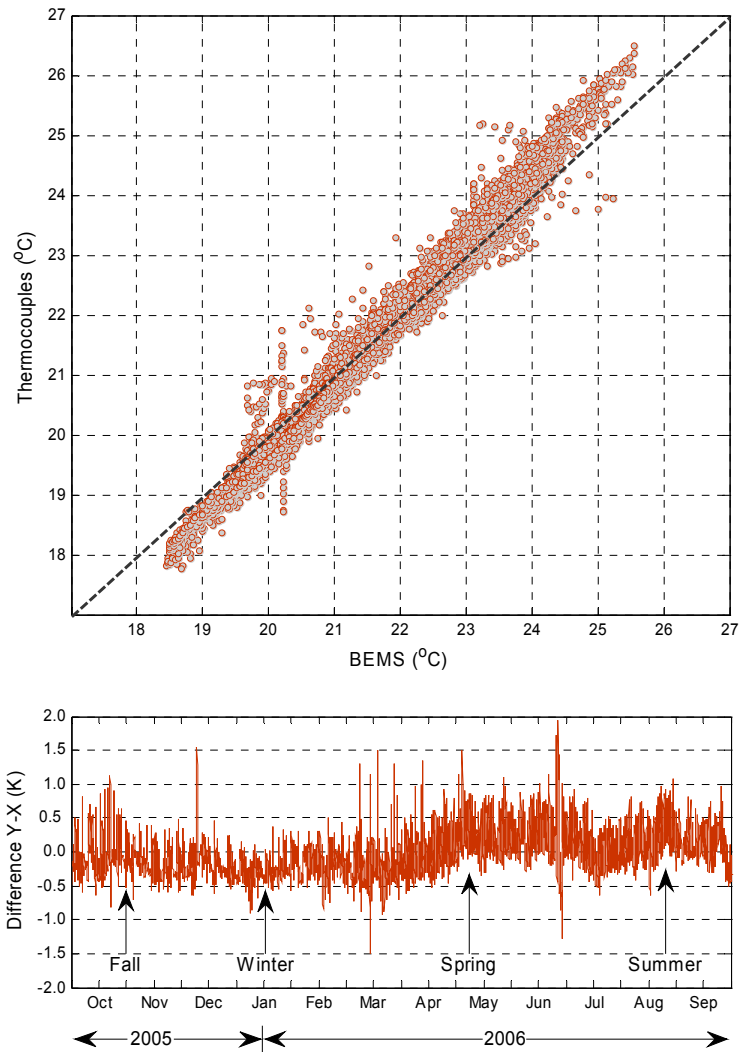


Figure 6-11 Room SC2: At the top, the air temperature measured by the thermocouples (Y-axis) plotted against the temperature registered by the BEMS (X-axis). Below, the temperature difference between the two values (Y-X). The periods chosen for further analysis are also marked

Ideally, all points in the scatter plots in Figure 6-10 and Figure 6-11 should be at the diagonal dashed line since the same parameter is measured. However, both figures show that the temperature recorded by the BEMS is slightly higher for low temperatures than the thermocouples, and conversely, lower for high temperatures. This may be due to the placement of the sensors. The thermocouples are placed just a metre from the window and may be exposed to warm and cold radiation from the window surface. The sensors may also have been exposed to direct solar radiation, which may explain the positive peaks in the

temperature difference plots. Negative peaks may have occurred when the windows were opened. In addition, the thermocouples are close to the workplace and may be influenced by heat emitted from equipment and the person at the desk. The BEMS temperature sensor is placed at the back wall in the room and is less exposed to these factors. The thermocouples are also more responsive than the BEMS temperature sensor, thus temperature changes are recorded earlier by the thermocouples. And lastly, the recorded temperatures from the thermocouples are averages within the interval, as opposed to the BEMS recordings, which are instantaneous values every quarter of the hour. This means that occasional swift temperature variations may have been registered by the BEMS sensor, and smoothed out for the averaged thermocouple registrations.

The occupancy data also have some degree of uncertainty. As mentioned, room absence was registered when no movement was detected for five minutes. In the post-processing of the data, this delay was subtracted from each time the room was registered as empty. This implies that absences shorter than five minutes were not registered, and affects the results in the direction of a higher registered occupancy than the actual occupancy. There may also have been false detections caused by non-human activity. On the other hand, false registrations may have occurred when a person was present but not detected because the movements were too small. This will result in a lower registered occupancy than the actual occupancy. In Appendix A3.2 the mean recorded occupancy with and without delay for the two rooms are shown.

The ventilation airflow rate is perhaps the element of highest uncertainty. The airflow rate is very sensitive to pressure variations caused by varying wind conditions. However, the pressure loss in the culvert between the supply ducts to two rooms is neglectable, thus the same damper opening will approximately supply the same rate of air to the rooms. Therefore, the damper positions rather than the absolute airflow is considered, since it is the difference between the rooms that is of interest.

6.3 Measurements

6.3.1 Criteria for data evaluation

To be able to compare the performance of *Room EC2* and *Room SC2* it was crucial that both rooms were approximately similarly used for periods of at least three days in a row in order to take the effect of heat storage into account. This is of great importance since the set-point for heating and ventilation is dependent on whether people are present or not. In addition, heat gains from lighting and people will also influence the room temperature considerably. During the entire year there were only 10 periods that more or less fulfilled these criteria.

6.3.2 Measurement results

For every period, the *EC2* and *SC2* temperatures are plotted for comparison. However, since the HVAC-system always will try to even out any deviation from the set-point, the damper positions, which directly affect the air flows, are also plotted. In addition, the set-points and the ventilation rates are (or may be) dependent on the presence of people. Therefore, the rather complex picture needs four plots for comparison – air temperature, damper position, actual occupancy and the recorded occupancy added a 20 minute delay.

Figure 6-12 to Figure 6-15 show the measurement results from the periods in each season that met the criteria for evaluation.

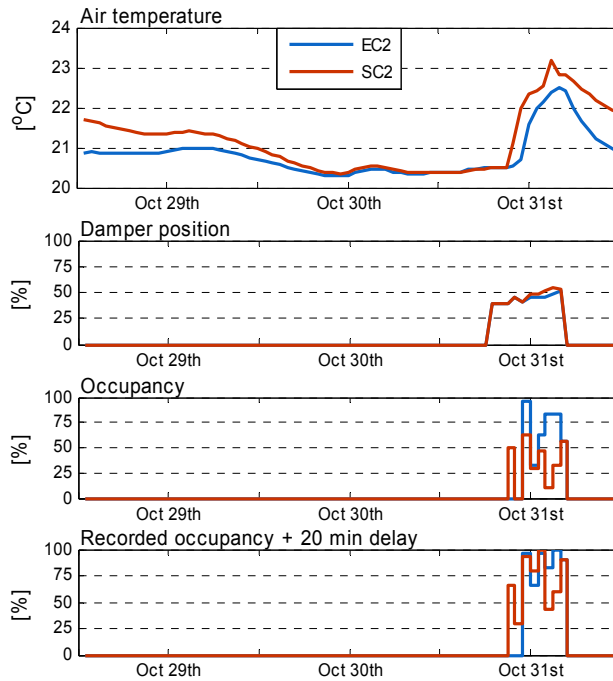


Figure 6-12 Autumn: Saturday October 29th to Monday October 31st 2005

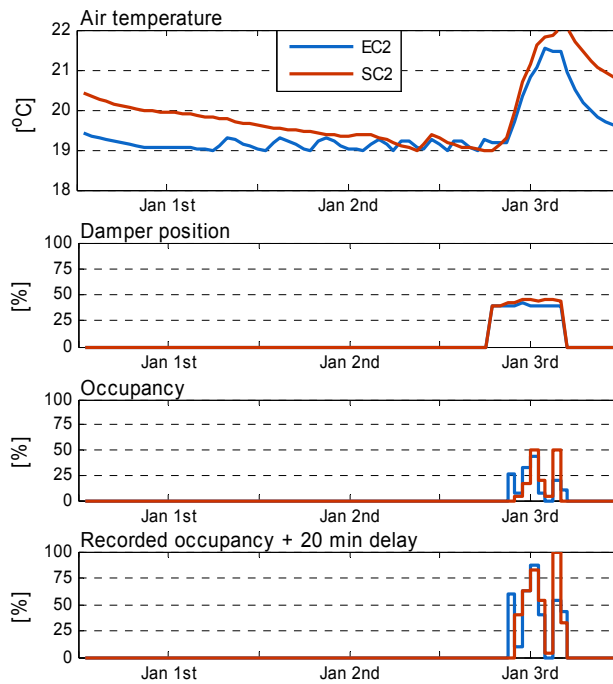


Figure 6-13 Winter: Saturday December 31st 2005 to Monday January 2nd 2006

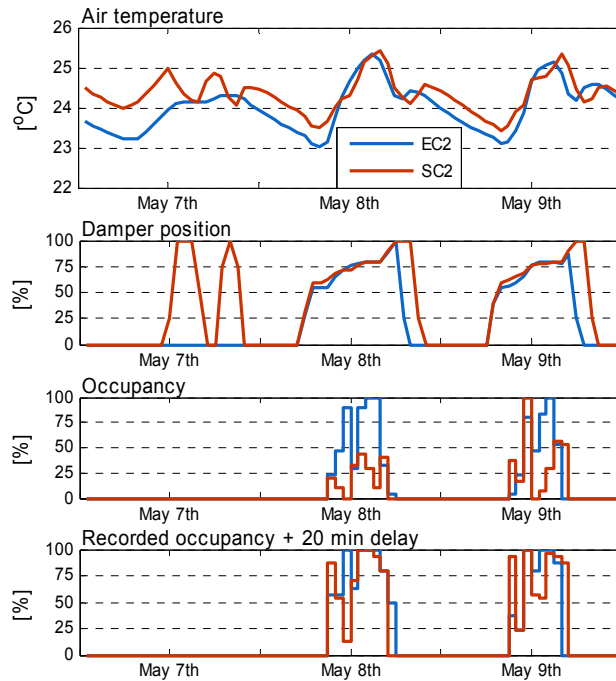


Figure 6-14 Spring: Sunday May 7th to Tuesday May 9th 2006.

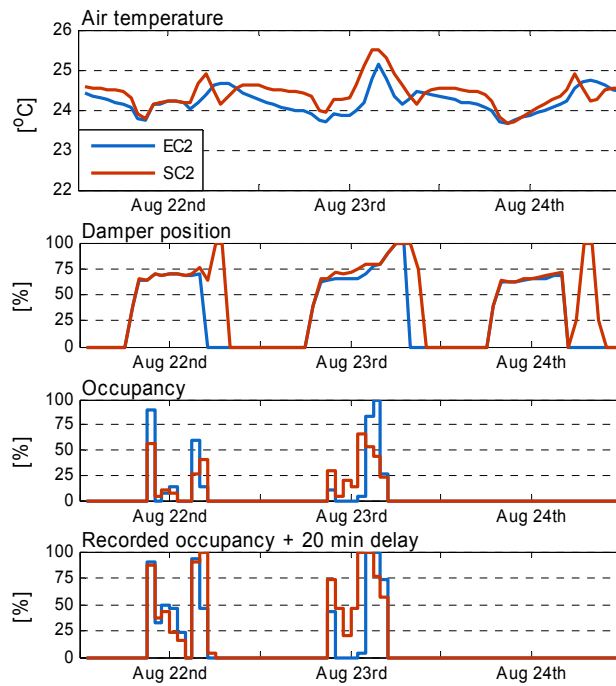


Figure 6-15 Summer: Tuesday August 22nd to Thursday August 24th 2006

Table 6-1 Summary of the key parameters from the 3-day periods given in Figure 6-12 to Figure 6-15

Season	Mean temp (°C)		Max temp (°C)		Total air flow (m ³)		Occupancy 0/20 (h)	
	<i>EC2</i>	<i>SC2</i>	<i>EC2</i>	<i>SC2</i>	<i>EC2</i>	<i>SC2</i>	<i>EC2</i>	<i>SC2</i>
Autumn	20.8	21.2	22.5	23.2	442	530	4.2/5.3	2.9/5.6
Winter	19.4	20.0	21.5	22.1	376	383	6.6/10.2	4.7/11.2
Spring	24.0	24.3	25.3	25.4	3208	5630	9.1/12.3	4.9/12.0
Summer	23.9	24.1	25.1	25.5	4158	5868	4.1/7.0	4.0/9.2

The radiators were operating only during the 3-day winter period, and required 3.5 kWh and 2.6 kWh for *EC2* and *SC2*, respectively.

The following observations can be made from the measurement results:

- Neither the mean temperatures nor the maximum temperatures are significantly different for the two rooms.
- To obtain the thermal conditions, Room *SC2* requires considerably higher air volume flow rate, especially during the summer and spring periods.
- Room *EC2* is according to the occupancy data used more intensively than Room *SC2*. The ratio between occupancy without delay and with a 20 minute delay is higher for Room *EC2*, which indicates that the room is used more continuously than Room *SC2*.

6.4 Discussion of the results

At first sight, the measurement results indicate that *Room EC2* performs best with respect to cooling demand. However, there are some factors that must be taken into consideration before crediting the exposed concrete ceiling. Although emphasis had been on making the conditions in the studied office rooms as similar as practically possible, by removing the suspended ceiling, the room volume increased about 4 m³ and the external wall area increased about 1.2 m². In addition, the suspended ceiling is made of a 22 mm plaster board and a 50 mm mineral wool board, which provided extra insulation. This may have had an influence on the results, and is also supported by the measurements of the radiator use during the 3-day winter period. The radiator uses about 35 % more heating energy during this period.

The presence of people does not only affect the indoor temperature by virtue of internal heat load, but also ventilation, lighting and room temperature set-points. In addition, the HVAC system will always try to equalize any deviation from the room temperature set-point, thus a quantitative evaluation of the performance of the office rooms under normal operation directly from the measurements has been difficult.

6.5 Closing remarks

The many parameters affecting the indoor temperature have made it difficult to use the measurements directly to address and quantify the apparent achievements of exposing the concrete in the ceiling to the indoor air. Hence, to be able to compare the performance of the rooms on an equal basis, a simulation model had to be used. The modelling, verification and calibration of the whole-building simulation model, and the simulation results can be found in the paper '*The effect of suspended ceilings on energy performance and thermal comfort*', which is enclosed in Appendix A4.

PART IV:

SUMMARY AND CONCLUSIONS

Part IV summarizes and synthesizes the findings from all parts and papers in this thesis. It discusses them based on principles and makes some final conclusions and suggestions for further study.

7. SUMMARY AND CONCLUSIONS

7.1 Summary of Part I

From the literature, strong evidence points towards thermal mass together with night ventilation can reduce the indoor maximum operative temperature, reduce the cooling energy demand, and offset the peak cooling demand. All studies reviewed, both the experimental and the analytical, support this. Thermal mass combined with night ventilation may reduce the maximum indoor temperature by 2-6 K, provided there is adequate diurnal outdoor temperature swing, preferably more than 10 K. To get the desired effect, the night temperature should also fall below 20 °C.

Dependent on the climate and building type, the cooling energy savings found in the literature span from 5 % to 36 %. Moreover, some studies conclude that if the heat gains are not too excessive in office buildings, thermal mass and night ventilation should be sufficient to cover the cooling demand alone in moderate climates. This latter point is an especially interesting finding from a Norwegian point of view, since the new building code regulations state that local space cooling should be avoided.

Studies found in the literature reviewed also conclude that thermally heavy buildings have lower space heating energy demand than light buildings. It is claimed that heavy residential and office buildings demand about 15 % and 20 % less heating energy respectively, compared to equivalent light buildings.

7.2 Summary of Part II

In the parametric study, the objective was to assess the efficiency of building thermal mass in a Norwegian context, i.e. Norwegian building code and climate, and to test some of the findings from the literature study. A simulation model of a single office room was implemented in ESP-r with properties according to the new building regulations. A thermally light, medium and heavy version of a single office cell was tested for different climates, occupancy patterns, internal heat loads, set-points for cooling and control ranges, and heat transfer correlations.

The results from the parametric study indicate that the climates similar to those evaluated here have a large potential to cool buildings outside working hours, and that night-time free cooling combined with thermal mass keep the operative temperature well within the limits of thermal comfort during working hours. The peak operative temperatures are also significantly decreased for the *medium* and *heavy* rooms compared to the *light* room. This is proven to be the case for the occupancy patterns, internal heat loads and climates similar to those evaluated here.

The parametric study indicates that thermal mass combined with night-time ventilation make the rooms capable of handling considerable higher daytime internal heat loads than light rooms. Provided that there is sufficient night ventilation and thermal capacity equivalent to the *medium* room or more, the temperatures in a single office room with normal use will stay well within the limits of comfort most of the year without the use of space cooling.

The energy assessments done in this study have been related to the cooling of ventilation air, fan power, space heating, and preheating of ventilation air. The simulations show that thermal mass improves the energy efficiency. In this study it is found that the *heavy* room demands about 10-20 % less total fan power and net cooling energy than the *light* room. Moreover, the *medium* room has almost the same potential for saving fan power and cooling energy as the *heavy* room. However, only minor differences in heating energy demand with respect to thermal mass were found.

7.3 Summary of Part III and Paper V

The experimental field study aimed at using measurements to determine potential energy savings and thermal environmental benefits of exposing the concrete ceiling to the indoor air as an alternative to the original suspended ceiling in a real building in operation. Although emphasis was on making the conditions in the studied office rooms as similar as practically possible, removal of the suspended ceiling enlarged both the room volume and the area exposed to the exterior. This may have had influence on the measurement results obtained in the study.

The number of different parameters affecting the indoor temperature made it difficult to use the measurements directly to address and quantify the apparent achievements of exposing

the concrete in the ceiling to the indoor air. To ensure full control of all parameters affecting the indoor air temperature and to quantify the benefits the measurements indicated, a whole-building simulation model was implemented in ESP-r. The measurements were used to calibrate and validate the simulation model.

The results from the measurements and simulations showed that exposed concrete in the ceiling reduces the number of hours with excessive temperatures considerably and also created a better and more stable thermal environment during the working day. Also, exposed concrete increased the achievements of utilizing night free cooling significantly. However, by exposing concrete in the ceiling, only minor annual heating energy savings were achieved.

7.4 Discussion of principles and transferability

Focus in the thesis has been on the Norwegian climate. However, the coastal climate in Norway with relative mild winters and cool summers has more in common with the moderate climates of Northern Europe and especially the British Isles than other inland countries at same latitude as Norway. Still, Norway is a country situated far north, and during winter time when the days are short and the sun is low in the horizon, the heat contribution from the sun is limited. Countries farther south may to a greater extent utilize solar gains to reduce the heating demand, thus thermal mass may be more important in relation to heating demand reduction.

The trend in office buildings today is towards extended use of open plan offices, or a mix of office rooms and open solutions. The question is whether the results obtained in this thesis are transferable to other room types. Obviously, the thermal capacity equivalent to the heavy office room is not practically possible to achieve without heavy mass partition walls. On the other hand, due to acoustics and increased demand for flexibility in office buildings, the heavy version of the office room presented in the parametric study is unrealistic in practice. However, as shown in Part II and Paper V, exposed concrete in the ceiling produces a significant thermal mass effect. This is also in accordance with findings in the literature, which state that the greatest effect is achieved when going from a light construction to a semi-heavy construction and that the effect is diminished by a further increase of thermal mass.

Since local space cooling should be avoided according to the new Norwegian building energy regulations, the focus in this thesis has been mainly concerned with means to avoid cooling and finding the limits for which cooling could be avoided. However, modern buildings have highly insulated envelopes, and persons, lighting and equipment emit considerable amounts of heat, which in some way or other must be removed from the space to avoid excessive temperatures. Consequently, the excessive heat must be removed from the building with the ventilation air. This is also the reason why this thesis mainly focuses on demand control ventilation (DCV), i.e. ventilation which meets the required supply airflow rate based on occupancy, contaminants, and/or temperature. Although most buildings in Norway have constant air volume (CAV) systems, more and more new buildings are being equipped with variable air volume (VAV) or DCV systems. For CAV systems, the results obtained here should be transferable regarding the thermal conditions. However, the fan energy and net cooling demands will be independent of the thermal mass amount, since the ventilation is solely time controlled. On the other hand, if the temperature is size dimensioning for the ventilation system, thermal mass may contribute to reducing investment cost, and oversized installations may be avoided.

Studies found in the literature, the simulations done in the parametric study, measurements carried out in the field study and the simulations with the whole-building simulation model show that thermal mass improves the energy efficiency in buildings, thus decreasing the operation costs. In addition, the thesis shows that the number of hours with high operative temperature is decreased and that the temperature is steadier during the working hours. In office buildings, this constitutes a great economic savings potential due to improved performance and working efficiency. This fact should provide an extra motivation for building owners, thus making utilization of thermal mass contributory to an improved environment – both indoors and outdoors.

7.5 Main conclusions

Within the limitations of this thesis and based on the findings from all parts and papers in this thesis, the following conclusions can be drawn:

- Thermally heavy rooms may reduce the daytime peak temperature, reduce the diurnal temperature swing, decrease the number of hours with excessive temperatures, and increase the ability of a space to handle daytime heat loads. Depending on the ventilation airflow rate, occupancy pattern, and prevailing convection regime, compared to a light room, a heavy room may;
 - reduce the daytime peak temperature by 1-2 K,
 - decrease the diurnal temperature variation by 2-3 K, and
 - increase the ability to handle daytime internal heat loads by 50-120 %.

- Thermal mass may contribute to decrease the net cooling demand in buildings. Dependent on set-point for ventilative cooling, control range, occupancy pattern, and airflow rate, compared to a light room, a heavy room may;
 - reduce the ventilation net pre-cooling and fan power demand by 10-20 %, or
 - eliminate the need for space cooling also for low ventilation airflow rates during working hours.

- Thermal mass is found to have only a minor influence on the total heating demand in office buildings. Dependent on the daytime internal heat load, occupancy pattern, and climate, a thermally heavy room compared to a light room may;
 - decrease the total space heating and ventilation pre-heating demand by less than 3-7 % on annual basis, but
 - reduce the energy savings potential of night temperature set-back with 3-4 %.

- As a consequence of the findings listed above, thermal mass *contributes* to;
 - fulfil the net energy frame for the building category in question,
 - eliminate the need for space cooling in new office buildings, and
 - improve the thermal climate and thereby increase working performance in office buildings without the need for energy intensive and expensive technical installations.

7.6 Suggestions for further work

- The main focus in this thesis has been on office buildings. Further studies could be done to investigate whether thermal mass is beneficial also for other building categories. The new Norwegian building regulations include most building categories, and a thermal mass assessment should be performed for the most relevant non-industrial building types with emphasis on the typical user characteristics for the building type in question.
- As discussed, barriers for utilization of thermal mass are loss of flexibility and acoustical challenges. Phase changing materials (PCM) is a promising technique, and can be included in building fabrics or even in mobile partitioning walls to increase the thermal capacity and still maintain the flexibility of the building. The efficiency of using PCM as “assisting” thermal mass should be an interesting continuance of this study.
- Another area is to investigate the potential for including weather forecast control in relation to the thermal response of buildings for optimal operation.
- In the parametric study, the influence of different convection correlations was assessed. Further studies could be done in order to find solutions for supply air diffusers for optimal utilization of heavy mass constructions – in particularly exposed concrete ceilings.
- A final suggestion is to investigate the combination of DCV systems and heavy building structures to develop recommendations for modern buildings. This could combine sub-hourly building and short time-step HVAC plant simulations.

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APPENDIX 1

A1.1 Diurnal heat capacity (DHC)

The *Diurnal Heat Capacity (DHC)* is a measure of the capacity of a building to absorb heat from the interior space, and to release the heat back to the space during a diurnal cycle. The *dhc* of a surface is given by (Balcomb, 1984):

$$dhc = F_1 s \quad (\text{Wh}/(\text{m}^2 \cdot \text{K})), \quad (\text{A1.1})$$

where

$$s = \sqrt{P\lambda\rho c / 2\pi} \quad (\text{Wh}/(\text{m}^2 \cdot \text{K})) \quad (\text{A1.2})$$

$$F_1 = \sqrt{(\cosh 2x - \cos 2x) / (\cosh 2x + \cos 2x)} \quad (-) \quad (\text{A1.3})$$

$$x = d\sqrt{\pi\rho c / P\lambda} \quad (\text{m}^2), \quad (\text{A1.4})$$

and

P = period (24h)

ρ = density (kg/m^3)

c = heat capacity ($\text{J}/(\text{kg} \cdot \text{K})$)

d = material thickness (m)

λ = conductivity ($\text{W}/(\text{m} \cdot \text{K})$)

The total *DHC* of a building is calculated by summing the *dhc*-values of each surface n exposed to the interior air, thus:

$$DHC = \sum_n dhc_i A_i \quad (\text{Wh}/\text{m}^2) \quad (\text{A1.5})$$

In Figure 2-3 in Section 2.2.2, the *DHC* for some selected building materials is plotted as a function of thickness.

A1.2 Thermal Time Constant (TTC)

The *Thermal Time Constant (TTC)* is defined as the product of the thermal resistance (R) and heat capacity (C) of a unit area of a building envelope element (Givoni, 1998). To calculate the TTC_A of an area, the heat capacity per unit area (C_A) is multiplied by the thermal resistance ($R_i = d_i/\lambda_i$) to the center of the layer in question, thus

$$C_{A_i} R_i = \left(R_0 + R_1 + \dots + \frac{1}{2} R_i \right) (d \rho c)_i \quad (\text{s}) \quad (\text{A1.6})$$

where R_0 is the external convective resistance. For a multi-layer construction comprising n layers:

$$TTC_A = C_{A1} R_1 + C_{A2} R_2 + \dots + C_{An} R_n \quad (\text{s}) \quad (\text{A1.7})$$

The TTC_s for each surface is the product of the TTC_A multiplied by its area:

$$TTC_s = A_s \cdot TTC_A \quad (\text{s}) \quad (\text{A1.8})$$

Glazed areas are assumed to have a TTC of 0. The total TTC_{tot} of the building envelope equals the sum of all TTC_s divided by the total envelope area, thus

$$TTC_{tot} = \frac{\sum TTC_s}{A_{tot}} \quad (\text{A1.9})$$

A1.3 Reverberation time

The reverberation time (t_r) is defined as the time it takes for a sound to decay by 60 dB. The choice of the relative intensity to use is of course arbitrary, but there is a good rationale for using 60 dB since the loudest crescendo for most orchestral music is about 100 dB and a typical room background level for a good music-making area is about 40 dB. Thus the standard reverberation time is seen to be about the time for the loudest crescendo of the orchestra to die away to the level of the room background.

The reverberation time can be calculated approximately from the Sabine formula (Smith et al., 1996):

$$t_r = 0.16 \frac{V}{A} \text{ (s)}, \quad (\text{A1.10})$$

where

$$A = \sum_n \alpha_i S_i \text{ (m}^2 \text{ sabin)} \quad (\text{A1.11})$$

V = room volume (m^3)

α_i = absorption coefficient of surface i (-)

S_i = area of surface i (m^2)

Typical values for the sound absorption coefficients for common building surfaces are given in Table A1-1 (adapted from Hall, 1991).

Appendix

Table A1-1 Typical values for the absorption coefficient (Hall, 1991)

Material	Sound absorption coefficient α_i at frequency (Hz)					
	125	250	500	1000	2000	4000
Acoustic tile, rigid mount	0.20	0.40	0.70	0.80	0.60	0.40
Acoustic tile, suspended	0.50	0.70	0.60	0.70	0.70	0.50
Acoustical plaster	0.10	0.20	0.50	0.60	0.70	0.70
Ordinary plaster on lath	0.20	0.15	0.10	0.05	0.04	0.05
Gypsum wallboard (22 mm) on studs	0.30	0.10	0.05	0.04	0.07	0.10
Plywood sheet (12 mm) on studs	0.60	0.30	0.10	0.10	0.10	0.10
Concrete block, unpainted	0.40	0.40	0.30	0.40	0.30	0.30
Concrete block, painted	0.10	0.05	0.06	0.09	0.10	0.10
Concrete, poured	0.01	0.01	0.02	0.02	0.02	0.03
Brick	0.03	0.03	0.03	0.03	0.03	0.02
Vinyl tile on concrete	0.02	0.03	0.03	0.03	0.03	0.02
Heavy carpet on concrete	0.02	0.06	0.04	0.40	0.60	0.60
Heavy carpet on felt backing	0.10	0.30	0.40	0.50	0.60	0.70
Platform floor, wooden	0.40	0.30	0.20	0.20	0.15	0.10
Ordinary window glass	0.30	0.20	0.20	0.10	0.07	0.04
Heavy plate glass	0.20	0.06	0.04	0.03	0.02	0.02
Draperies, medium velour	0.07	0.30	0.50	0.70	0.70	0.60
Upholstered seating, unoccupied	0.20	0.40	0.60	0.70	0.60	0.60
Upholstered seating, occupied	0.40	0.60	0.80	0.90	0.90	0.90
Wood seating, unoccupied	0.02	0.02	0.06	0.06	0.06	0.05
Wooden pews, occupied	0.40	0.40	0.70	0.70	0.80	0.70

APPENDIX 2

A2.1 Heat capacity and effusivity for multi-layered constructions

In order to compare the degree of thermal mass of the three office rooms in the parametric study, the heat capacities and the mean thermal effusivities for the three office rooms were considered. The methods employed to calculate the values listed in Table 5-2 are listed below.

Since the values only are used for comparison, the simplified method described in Annex A of the NS-EN ISO 13786 (2000) is used to find the heat capacities. The method defines the effective thickness of a component, d , as the minimum of

1. half the thickness of the component;
2. the thickness of materials between the surface of interest and the first thermal insulating layer;
3. or, a maximum effective thickness, depending on the period of variations; 1 hour 1 day and 1 week, applies to a thickness of 2 cm, 10 cm, and 25 cm, respectively

The heat capacity by surface area, C_T , is calculated by summing each layer with the construction thicknesses restricted by the list above considering 1 day variation. Thus,

$$C_T = \sum_n \rho_i d_i c_i \quad (\text{A2.1})$$

Using the values from Table 2-1 in Section 2.1.1, Table A2-1 shows the calculated total heat capacities for surfaces in the office rooms.

Table A2-1 Total heat capacities per surface area for the office rooms distributed on building parts

Office room	Total heat capacity (kJ/(m ² ·K))			
	Internal walls	External wall	Floor	Ceiling
Light	517	64	155	174
Medium	517	64	1635	1680
Heavy	2495	346	1635	1680

Table 5.2 in Section 5.2 gives the total heat capacities per floor area, found by multiplying the values in the table above with the respective surface areas, summarized and divided by the floor area of the rooms.

To find the equivalent thermal effusivity, following relation is used to reduce the multiple layer construction to one equivalent homogeneous layer (Clarke, 2001):

$$(\lambda\rho C)_e = \frac{1.1R_i(\lambda\rho C)_i + \sum_m 1.1R_m(\lambda\rho C)_m}{R_e} + \frac{(\lambda\rho C)_o}{R_e} \left(R_o - 0.1R_i - \sum_m 0.1R_m \right) \quad (\text{A2.2})$$

Where R is thermal resistance, and subscripts e , i , m and o refer to the equivalent, inner, intermediate, and outer layers of the constructions, respectively. The equivalent thermal resistance is given by

$$R_e = \frac{d_o}{\lambda_o} + \sum_m \frac{d_m}{\lambda_m} + \frac{d_i}{\lambda_i} \quad (\text{A2.3})$$

Using values from Table 2-1 in Section 2.1.1, Table A2-2 shows the calculated thermal effusivities for the surfaces in the office rooms.

Table A2-2 Equivalent thermal effusivities per surface area for the office rooms distributed on building parts

Office room	Equivalent thermal effusivity β ($\text{W}\cdot\text{s}^{1/2}/(\text{m}^2\cdot\text{K})$)			
	Internal walls	External wall	Floor	Ceiling
Light	306	302	190	308
Medium	306	302	1481	1495
Heavy	815	756	1481	1495

Table 5.2 in Section 5.2 lists the thermal effusivities per floor area, found by multiplying the values in the table above with the respective surface areas, summarized and divided by the floor area of the rooms.

A2.2 Calculation example using the CCP

Assuming a diurnal cycle in an office room where the heat storage equals the heat release, the mean heat flux during the storage process per room area can be calculated by (Artmann et al., 2007):

$$\dot{q} = \frac{Q_{charge}}{At_o} = \frac{Q_{release}}{At_o} = \frac{\dot{V}c_p}{At_o} CCP \quad (\text{W/m}^2) \quad (\text{A2.4})$$

where

- \dot{q} = mean heat flux (W/m²)
- Q_{charge} = heat absorbed in zone (Wh)
- $Q_{release}$ = heat released during the night (Wh)
- A = floor area (m²)
- t_o = hours of occupation (h)
- \dot{V} = effective volume flow rate (m³/s)
- c_p = specific heat of the air (Wh/(m³·K))
- CCP = climatic cooling potential (Kh)

The effective mass flow rate can be written as $\dot{V} = AHR\eta$, where H is the room height and R the air change rate. η is the temperature efficiency, defined as

$$\eta = \frac{T_{ext} - T_e}{T_b - T_e}, \quad (\text{A2.5})$$

which takes into account that the temperature of the air extracted from the building, T_{ext} , is lower than the building temperature, T_b . T_e is the ambient temperature. Assuming the room height $H = 2.75$ m, the occupancy duration $t_o = 8$ h, and a constant effective air change rate of $R\eta = 3 \text{ h}^{-1}$, the heat flux per degree hour of the CCP can be calculated as:

$$\frac{\dot{q}}{CCP} = \frac{HR\eta c_p}{t_o} = \frac{2.75\text{m} \cdot 3\text{h}^{-1} \cdot 0.33\text{Wh/m}^3\text{K}}{8\text{h}} = \underline{\underline{0.34 \frac{\text{W/m}^2}{\text{Kh}}}} \quad (\text{A2.6})$$

A2.3 Finding the relative humidity from the dew point temperature

The DRY files used in the parametric study in Part II gives the dew point temperature instead of the relative humidity (RH) needed as input in ESP-r. Given the dry bulb temperature (T), dew point temperature (T_d), and pressure (p), the relative humidity (RH) can be found. The RH is given by (adapted from ASHRAE, 2005b)

$$RH = \frac{\mu}{(1 - (1 - \mu)(p_w / p))} \quad (A2.7)$$

where p_w is the partial pressure of water vapour and μ the degree of saturation given by

$$\mu = \frac{W}{W_s} \Big|_{t,p} \quad (A2.8)$$

where W is the humidity ratio

$$W = 0.62198 \frac{p_w}{p - p_w} \quad (A2.9)$$

and

$$W_s = 0.62198 \frac{p_{ws}}{p - p_{ws}} \quad (A2.10)$$

p_w and p_{ws} can be found for saturation pressure over ice by employing (temperatures in Kelvin), respectively

$$\ln p_w = C_1 T_d^{-1} + C_2 + C_3 T_d + C_4 T_d^2 + C_5 T_d^3 + C_6 T_d^4 + C_7 \ln T_d \quad (A2.11)$$

and

$$\ln p_{ws} = C_1 T^{-1} + C_2 + C_3 T + C_4 T^2 + C_5 T^3 + C_6 T^4 + C_7 \ln T \quad (A2.12)$$

For saturation pressure over water

$$\ln p_w = C_8 T_d^{-1} + C_9 + C_{10} T_d + C_{11} T_d^2 + C_{12} T_d^3 + C_{13} \ln T_d \quad (\text{A2.13})$$

and

$$\ln p_{ws} = C_8 T^{-1} + C_9 + C_{10} T + C_{11} T^2 + C_{12} T^3 + C_{13} \ln T \quad (\text{A2.14})$$

where

$C_1 = -5.6745359\text{E}+03$	$C_6 = -9.4840240\text{E}-13$	$C_{11} = 4.1764768\text{E}-05$
$C_2 = 6.3925247\text{E}+00$	$C_7 = 4.1635019\text{E}+00$	$C_{12} = -1.4452093\text{E}-08$
$C_3 = -9.6778430\text{E}-03$	$C_8 = -5.8002206\text{E}+03$	$C_{13} = 6.5459673\text{E}+00$
$C_4 = 6.2215701\text{E}-07$	$C_9 = 1.3914993\text{E}+00$	
$C_5 = 2.0747825\text{E}-09$	$C_{10} = -4.8640239\text{E}-02$	

A2.4 Finding the most critical façade orientation

Finding the most unfortunate façade orientation concerning cooling demand is not a trivial task. A lot of parameters are influential, such as glass area, climate, glass type and coating, time of operation, occupancy pattern, thermal mass. For instance, a very heavy building may experience the highest indoor temperatures long after the peak outdoor temperatures occurred. A thermally light building with large glass areas may experience the peak indoor temperature when the sun is low in the horizon and the solar beam strikes the glass surfaces perpendicularly.

To evaluate the orientation for the single offices in this study, at first, only the solar heat load is considered. Figure A2-1 shows the integrated solar flux entering through the window from June 1st to August 31st. The Oslo DRY climate file is used.

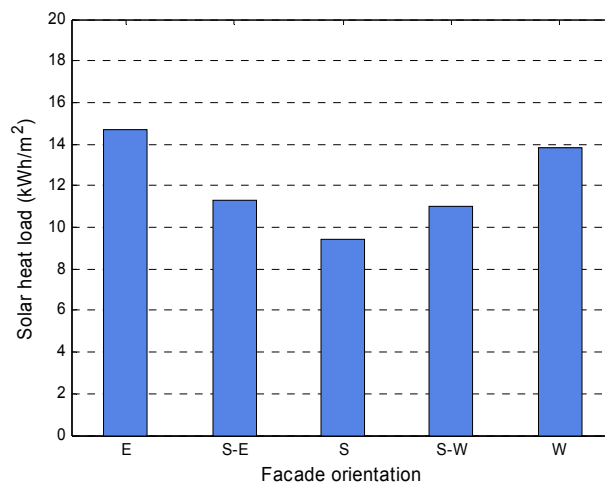


Figure A2-1 Total solar energy entered through the window during for different façade orientations.

Figure A2-1 shows that the east (E) and west (W) oriented façades have the largest integrated solar heat load throughout the period. However, the simultaneity of the solar load, room use and the outdoor temperature must also be taken into account. Therefore, the worst façade orientation is considered from largest peak cooling load and total cooling energy demand point of view.

Following simulations are based on the light version of the office room, and to further minimize thermal mass effects, the set-point for cooling is 21 °C. I.e., there is no dead band between heating and cooling, thus temperature fluctuation and excessive heat storage in the building structures is minimized. The space cooling device is given unlimited cooling capacity and is ideally controlled. Beyond that, the input parameters are similar to the basic model presented in Section 5.2.

Figure A2-2 shows the peak space cooling demand for the office room for different façade orientations, and Table A2-3 at which time the peak occurred.

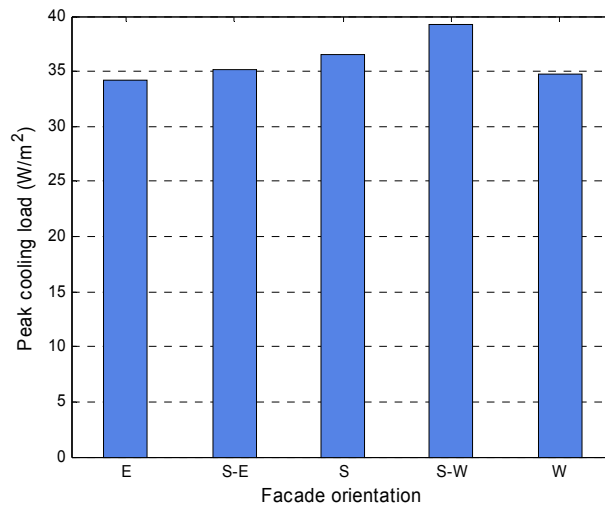


Figure A2-2 Peak cooling demand for the office room for different façade orientations

Table A2-3 Occurrence of the peak cooling demands for the different façade orientations

Occurrence	Orientation				
	E	S-E	S	S-W	W
Date	Aug. 3 rd	Aug. 18 th	Aug. 16 th	Aug. 16 th	Aug. 16 th
Time	09:30	10:55	13:55	15:05	17:35

Figure A2-3 shows the cooling energy demand for the office room. Maybe surprisingly, the orientation that has the least cooling energy demand is the one that faces south (S). The reason for this is that the angle of the incident solar radiation is large, thus a lot of the radiation is reflected. East (E) and west (W) facing facades have the largest demand. The reason for this is partly that the largest cooling loads, due to solar radiation, for these orientations occur early in the morning and afternoon, respectively, thus lose some of the “help” from the ventilation system that operates from 6 am to 6 pm. Since the temperature is kept below 21 °C by the cooling device, the temperature compensation of the supply air in according to Eq. (5.1) will not be invoked. The external sun shading control and set-points also influence the results.

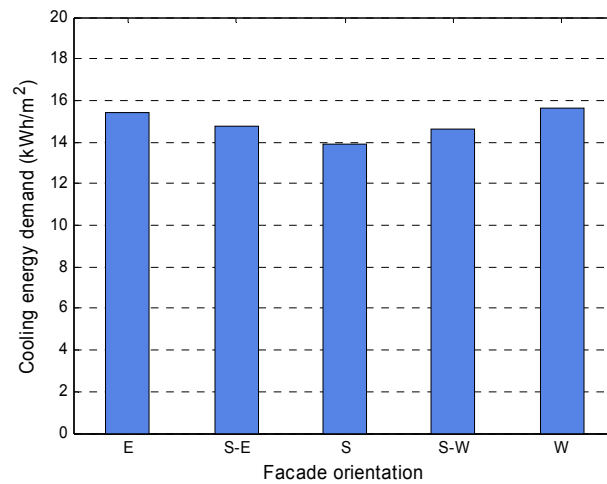


Figure A2-3 Cooling energy demand for the office for different façade orientations

All in all, the most exposed façade is considered to be the south-west (S-W) oriented facade, since the south-west façade has the greatest peak demand and only a slightly lower total cooling energy demand than east and west.

A2.5 Total heating energy savings by temperature set-back

Table A2-4 to Table A2-6 give the absolute and relative heating energy savings by night and weekend temperature set-back for high (10 m³/h per m²) and low (5 m³/h per m²) ventilation air flow rates.

Table A2-4 Temperature set-back energy savings for the light office room for high and low air flow rates

	Space (kWh/m ²)		Ventilation (kWh/m ²)		Tot. relative savings (%)	
	High	Low	High	Low	High	Low
Oslo	2.4	2.4	0.0	-0.1	6.3	8.9
Bergen	2.8	2.7	-0.1	-0.1	7.9	11.2
Andøya	2.8	3.0	0.2	-0.9	6.2	7.0

Table A2-5 Temperature set-back energy savings for the medium office room for high and low air flow rates

	Space (kWh/m ²)		Ventilation (kWh/m ²)		Tot. relative savings (%)	
	High	Low	High	Low	High	Low
Oslo	1.1	1.6	0.4	0.1	4.2	6.8
Bergen	1.4	1.9	0.4	0.0	5.4	8.5
Andøya	1.2	2.0	0.7	-0.6	4.1	4.6

Table A2-6 Temperature set-back energy savings for the heavy office room for high and low air flow rates

	Space (kWh/m ²)		Ventilation (kWh/m ²)		Tot. relative savings (%)	
	High	Low	High	Low	High	Low
Oslo	0.1	0.9	1.3	0.1	3.6	4.1
Bergen	0.2	1.0	0.8	0.3	3.2	5.9
Andøya	0.0	1.1	1.2	-0.4	2.4	2.7

A2.6 Operative temperature employing different occupancy profiles

In Section 5.5, the influence of occupancy patterns were examined. In Figure A2-4 and Figure A2-5, the resultant operative temperatures in the office rooms during the entire warm week are shown for high and low ventilation airflow rates respectively.

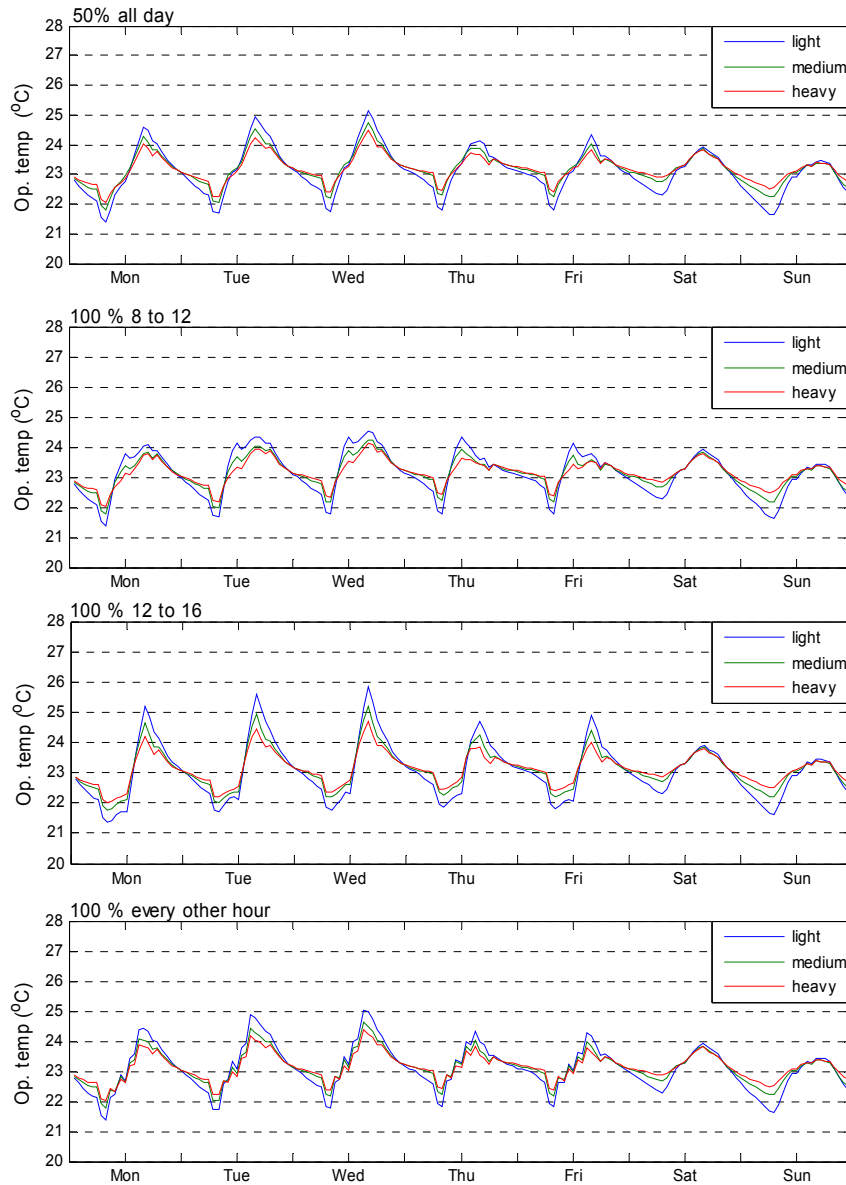


Figure A2-4 Operative temperature for the offices employing the different occupancy profiles with high airflow rate ($10 \text{ m}^3/\text{h per m}^2$)

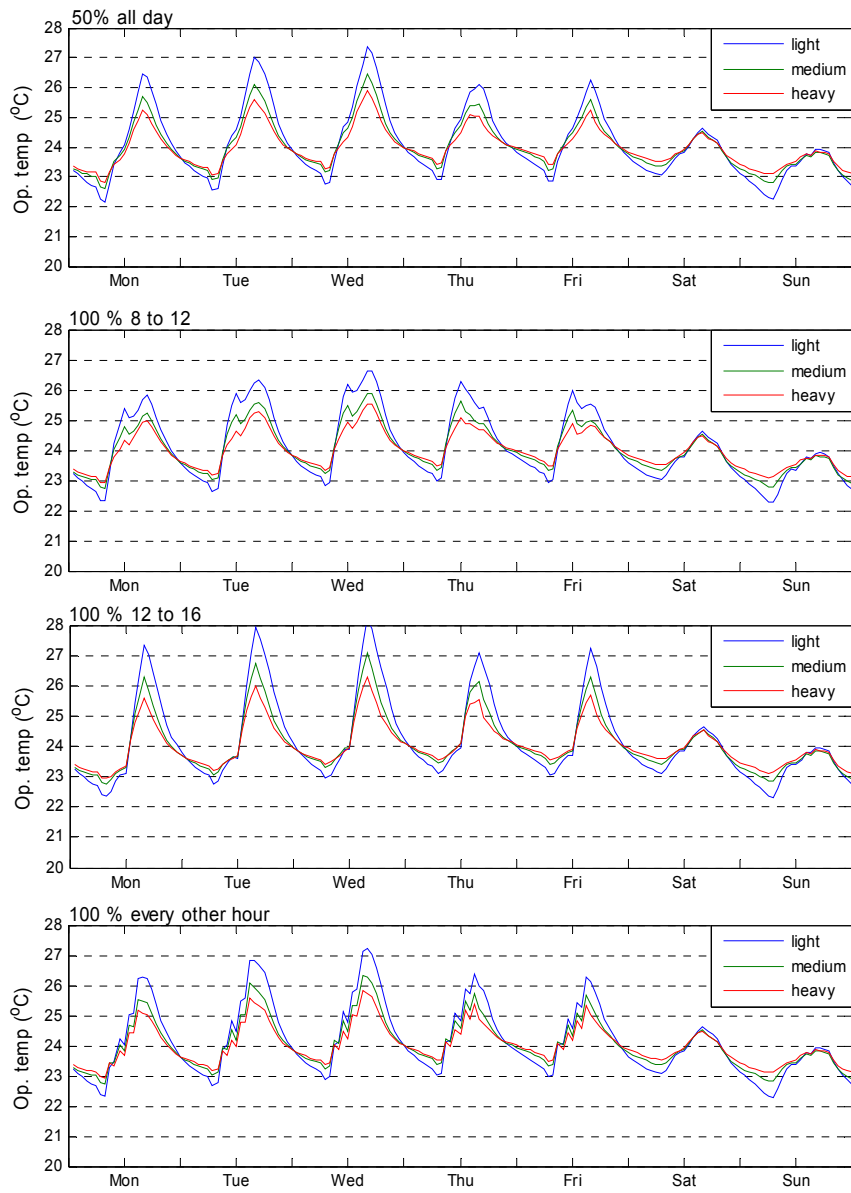


Figure A2-5 Operative temperature for the offices employing the different occupancy profiles with low airflow rate ($5 \text{ m}^3/\text{h per m}^2$)

A2.7 Convection correlations

ESP-r classifies convection regimes into 5 principle classes as listed in Table A2-7 (adapted from Beausoleil-Morrison, 2000).

Table A2-7 The principle convection regimes as classified in ESP-r (Beausoleil-Morrison, 2000)

convection regime	driving force	cause of driving force
A	buoyant	Surface-to-air temperature difference caused by: <ul style="list-style-type: none"> • heat transfer through the building envelope; • solar insolation to walls or floor • in-floor heating • chilled ceiling panels • heated walls
B	buoyant	Heating device (e.g. radiator) located within the zone
C	mechanical	Air handling system delivering supply air to room through ceiling, floor, or wall-mounted diffusers
D	mechanical	Heating or cooling device with circulation fan. No intentional supply or extract of air from room
E	mixed flow (mechanical and buoyant)	Mechanical forces caused by air handling units supplying heated or cooled air to the zone through ceiling, floor, or wall-mounted diffusers. Buoyant forces caused by surface-to-air temperature differences.

For convection regime A, the Alamdari and Hammond (1983) correlations are applied. These correlations are default in ESP-r, and are applied if nothing else is specified;

$$\text{Vertical surfaces: } h_c = \left[\left(1.5 \left(\frac{\Delta T}{H} \right)^{\frac{1}{4}} \right)^6 + \left(1.23 \Delta T^{\frac{1}{3}} \right)^6 \right]^{\frac{1}{6}} \quad (\text{A2.15})$$

$$\text{Horizontal surfaces (buoyant): } h_c = \left[\left(1.4 \left(\frac{\Delta T}{D_h} \right)^{\frac{1}{4}} \right)^6 + 1.63 \left(1.23 \Delta T^{\frac{1}{3}} \right)^6 \right]^{\frac{1}{6}} \quad (\text{A2.16})$$

$$\text{Horizontal surfaces (stably stratified): } h_c = 0.6 \left(\frac{\Delta T}{D_h^2} \right)^{\frac{1}{5}} \quad (\text{A2.17})$$

where

- ΔT = absolute value of the surface to air temperature difference (K)
- H = height of the vertical surfaces (m)
- D_h = hydraulic diameter of horizontal surfaces ($4A/P$)
- A = surface area (m²)
- P = perimeter length (m)

For convection regime B, and radiator placed under window, the Khalifa (1989) correlations are applied to the building surfaces when the radiators are on:

Table A2-8 The Khalifa (1989) correlations for radiator under window

Surface type	Correlation	
walls	$h_c = 2.30 \cdot \Delta T^{0.25}$	(A2.18)
windows	$h_c = 8.07 \cdot \Delta T^{0.11}$	(A2.19)
ceilings	$h_c = 2.72 \cdot \Delta T^{0.13}$	(A2.20)
ΔT is the absolute value of the surface to air temperature difference (K)		

The Fisher and Pedersen (1997) correlations for radial ceiling jets (convection regime C) for different surface types:

Table A2-9 The Fisher and Pedersen correlations for radial ceiling jets

Surface type	Correlation	
Walls	$h_c = 0.19 \cdot ACH^{0.8}$	(A2.21)
Floor	$h_c = 0.13 \cdot ACH^{0.8}$	(A2.22)
Ceiling	$h_c = 0.49 \cdot ACH^{0.8}$	(A2.23)

ACH is the enclosure air change rate within ($3 < ACH < 100$)

Further information on convection correlations and how they are implemented in ESP-r can be found in Beausoleil-Morrison (2000).

APPENDIX 3

A3.1 Calculation of the direct normal radiation

BF3 Sunshine Sensor measures the global radiation, and through a special pattern of shadowing of the glass bulb and built-in algorithms, calculates the diffuse radiation. The input solar parameters in ESP-r are diffuse and direct normal radiation. To find the direct normal radiation from the global and diffuse radiation, following relation is used (Lund, 1985):

$$I_{dn} = \frac{G_h - D_h}{\sin(\alpha)} \quad (\text{A3.1})$$

where

- I_{dn} = direct normal radiation (W/m²)
- G_h = global radiation (W/m²)
- D_h = diffuse radiation (W/m²)
- α = solar altitude, average for the hour over horizon

To avoid excessive values of the direct normal radiation (which may occur for low α), a test is carried out for all hourly averaged values. The test is based upon the air mass m , an “apparent solar constant” $C_{a,sol}$, at ground surface, and an extinction coefficient related to $C_{a,sol}$. The morning minimum temperature T_{min} is used as an indicator for the absolute humidity (Lund, 1985).

The extinction coefficient EXT for the day is given by:

$$EXT = 0.007 \cdot T_{min} + 0.12, \text{ for } T_{min} > -10 \text{ }^\circ\text{C:}$$

$$EXT = 0.05, \text{ for } T_{min} < -10 \text{ }^\circ\text{C}$$

Lund (1985) defines the maximum reasonable direct normal radiation integrated over an hour as:

$$I_{max} = C_{a,sol} \cdot DA \cdot e^{(-EXT \cdot m)} \quad (A3.2)$$

where

$$\begin{aligned} C_{a,sol} &= \text{“apparent solar constant”, } 1163.89 \text{ (Wh/m}^2\text{)} \\ DA &= \text{sun-to-earth distance factor, } 1 + 0.0334 \cos\left(\frac{2\pi}{365} DN\right) \\ DN &= \text{day number (1-365)} \\ m &= \text{relative air mass, } \frac{1.02}{\sin(\alpha) + 0.02} \end{aligned} \quad (A3.3)$$

The computed direct normal radiation from Eq. (A3.1) is compared to I_{max} .

For $I_{dn} > I_{max} + 55.55 \text{ Wh/m}^2$, $I_{dn} = I_{max}$, and diffuse radiation is recalculated:

$$D_h = G_h - I_{dn} \cdot \sin(\alpha) \quad (A3.4)$$

A3.2 Measurement results

Below follow some of the measurement results carried out at Røstad during the period from October 1st 2005 to September 30th 2006;

- Figure A3-1: Temperature measured by the thermocouples and BEMS in room EC2
- Figure A3-2: Temperature measured by the thermocouples and BEMS in room SC2
- Figure A3-3: Averaged occupancy for EC2, holidays included.
- Figure A3-4: Averaged occupancy for EC2, holidays excluded
- Figure A3-5: Averaged occupancy for SC2, holidays included.
- Figure A3-6: Averaged occupancy for SC2, holidays excluded

A3.2.1 Temperature measurements Room EC2

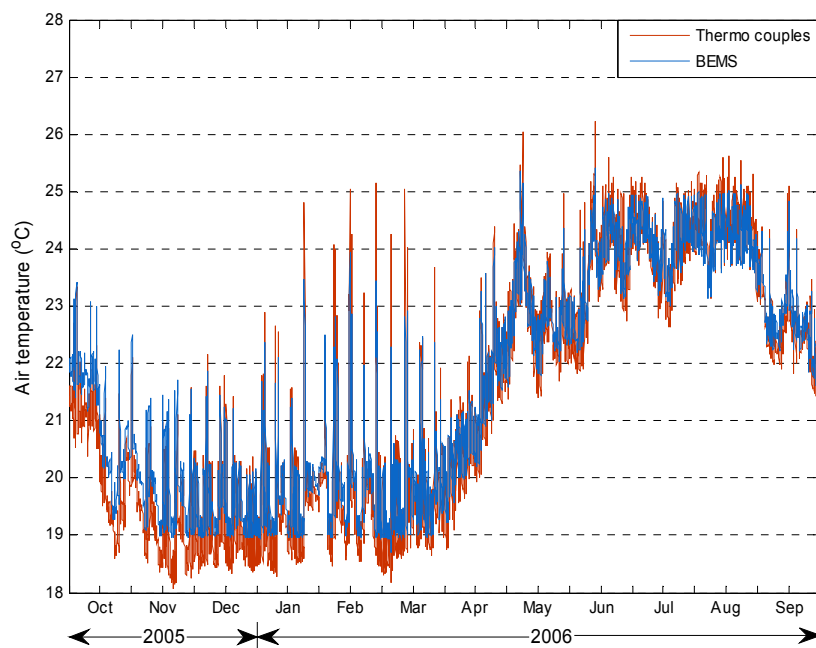


Figure A3-1 The air temperature measured by the additional thermocouples and the air temperature registered by the BEMS for room EC2

A3.2.2 Temperature measurements Room SC2

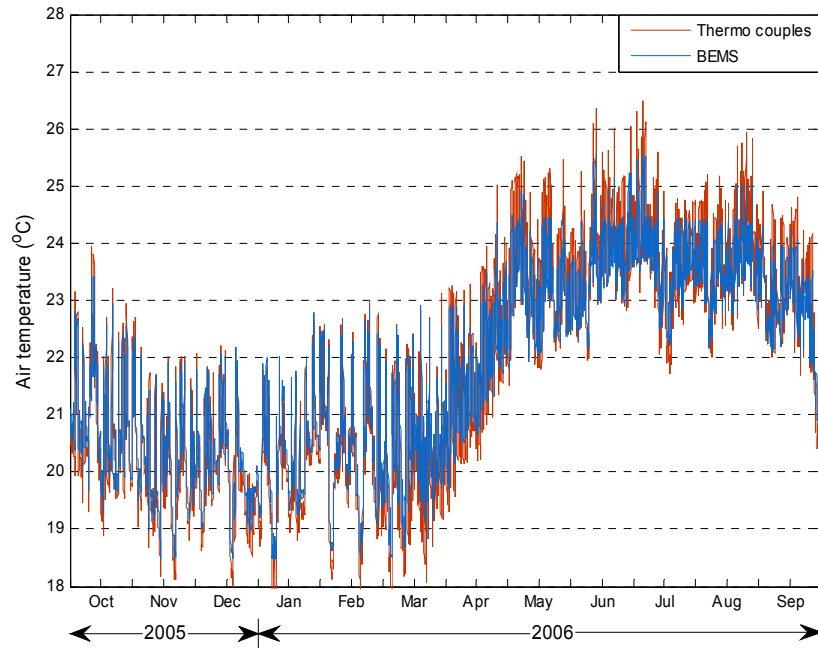


Figure A3-2 The air temperature measured by the additional thermo couples and the air temperature registered by the BEMS for room SC2

A3.2.3 Occupancy recordings room EC2

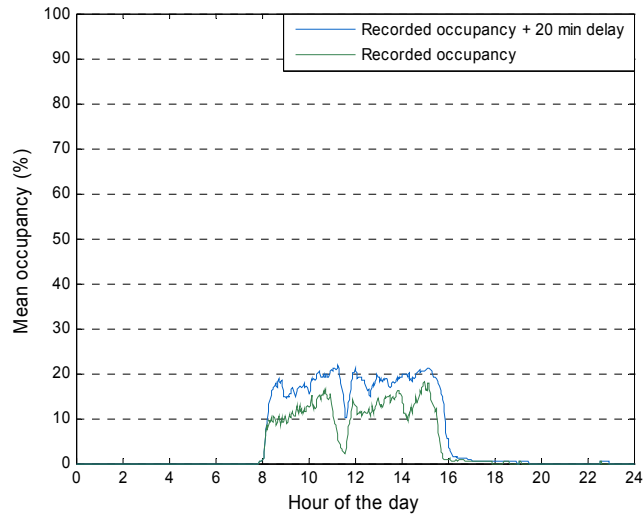


Figure A3-3 Averaged weekday occupancy for Room EC2 – vacations and holidays included

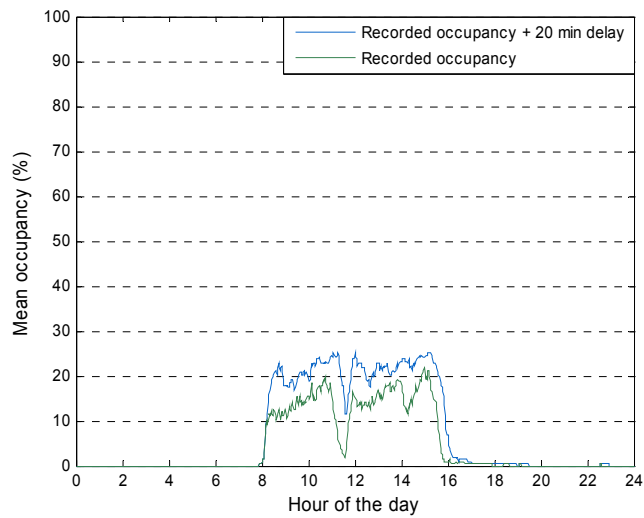


Figure A3-4 Averaged weekday occupancy for Room EC2 – vacations and holidays excluded

A3.2.4 Occupancy recordings room SC2

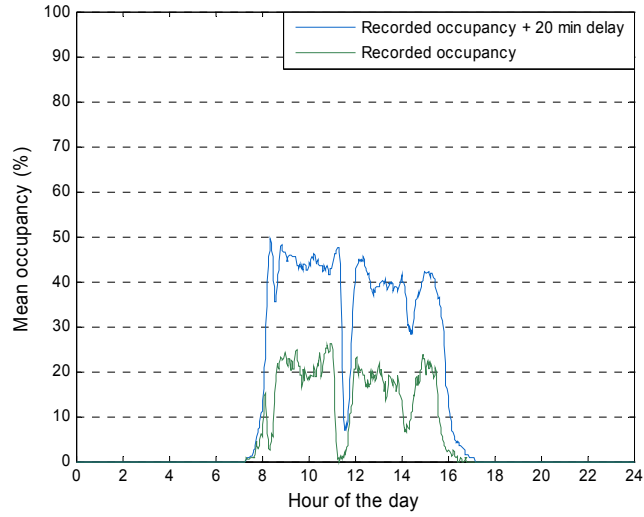


Figure A3-5 Averaged weekday occupancy for Room SC2 – vacations and holidays included.

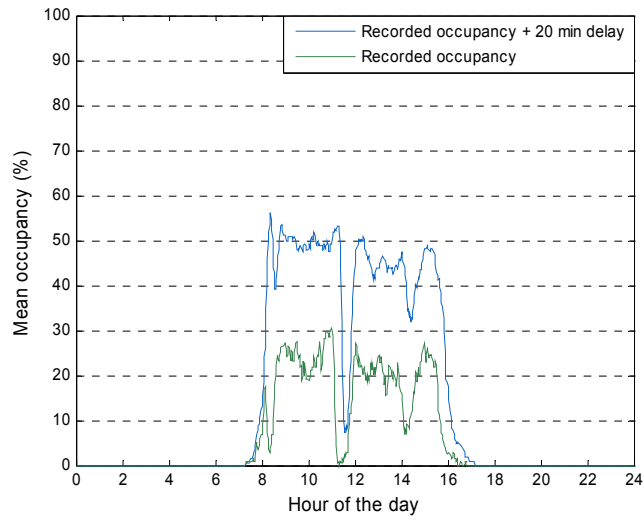


Figure A3-6 Averaged weekday occupancy for Room SC2 – vacations and holidays excluded

APPENDIX 4 – PAPERS

- Paper I Mathisen H.M., Høseggen R., Hanssen S.O. (2005), *Measurement and Simulation of Energy Use in an Office Building with Hybrid Ventilation*, (Presented at the Nordic Symposium on Building Physics, Reykjavik, Iceland, June 2005)
- Paper II Høseggen R., Mathisen H.M., Hanssen S.O. (2005), *Comparison of multi- and single zone modeling for estimation of energy use in an office building*, (Presented at Indoor Air, Beijing, China, September 2005)
- Paper III Høseggen R., Wachenfeldt B.J., Hanssen S.O. (2006), *Reduced energy demand by combining natural and mechanical ventilation in a new office building with an atrium*, (Presented at Cold Climate HVAC, Moscow Russia, May 2006)
- Paper IV *Building Simulation as an assisting tool in decision making Case study: With or without a double-skin façade?* Energy and Buildings (2007), doi:10.1016/j.enbuild.2007.05. (in press).
- Paper V Høseggen R., Mathisen H.M., Hanssen S.O., *The effect of suspended ceilings on energy performance and thermal comfort*, (Submitted for review)

Paper I: Measurement and Simulation of Energy Use in an Office Building with Hybrid Ventilation

Hans M. Mathisen, Rasmus Z. Høseggen and Sten O. Hanssen

Abstract

The objective of this work is to demonstrate the consequences on energy use by adjusting control parameters in a building with hybrid ventilation and displacement air supply. The interest for hybrid ventilation has increased during the last few years. Hybrid ventilation has mostly been used in schools, but near Trondheim in Norway an office building with displacement ventilation was erected during 2002. The building has a culvert embedded in the ground and under the building, and displacement ventilation in the office rooms. For more than a year the building's energy use for heating, ventilation and electric equipment has been monitored. In addition, measurements of room temperatures and the presence of people were carried out. These latter parameters are important in order to calibrate a simulation model. The energy use seemed to differ from what was expected in the design phase. To study why the calculated energy use differed from the measured values, simulations were carried out with an integrated simulation model using ESP-r. The results show that the allowed temperature swing in the room has significant influence on the energy use. The simulations indicate that it is possible to save up to 25 % of the total energy use for heating by adjusting the allowance in temperature swing. In comparison, improving the heat exchanger efficiency by 10 %, the energy savings were 7 to 11 %. The improved energy performance did not significantly affect the thermal comfort.



7th Nordic Building Physics Symposium

Reykjavik, June 13-15 2005

Measurement and Simulation of Energy Use in an Office Building with Hybrid Ventilation

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KEYWORDS: Hybrid ventilation, simulations, measurements, energy use, control strategies.

SUMMARY:

The objective of this work is to demonstrate the consequences on energy use by adjusting control parameters in a building with hybrid ventilation and displacement air supply. The interest for hybrid ventilation has increased during the last years. Hybrid ventilation has mostly been used in schools, but near Trondheim in Norway an office building with displacement ventilation was erected during 2002. The building has a culvert embedded in the ground and under the building, and displacement ventilation in the office rooms. For more than a year the building's energy use for heating, ventilation and electric equipment has been monitored. In addition, measurements of room temperatures and the presence of people were carried out. These latter parameters are important in order to calibrate a simulation model. The energy use seemed to differ from what was expected in the design phase. To study why the calculated energy use differed from the measured values, simulations were carried out with an integrated simulation model using ESP-r. The results show that the allowed temperature swing in the room has significant influence on the energy use. The simulations indicate that it is possible to save up to 25 % of the total energy use for heating by adjusting the allowance in temperature swing. In comparison, improving the heat exchanger efficiency by 10 %, the energy savings were 7-11 %. The improved energy performance did not significantly affect the thermal comfort.

1. Background and objectives

The interest in hybrid ventilation has increased during the last years (Schild, 2003), but has so far mostly been used in schools. However, near Trondheim in Norway an office building with hybrid ventilation was erected during 2002. This provided an opportunity to measure energy use in a building with this type of ventilation and to compare simulations with real data. The simulation model can further be used to study how adjusting of control parameters and changing the system design, can influence on energy use and thermal indoor environment in a hybrid ventilated building.

The objective of the work presented in this paper is to demonstrate the consequences on energy use by adjusting control parameters in a building with hybrid ventilation and displacement air supply.

2. Method

Hourly measurements were carried out during one year from summer 2003 to summer 2004 and included water based energy for space heating and ventilation together with electricity for lighting and equipment. Simulation of measured values was done by modelling the complete building. Validation of the simulations was done by comparing with measurements. The last step was to use the simulation model to evaluate the effect of different changes related to room temperature control, heat recovery and type of ventilation.

3. Description of the building and technical installations

The case building presented in this paper belongs to Nord-Trøndelag College (HiNT) in Levanger, located 80 km north of Trondheim, Norway. The building was ready for occupation in August 2002. It has a common wing with meeting rooms and educational areas. Two other wings are office areas, see Figure 1. To simplify the project, only one of the office wings (the HiNT wing) was used in the study. The building has two storeys, see Figure 2. It has no basement, but a culvert for supply of ventilation air is embedded in the ground along the central axis of the wing. A more detailed description of the building, measurements and LCC calculations can be found in (Mathisen, 2004).

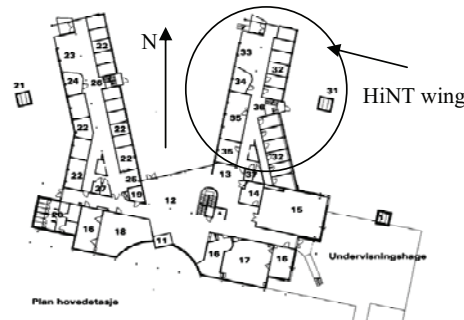


Figure 1. Plan view of the building showing the common area and the two office wings. "31" is the air intake tower



Figure 2. The east façade with the air intake towers in the foreground. The right tower supplies the office wing. The glazed area is the stairwell. The tower above the stairwell is the exhaust air tower.

The HiNT-wing has a net area of 478 m², of which 269.5 m² are used as office cells. Each cell is about 9 m². The gross area of the wing is 835 m², of which 112 m² is used for culvert, air intake tower and exhaust tower.

The hybrid ventilation is of so called culvert type. In principle it is constructed as shown in Figure 3. The ducts from the culvert to the rooms are buried in the ground beneath the floor. At the façade the ducts turn 90° upwards. The ducts end in a damper placed inside the supply air terminal device. The air terminals are placed at the floor beneath the windows. Air to the first floor is supplied through enclosed ducts at the inside of the façade.

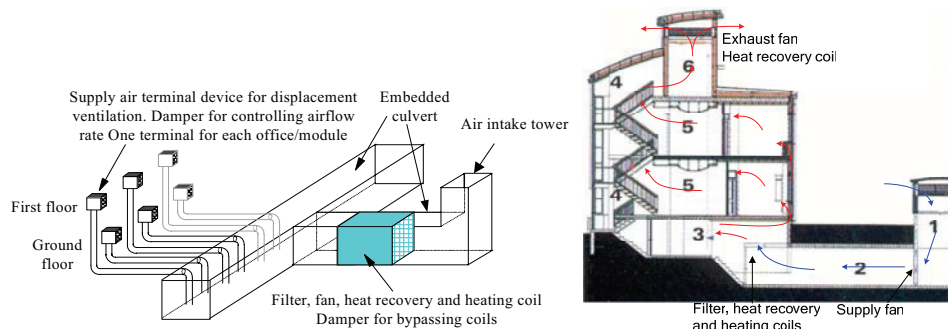


Figure 3. Culvert in principle and a section through the building. 1: Air intake tower. 2: Air intake culvert. 3: Air distribution culvert. 4: Stairway. 5: Corridors. 6: Exhaust air tower.

From the offices the air flows through grilles placed close to the ceiling and into the corridor. Exhaust of air takes place through corridors and stairway up to the tower on the roof. The exhaust air tower contains a heat recovery coil and a fan.

Radiators are placed beneath each window. None of the offices have mechanical cooling. Each office cell has a presence detector and a temperature sensor and a digital controller unit that controls heating, ventilation and lighting. Venetian blinds are automatically and simultaneously controlled for each façade, with a possibility for individual control. Bus is used for communication.

Figure 4 shows the ventilation and controller system in principle. The supply and exhaust fans are controlled by the pressure difference between the culvert and the corridor at the ground floor, i.e. the fans should keep the pressure difference constant. When there is no heating demand for the ventilation air and the outdoor temperature exceeds 15 °C, the bypass dampers opens to reduce the pressure drop.

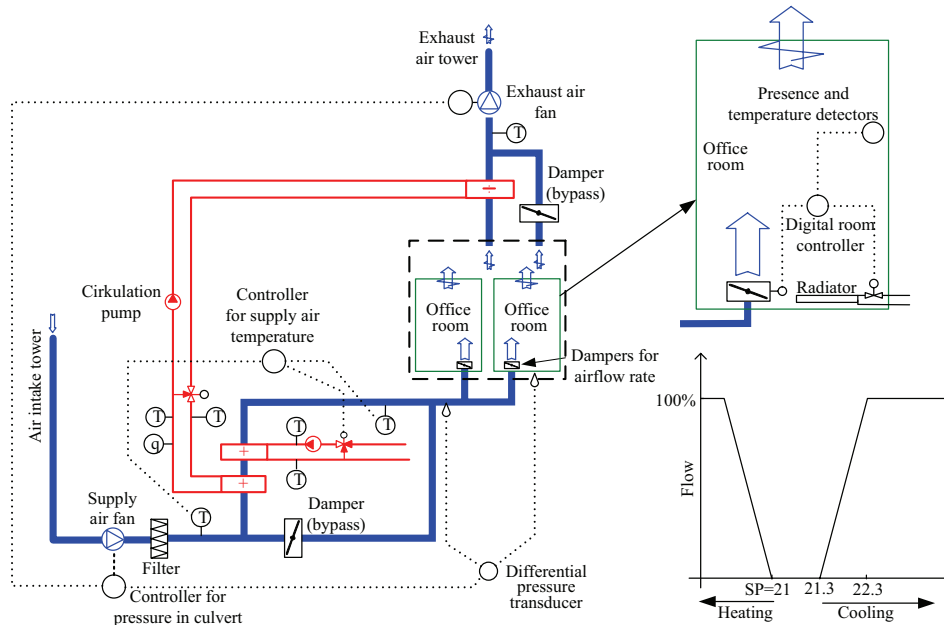


Figure 4. Ventilation system in principle and the control strategy for the room controller.

The dampers in the supply air terminal devices control the air flow rate to each room. They operate as follows: Normally they close at 4 pm. At 6 am open to give approximately 25 m³/h (2.8 m³/hm²) of air. When a person enters the room they open to give 43 m³/h (4.8 m³/hm²). If the room air temperature exceeds the set-point they open further. When the temperature is one degree above the set-point they are fully opened and give about 200 m³/h (22 m³/hm²).

If the room air temperature is above the set-point, the dampers also open at night. Total airflow rate supplied to the HiNT-wing is 8 600 m³/h. 800 m³/h is drawn from toilets without any heat recovery.

4. Description of the simulation model

The simulation model is implemented in the building/plant simulation program ESP-r (ESRU, 2000). The model is a whole-building model. Figure 5 shows how the model appears in the cad-window in the ESP-r Project Manager.

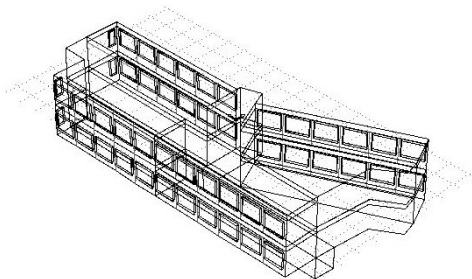


Figure 5. The whole-building simulation model

Air supply control

The air flow is controlled both by the presence of people and temperature in the specific zones. The minimum air flow for an occupied zone is approximately $4.8 \text{ m}^3/\text{hm}^2$ and $2.8 \text{ m}^3/\text{hm}^2$ in an empty zone during the working hours. If the temperature raises above the set-point the air volume increases proportionally with the temperature until full opening at $22 \text{ m}^3/\text{hm}^2$. The presence of people is fixed in the control strategy, and is given by the occupational load in the internal gains. Table 2 outlines the air control strategy.

Table 2. Air flow rates at different times of a working day. Set-points (SP) change over the year and room-type. See Figure 7 for the SP-values. The office rooms are occupied 8-10 and 13-16.

Time	Min airflow [m^3/hm^2]	Max airflow [m^3/hm^2] (Temp. controlled)	Control range [$^{\circ}\text{C}$]
0-6	0.0	22.0	SP + 1.5 $^{\circ}\text{C}$ → SP + 2.0 $^{\circ}\text{C}$
6-8	2.8	22.0	SP + 0.3 $^{\circ}\text{C}$ → SP + 1.3 $^{\circ}\text{C}$
8-10	4.8	22.0	SP + 0.3 $^{\circ}\text{C}$ → SP + 1.3 $^{\circ}\text{C}$
10-13	2.8	22.0	SP + 0.3 $^{\circ}\text{C}$ → SP + 1.3 $^{\circ}\text{C}$
13-16	4.8	22.0	SP + 0.3 $^{\circ}\text{C}$ → SP + 1.3 $^{\circ}\text{C}$
16-24	0.0	22.0	SP + 1.5 $^{\circ}\text{C}$ → SP + 2.0 $^{\circ}\text{C}$
Other days	0.0	22.0	SP + 1.5 $^{\circ}\text{C}$ → SP + 2.0 $^{\circ}\text{C}$

Set-points for heating and ventilation

The set-points for heating and ventilation vary over the year. The office cells have individual set-points for heating, while the set-points for the common area and the supply air are set by the operator. All rooms have 2 K night set-back relative to the set-point during working hours. Figure 7 shows the actual monitored set-points that were used in the simulations. The reason for the increased set-point for the supply air during winter period is due to complaints about draught against ankles.

Internal gains

The internal gains are persons (occupational load), equipment and lighting. In every zone type, except for one server room, all internal loads are shut down during night hours and weekends. Figure 8 shows the internal gains during a working day.

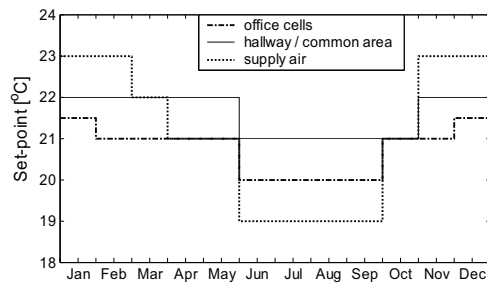


Figure 7. Set-points for the office cells, the hallways/common areas and the supply air.

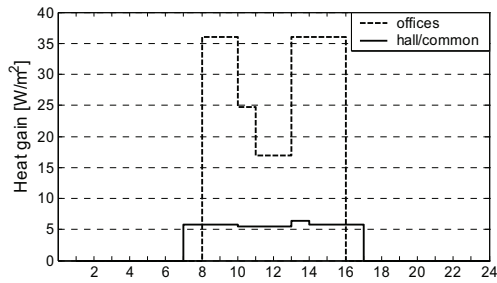


Figure 8. The internal gains in the office cells and the hallway/common area

Climate

Ideally, the climate data should have been recorded at the location in the same period the energy consumption measurements were done. As this data were insufficient, it was decided to use the hourly Test Reference Year (TRY) data produced by the program Meteonorm (Meteonorm, 2004) for Trondheim Airport Vaernes, located 30 km south of Levanger. The official air temperature measurements from the meteorological station at Vaernes (MET.no, 2005) show that the annual average temperature during the time of this study was $1.5 \text{ }^{\circ}\text{C}$ above the standard year. The TRY values however, are somewhere between the standard year and the actual measurements. Table 3 shows the monthly average air temperatures from Vaernes (met.no), the TRY and the standard year, respectively.

Table 3. Monthly average air temperatures [$^{\circ}\text{C}$] for the time of study, TRY and the standard year

	jan	feb	mar	apr	may	jun	jul	aug	sep	oct	nov	dec	avg
met.no	-3.4	-0.4	2.7	7.9	9.2	13.1	17.6	14.2	10.7	3.8	2.5	0.5	6.5
TRY	-4.4	-4.3	-0.1	4.4	10.8	15.1	16.5	15.3	10.6	6.4	0.7	-3.0	5.7
Standard	-3.4	-2.5	-0.1	3.6	9.1	12.5	13.7	13.3	9.5	5.2	0.5	-1.7	5.0

Simulation strategy

The simulations were run month by month to compare directly to measured values. The heat recovery unit had a failure during December and January. This has been taken into account in the calibrated simulation model by reducing the efficiency to 20 %.

5. Simulations

Basic model

The results for monthly heating of the rooms and the ventilation air from the calibrated basic simulation model are given in Figure 9. Table 4 shows the total energy use and the relative agreement to measured values.

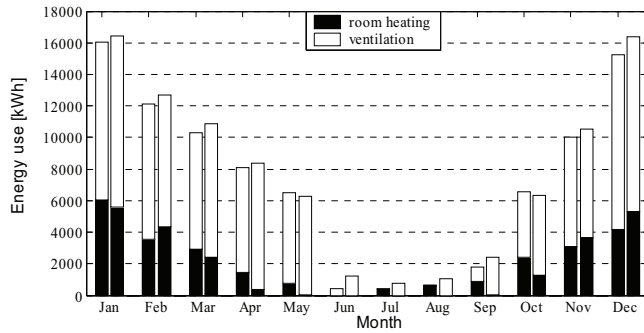


Figure 9. Energy use distributed on room heating and ventilation air preheating for each month. First bar is measured values and the second is the results from the simulation model.

Effect of adjusting the control range

The objective of this section is to study the impact of allowing the room temperature to glide somewhat more than in the basic model before full damper opening occurs in the air supply terminals. The same set-points for room and ventilation air temperature as in the basic model are used. The only modification from the basic model is that the heat exchanger efficiency is constant throughout the year. Two cases are simulated with 45 % and 55 % heat exchanger efficiencies, respectively.

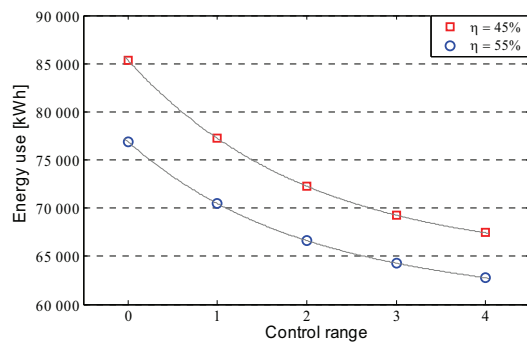


Figure 10. Energy use as a function of the control range and efficiency of the heat recovery unit. The meaning of the control ranges are explained in Figure 11.

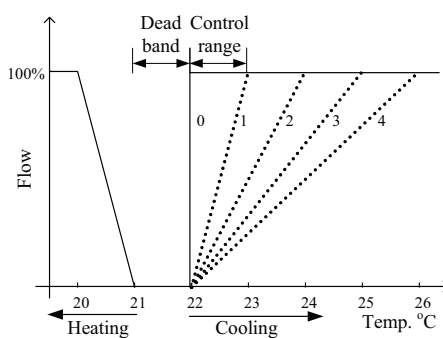


Figure 11. The different control ranges. The set-point (SP) is 21 $^{\circ}\text{C}$. The dead band (DB) is 1 $^{\circ}\text{C}$.

Comparing measured energy use with results from Figure 10 indicates that increasing the dead band up to 1 K and allowing 3 K temperature glide before full damper opening, makes it possible to save about 25% of the total energy use for heating.

Excessive temperature

Hours with room air temperature above 26 °C increase about 6 hours during a year when changing the upper set-value from 23 °C to 24 °C, and additional 10 hours for 25 °C. However, temperatures above 26 °C occur less than 50 hours during a year within working hours, but will exceed 50 hours with higher upper set-values.

Other simulations

Table 4 summarizes some of the simulations carried out in this study. The first two rows in Table 4 are the measured values and the simulated values, respectively, as presented in Figure 9. The third row (Basic_CR3) is a slight modification of the basic model. The only difference from the basic model is that the control range is set to 22-25 °C instead of 21.3-22.3 °C. The fourth row presents the results from a model where the set-points and technical solutions are in accordance with the designed values, i.e. the supply air is held at 20 °C the entire year and the heat exchanger efficiency is constantly 45 %. The “Improved design” configuration has a control range of 22-25 °C and a heat exchanger efficiency at 55 %. The latter two configurations presuppose that the draught problems are solved.

Table 4. Comparison of different cases

Case	Supply air temperature, summer/winter	Room heating [kWh]	Ventilation air preheating [kWh]	Total energy use [kWh]	Rel. difference from measured, total [kWh]
Measured values	As Figure 7	26 317	61 874	88 192	-
Basic model	As Figure 7	23 071	70 289	93 360	+ 6 %
Basic_CR3	As Figure 7	23 673	49 108	72 782	- 17 %
Designed	20/20	27 922	43 455	71 377	- 19 %
Improved design	19/20	25 775	32 867	58 642	- 34 %

6. Discussion

The simulated basic model and the measured values agree well. It must be remembered that measured values has uncertainty due to measurement accuracy and missing data in some periods. The simulations are based on a climatic year different from the real climate. Further, the real case’s set-points differ from room to room and frequently during the year, while the values used in the simulation model are monthly averages of monitored values. The calculated U-values due to taking the thermal bridges into account are another uncertainty in the simulation model.

May and October are the only months the simulation model determines less energy use than the measurements. This can be explained by the fact that these months have a higher average outdoor air temperature (see Table 3).

In comparing measurements and simulations, the total energy use for heating agrees better than the separate energy use for heating the rooms and the ventilation air. This can be explained by the narrow dead band between radiator heating and cooling by increasing the ventilation air. For instance, if the simulation model over-predicts room temperature by as little as 0.3 °C (21.3 °C instead of 21.0 °C see Figure 4) this leads to ventilation air heating demand instead of room heating demand.

The embedded ducts have a heat loss that is difficult to estimate because the temperature in the ground varies during the year. On the contrary, the lost heat leads to higher temperature in the ground that might reduce the heat loss through the floor.

The “Designed” and “Improved design” cases require that the ventilation works without draught problems. Today the supply air temperature is increased by the operator in order to reduce the draught sensation. High supply air temperature gives increased heat loss from the culvert, the embedded ducts, and reduced ventilation efficiency (Mundt, 2004). If the room temperature is allowed to glide more before full opening of the dampers, the airflow rate is reduced. Reduced airflow rate implies lower air velocity in the vicinity to the air terminal. On the other hand, higher temperature difference between room air and inlet air increases the density differences and

hence the acceleration of the air. However, higher room air temperature implies that higher air velocities are accepted before the users sense draught. (Skistad, 2002), (Mathisen, 1989).

7. Conclusions

- Annual total specific energy use is measured to be 161 kWh/m² heated gross area, of which 111 kWh/m² are hydronic space heating and heating of ventilation air.
- The simulations carried out in ESP-r agree well with the measurements.
- The thermal environment is satisfactory during summertime with less than 50 hours with temperature above 26 °C. Even if the temperature glide is allowed up to 25 °C before full damper opening, the number of hours with over temperature increases with only a few hours.
- The total airflow rates during working hours are relatively large; this is due to the fact that the control range for the room temperature was only 1 K. On cold days this means that preheating in addition to recovered heat is necessary. If the dead band between radiator heating and cooling by increased ventilation is expanded to 1 K, and the temperature is allowed to glide 3 K before full opening, it is possible to save about 25 % of the total energy use for heating compared to the measurements.
- Increasing the heat exchanger efficiency from 45 to 55% will save about 7-11% of the total energy use, depending on the control range. That is, the energy use is more dependent on the control range for the room temperature than the heat exchanger efficiency.
- The simulations show that if the type of hybrid ventilation system presented in this paper is well designed and operated, moderate energy use can be obtained.

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Paper II: Comparison of multi- and single zone modeling for estimation of energy use in an office building

Rasmus Z. Høseggen, Hans M. Mathisen and Sten O. Hanssen

Abstract

Building simulation programs are powerful tools to predict the thermal climate and the energy use in buildings. However, simulations for estimating the energy use are rarely used by consultants.

This paper describes two ways of modeling an office building. A case building has been monitored for energy use over a year, and the measurements are used to calibrate two simulation models. The first model simulates a single-zone and takes into account only the adjacent rooms. The cell is further modified to cover all room types in the building. The other model is a full-building model.

The objective of this study is to find whether the single-zone modeling approach is appropriate for energy use estimations. Results from the simulations show that it is possible to come very close to measured values with the full-building model. The single-zone tends to over-estimate the need for heating of ventilation air. This must be taken into consideration if such an approach is chosen.

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COMPARISON OF MULTI AND SINGLE-ZONE MODELLING FOR ESTIMATION OF ENERGY USE IN AN OFFICE BUILDING

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ABSTRACT

Building simulation programs are powerful tools to predict the thermal climate and the energy use in buildings. However, simulations for estimating the energy use are rarely used by consultants.

This paper describes two ways of modelling an office building. A case building has been monitored for energy use over a year, and the measurements are used to calibrate two simulation models. The first model simulates a single-zone and takes into account only the adjacent rooms. The cell is further modified to cover all room types in the building. The other model is a full-building model.

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INDEX TERMS

Simulation, measurements, annual energy use, ESP-r, modelling approach

INTRODUCTION

Building simulation is considered one of the most important efforts in increasing buildings' energy efficiency. Still, simulations for estimating the energy use are rarely used by consultants. Constructing building simulation models is considered too time-consuming and, consequently, too expensive to be a part of the design process. Full-building modeling is only applied in large scale projects (TIP-vent 2000). In recent years, however, it has become more common to use simulation tools to predict the thermal environment in extreme rooms e.g. to identify the need for cooling. A skilled engineer will not use more than one hour to establish such a single-zone model.

The objective of this paper is to analyze whether or not the single-zone model, basically meant for thermal environmental analysis, is applicable to estimate annual energy use for an office building.

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RESEARCH METHODS

The case building

The case presented in this paper is connected to the Nord-Trøndelag College (HiNT) building in Levanger, located 80 km north of Trondheim, Norway. The building has a joint wing with meeting rooms and auditoriums. Two separate wings are office areas, see Figure 1. To simplify the project only one of the office wings (the HiNT-wing) were used in the study. The building has two storeys. There is no basement, but a culvert for supply of ventilation air is embedded in the ground along the middle of the wing.

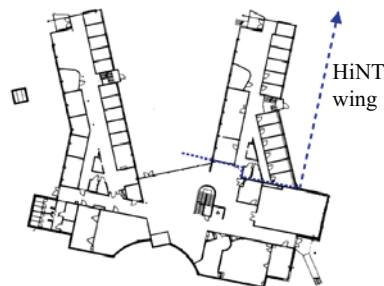


Figure 1. Plan view of the building, showing the common area and the two office wings.

The HiNT-wing has a net area of 478 m², of which 269.5 m² are for office cells of about 9 m² each. The gross area of the wing is 835 m², of which 112 m² is used for culvert, air intake tower and exhaust tower.

The hybrid ventilation is of the so-called culvert type. Figure 2 shows the principle of the construction. From the culvert ducts, transporting air to the rooms, are placed in the ground beneath the floor. At the façade the ducts turn 90° upwards. The ducts end in a damper placed inside the supply air terminal device. From the offices the air flows through grilles to the corridor. There is a heat recovery unit and a fan in the exhaust air tower. Radiators are placed beneath each window. None of the offices have mechanical cooling. Lighting, heating, and ventilation are demand-controlled, and venetian blinds are simultaneously controlled for each façade with a possibility for individual control.

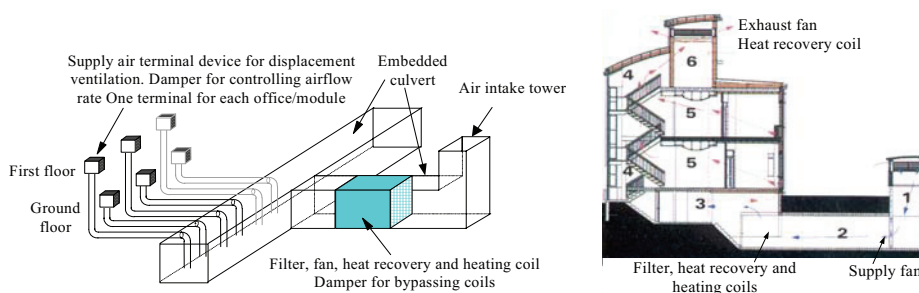


Figure 2. Culvert in principle and as a section through the building

Simulation models

The simulation models are implemented in the building/plant simulation program ESP-r (ESRU, 2000) in two ways. The simplest model consists of a single-zone that covers an office cell and a narrow part of the hallway. The other is a full-building model. Both models are

given the same properties in terms of building materials, set-points for temperature, ventilation and operation. Figure 3 and Figure 4 show how the models appear in the cad-window in the ESP-r's Project Manager, respectively.

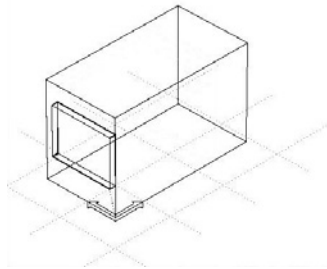


Figure 3. Single-zone model in the Project manager

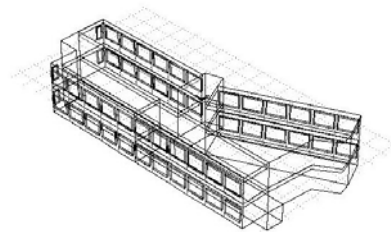


Figure 4. Full-building model

Construction elements

The modelling of the building elements are done by investigating the construction drawings. In principle the building envelope is very well insulated, far more than required by the national building code. However, the extensive occurrences of thermal bridges make the U-value increase significantly. Table 1 shows the basic U-values and the thermal bridge-corrected U-values calculated by methods described by the Norwegian guidelines and EN-ISO-6946 (EN-ISO 6946:1996)

Table 1. U-values for selected building constructions

Building construction	Basic U-value [W/m ² K]	Thermal bridge-corrected U-value [W/m ² K]
External walls	0.18	0.33
Roof	0.12	0.17
Floor on ground	0.15	0.17
Windows	1.31	1.40

Air flow network model

Supply air is taken through the culvert before it is mixed with extract air from the zone in the heat recovery unit. The mixing of the two air flows is a simple approach to model the heat exchanger, and is not meant to be a recirculation of air. If necessary, the air is preheated to the given set-point. The air flows further into the air distribution culvert, which can also be seen in Figure 4 under the office building. From the distribution culvert the zones are supplied with air by a fan. For the full-building model there are a total of eight fans supplying the eight air-conditioned zones. The air is extracted from the zones by two fans, one feeds the heat recovery zone, and the rest is led to the outside air.

Both models have a free cooling system. When the outside dry bulb temperature exceeds 15°C, the air is by-passed the heat recovery unit. In Figure 5 that means that damper 4 opens, while dampers 2 and 3 are shut. Damper 1 opens to fulfil the mass balance.

The air intake culvert and the air distribution culvert are embedded ground coupled concrete ducts. The floors and the walls are modelled as a concrete layer and an insulation layer. One meter of earth is also included in these constructions. The construction is coupled to a constant temperature in the ground; 10°C underneath the building and 5°C outside the building. This is a fairly rough simplification, but gives a good approximation to the complex

physics in the ducts (Wachenfeldt 2003). This approximation is also done for the floor constructions with ground coupling.

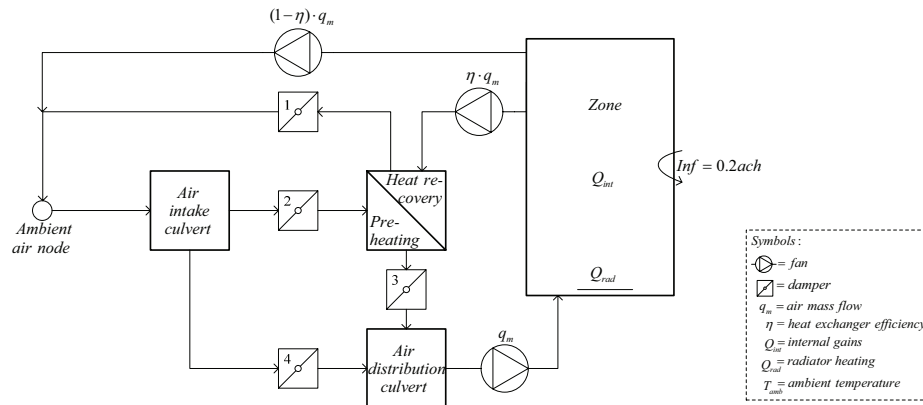


Figure 5. Principal scheme of the model with an air flow network

The air intake culvert and the air distribution culvert are not modelled in the single-zone model, but are replaced by adiabatic dummy zones. However, the additional heat loss from the air distribution culvert is taken into account.

Air supply control

The air flow is controlled both by the presence of people and temperature in the specific zones. The minimum air flow for an occupied zone is approximately $5 \text{ m}^3/\text{m}^2\text{h}$ and $2.5 \text{ m}^3/\text{m}^2\text{h}$ in an empty zone during the working days. If the temperature raises above the set-point the air volume increases proportionally with the temperature until full opening at $20 \text{ m}^3/\text{m}^2\text{h}$. The presence of people is hard coded in the control strategy, and is given by occupational load in the internal gains. Table 2 outlines the air control strategy.

Table 2. Air flow rates at different times of a working day. Set-points (SP) change over the year and room-type. See Figure 6 for the SP-values.

Time	Min airflow [$\text{m}^3/\text{m}^2\text{h}$]	Max airflow [$\text{m}^3/\text{m}^2\text{h}$] (Temp. controlled)	Regulation band [$^{\circ}\text{C}$]
0-7	0.0	20.0	SP + 1.5°C → SP + 2.0°C
7-8	2.5	20.0	SP + 0.3°C → SP + 1.3°C
8-10	9.0	20.0	SP + 0.3°C → SP + 1.3°C
10-13	2.5	20.0	SP + 0.3°C → SP + 1.3°C
13-16	9.0	20.0	SP + 0.3°C → SP + 1.3°C
16-24	0.0	20.0	SP + 1.5°C → SP + 2.0°C
Other days	0.0	20.0	SP + 1.5°C → SP + 2.0°C

Set-points for heating and ventilation

The set-points for heating and ventilation vary over the year. The office cells have individual set-points for heating, while the set-points for the common area and the supply air are set by the operator. Figure 6 shows the actual monitored set-points that were used in the simulations.

Internal gains

The internal gains are persons (occupational load), equipment and lighting. In every zone type, except for the server room, all internal loads are shut down during night hours and weekends. Figure 7 shows the internal gains during a working day.

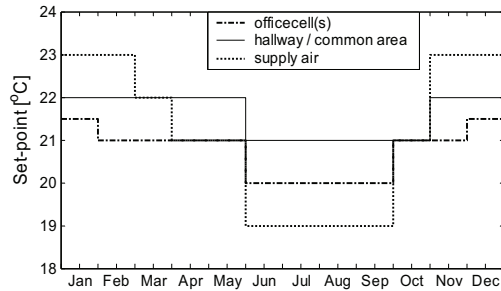


Figure 6. Set-points for the office cells, the hallways/common areas and the supply air.

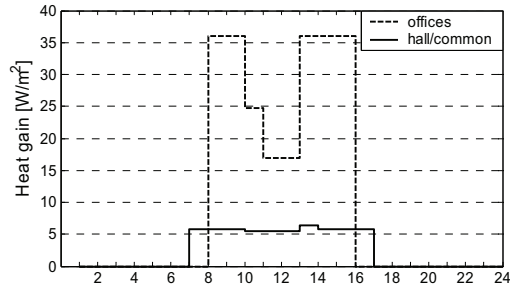


Figure 7. The causal gains in the office cells and the hallway/common area

Climate

Ideally, the climate data in this study should be measured at a place close to the building in the same year the energy consumption measurements were done. As this data were not available or insufficient, it was decided to use the hourly Test Reference Year data produced by the program Meteonorm (Meteonorm 2004) for Trondheim Airport Vaernes, located 30 km south of Levanger.

Simulation strategy

The single-zone's constructions and boundary conditions were modified in order to cover all rooms in the building. All these single simulation were finally synthesized for the whole building. Both for the single-zone model and the full-building model, the simulations were run month by month to compare directly to measured values.

RESULTS

Measured energy use for room heating is 26 317 kWh. Single-zone simulations show 17 389 kWh and the full-building model 22 820 kWh. For ventilation the numbers are 61 874, 67 507 and 83 382 kWh, respectively. The results for monthly heating of the rooms and the ventilation air are given in Figure 8. Table 3 shows the total energy use and the relative agreement to measured values.

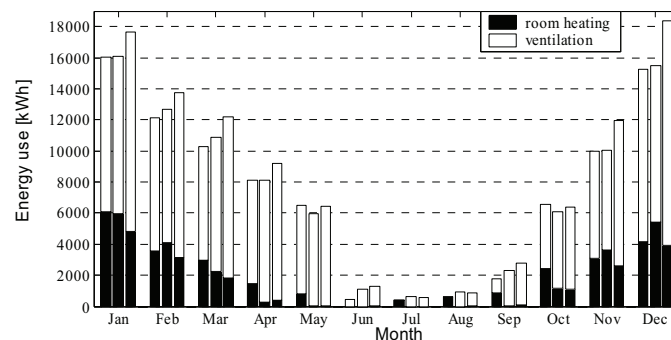


Figure 8. Energy use distributed on room heating and ventilation air for each month. First bar is measured values, second the whole building model and third bar is the single-zone model.

Table 3. *Total energy use for heating of rooms and ventilation air*

Model	Energy use [kWh]	Relative difference from measured
Measured value	88 192	-
Full-building model	90 327	+ 2.4 %
Single-zone model	101 346	+ 14.9 %

DISCUSSION

The results from the simulations show that the single-zone model tends to over estimate the heating needed for ventilation air compared both to the measurements and the full-building model, especially for months with need for ventilation preheating. There are mainly two reasons for this. (1) The air intake culvert is not included in the single-zone model. The culvert transfers heat by convection to the intake air on cold days. This will reduce the need for heating of the ventilation air. The estimated gain from the culvert is approximately 500 kWh per month from November to April. (2) The single-zone model has no opportunity to utilize the fact that zones may have different needs for heating at different times of the day. For instance, in the morning the sun will heat the east façade and may create an over-temperature in these zones. This over-temperature is utilized in the heat exchanger for the full-building model, but for the single-zone this excessive heat is dumped. This must be taken into consideration when the single-zone model approach is being used.

CONCLUSION AND IMPLICATIONS

Both models agree reasonably compared to measurements, but the single-zone model tends to over-estimate the energy use for ventilation air. This must be taken into consideration if a single-zone model is being used to estimate annual energy use in buildings. For buildings with traditional mechanical ventilation systems this approach will be more feasible, since the losses/gains in the culverts are not influencing the results. However, the issue on simultaneity will be equally important. The single-zone modelling is a relatively quick way to estimate the energy use compared with the full-building model, but as discussed above, it will somewhat over-estimate the ventilation heating need. Further studies on other types of buildings must be done in order to conclude on a general level.

ACKNOWLEDGEMENTS

This research could not been done without the support from the Directorate of Public Construction and Property in Norway.

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Paper III: Reduced energy demand by combining natural and mechanical ventilation in a new office building with an atrium

Rasmus Z. Høseggen, Bjørn J. Wachenfeldt and Sten O. Hanssen

Abstract

A new office building is will raised in the city centre of Trondheim, Norway. The new building consists of an atrium and an office wing. The east façade of the office wing is planned with a double façade. The building is planned with mechanical ventilation with cooling. The only possibility for utilising natural ventilation is through some fire hatches in the atrium. A computer model of the building is implemented in the building/plant simulation program ESP-r. Simulations showed that in warm periods, satisfactory thermal conditions will be achieved with the originally proposed design. However, the ventilation system must run day and night in warm periods in order to cool the thermal mass of the building. The simulations also revealed that the possibility to ventilate naturally will not work as intended. The objective of this study is to investigate a potential improvement of the proposed design via use of building simulation technology. An alternative design that aims to utilise natural ventilation more actively to reduce the demand for mechanical cooling was implemented in the computer model. Simulations indicate that natural ventilation can reduce the demand for mechanical cooling with more than 50% while reducing the number of hours with unacceptable high temperatures in the occupied zones and avoiding night time fan operation.



**The 5th International Conference on Cold Climate
Heating, Ventilation and Air-Conditioning**

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REDUCED ENERGY DEMAND BY COMBINING NATURAL AND MECHANICAL VENTILATION IN A NEW OFFICE BUILDING WITH AN ATRIUM

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Keywords: Night ventilation, energy savings, ESP-r

ABSTRACT

A new office building is will raised in the city centre of Trondheim, Norway. The new building consists of an atrium and an office wing. The east façade of the office wing is planned with a double façade. The building is planned with mechanical ventilation with cooling. The only possibility for utilising natural ventilation is through some fire hatches in the atrium. A computer model of the building is implemented in the building/plant simulation program ESP-r. Simulations showed that in warm periods, satisfactory thermal conditions will be achieved with the originally proposed design. However, the ventilation system must run day and night in warm periods in order to cool the thermal mass of the building. The simulations also revealed that the possibility to ventilate naturally will not work as intended. The objective of this study is to investigate a potential improvement of the proposed design via use of building simulation technology. An alternative design that aims to utilise natural ventilation more actively to reduce the demand for mechanical cooling was implemented in the computer model. Simulations indicate that natural ventilation can reduce the demand for mechanical cooling with more than 50% while reducing the number of hours with unacceptable high temperatures in the occupied zones and avoiding night time fan operation.

BACKGROUND AND OBJECTIVES

The background for this study is a request from BARK Architects who wanted to investigate the thermal conditions in the building, particularly in the upper parts of the atrium. In addition, the project team wanted to look at the possibility for utilizing natural ventilation via a double-skin façade and an atrium in order to reduce or eliminate the need for mechanical cooling of a new office building.

METHODOLOGY AND MODELLING

A simulation model of the planned building is implemented in the building/plant simulation program ESP-r (ESRU, 1999), which is a transient simulation program based on the finite volume technique. ESP-r is capable of modeling the energy and fluid flows within combined building and plant systems when constrained by control actions and subject to dynamically varying boundary conditions.

The method used in this paper is to model the building as planned, and compare it against a modified version that utilizes natural ventilation. From here on the models will be referenced to as Case 1 and Case 2, respectively.

The building

The new office building is to be raised in the city centre of Trondheim, Norway. The building will be integrated with existing brick buildings. The new building consists of an atrium and an office wing, both five storeys tall. The east facing façade has a double-skin façade. The ground floor in the atrium is planned as a reception and the first floor is a common area. Both floors are planned to have open-plan offices. Further up the atrium there are balconies with working places and space for group activities. The office wing has both cell offices and open-plan offices distributed on all five floors. There are no walls obstructing the air flow between the office wing and the atrium. In total there are about 150 and 100 working places in the office wing and the atrium respectively. Figure 1 shows the cross-section of the building with the atrium in the middle, and the office wing to the right. Light gray lines represent existing buildings at the site.

In this study, only the atrium and the 2nd, 3rd, and 4th floor of the office wing are modeled (about 2000 m² of floor surface area in total). The reason is that the remaining part of the building is designed with a separate mechanical ventilation system. Figure 2 shows how the simulation model appears in the cad-window in ESP-r.

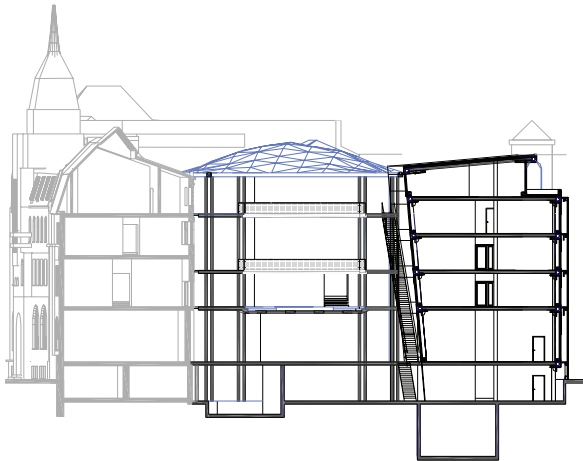


Figure 1. Cross section of the case building. Light gray lines represent existing buildings.

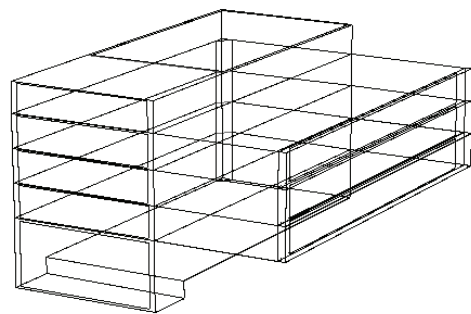


Figure 2. The simulation model, how it appears in the cad-tool in ESP-r.

Construction elements

Properties of some selected building materials are presented in Table 1. The optical data are calculated by WIS (WIS, 2005) which is software that determines the thermal and solar characteristics of

window systems. The U-values for the windows are given by the manufacturer, and the wall's U-value is from the standard building code (REN, 2003).

Table 1 Properties of selected construction elements

Construction	U-value [W/m ² K]	Visible transmittance (g-value) [-]
Windows facing the cavity in the double-skin façade and in the south façade	1.40	0.253
External glazing in the double-skin façade without blinds	5.10	0.416
External glazing in the double-skin façade with blinds angled 45°	3.00	0.235
Glass roof in the atrium with semi-transparent canvas	1.40	0.212
External walls	0.22	-

The floors in the office wing are suspended concrete floors with a wooden finish. The concrete is exposed in the ceiling. The external walls of the existing buildings that face the interior space of the new building are made of brick. The use of concrete and brick results in relatively high thermal capacity of the interior construction, making it an interesting case for investigating night cooling strategies.

Mechanical ventilation

The mechanical ventilation rates are set to meet the Norwegian codes and guidelines (REN, 2003), which is 7 l/s per person and an additional 2 l/s/m² for emissions from building materials. The resulting specific ventilation rates for every room type are given in Table 2. The temperature for the supply air is 15°C. The operating time for the mechanical ventilation is set to 6 am – 8 pm.

Two cases are analyzed. Case 1 considers the proposed design, while Case 2 considers an alternative design utilizing natural ventilation to reduce or avoid mechanical cooling. For Case 1, the ventilation rate outside working hours is controlled by the operative temperature individually for each zone. That is, if the temperature indoors is above the 20°C, the ventilation rate increases proportionally with the temperature until full flow rate at 23°C. For Case 2, the ventilation system is always shut down outside operating time.

Table 2. Ventilation rates in different zones, during working hours

Zone	Ventilation rate [m ³ /h/m ²]
Office wing	10.2
Common area	8.9
Reception	8.9

Internal heat gains

Values for the internal heat gains are taken from the Norwegian Standard (NS3031, 1987). Maximum load is assumed in the office wing, and 50% of maximum is used as mean load in the reception and the common area. The heat gains apply during the working hours, Monday to Friday 8 am to 5 pm, while they are switched off outside working hours. Equivalent heat gains are listed in Table 3.

Table 3 Equivalent internal heat gains according to the NS3031

Heat source	Office wing [W/m ²]	Reception and common area [W/m ²]
Persons	12	8
Lighting	8	4
Equipment	4	2

Climate data

In this study, hourly Test Reference Year (TRY) data produced by the program Meteonorm (Meteonorm, 2005) for Trondheim was used as climate data. Figure 3 shows the most influential parameters during a warm and sunny week in August.

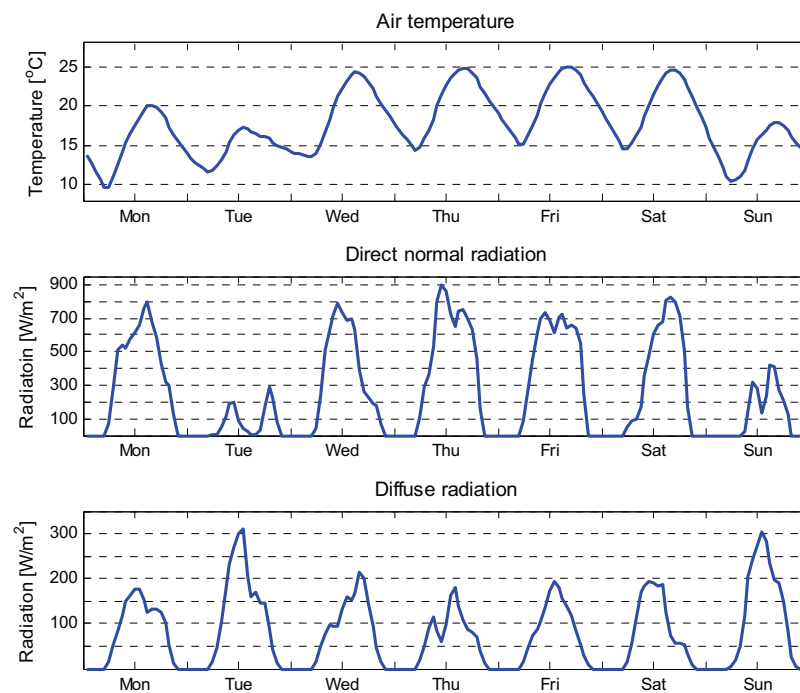


Figure 3 Selected climate parameters for August 13.-19.

Simulation model

The following assumptions and simplifications have been done to the simulation model:

- The base area is modeled as rectangular for both the atrium and the office wing, although the geometric shape in reality is trapezoidal. However, the floor and external wall surface areas are in accordance with the drawings.
- The atrium is divided horizontally in three zones. Although this is a rough approximation, it makes it possible to take thermal stratification into account, and improves the accuracy of the air flows through the atrium.
- The double-skin façade has venetian blinds within the cavity which shut when the global radiation on the façade exceeds 200 W/m². The blinds are individually controlled for each floor.

- The solar obstruction caused by the high rise buildings situated to the east is taken into account, while the potential effect from the low rise buildings situated to the south is ignored.

Modeling the double-skin façade and the airflow network

Guidelines for modeling double-skin façades are practically non-existent and even examples are rare. However, some studies on this issue are done by Dickson (Dickson 2004) and Hensen (Hensen et. al 2002). Both studies underline that the modeling of double-skin façades is particularly complex because of the high degree of interactions between the many heat transfer processes. To obtain a tolerably accurate prediction of the façade performance, it is important to ensure appropriate treatment of the solar insolation, the cavity convection regimes, the surface view-factors, blind spatial position, airflow resistances, vertical temperature gradient and cavity divisions with fictitious divisions (Dickson 2004).

For this particular case, the double-skin façade is divided into a stack of three zones adjacent to each of the office levels. The zones are divided by fictitious transparent surfaces with high conductivity, negligible thermal mass and high emissivity, and coupled by an airflow network which also includes the inlet opening at the bottom and the top outlet opening at the top of the façade. Since the double-skin façade will be ventilated, the *Bar-Cohen & Rohsenow* correlation is used to predict the convective heat transfer for the surfaces facing the cavity.

In Case 2 with natural ventilation, some of the office windows facing the cavity of the double-skin façade are operated automatically. The operation depends on both the temperature in the office and in the cavity of the double-skin façade. That is, if the temperature indoor reaches the low set-point, the window opening area increases proportionally with the temperature until the window is fully open at the high set point. If the temperature in the double-skin façade exceeds 25°C, the windows close in order to avoid increased cooling demand. Figure 4 shows that this is especially relevant for the windows in the upper part of the double-skin façade, where the temperature can exceed 30°C on warm and sunny days.

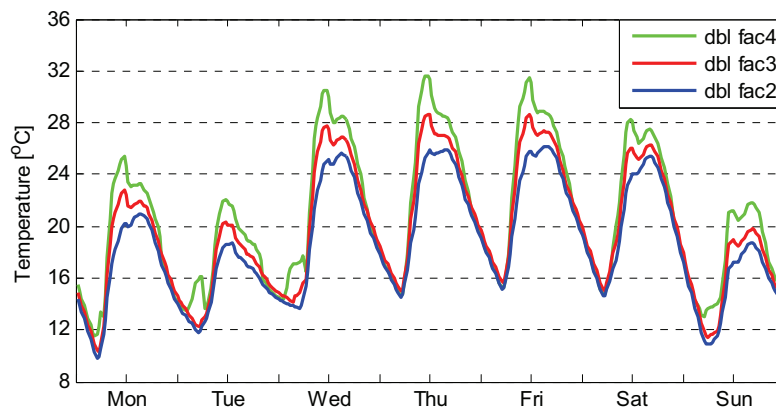


Figure 4. Simulated air temperatures in the double-skin façade for the three floors August 13-19

In ESP-r there are currently no control functions that are able to sense two parameters at the same time. Since the operation of the windows depends on the temperature in the office as well as in the cavity of the double-skin façade, it is necessary to introduce a dummy air node in the simulation model. This makes it possible to have two openings between the indoor air node and the node in the double-skin façade. One opening represents the actual window, and the other is an opening with negligible fluid resistance when open. When the temperature in the double-skin façade exceeds 25°C, the latter opening shuts in order to prevent this hot air from entering the offices. Figure 5 illustrates how it is implemented

for the window openings in the 3rd floor. Damper A represents the window, and damper B is the fictitious opening.

The control strategy is in principle also implemented for the windows and hatches facing the ambient, in which case damper B in Figure 5 is controlled by the ambient temperature. Figure 6 shows the control strategy for damper A, i.e. the operation of the windows and hatches as a function of room air temperature.

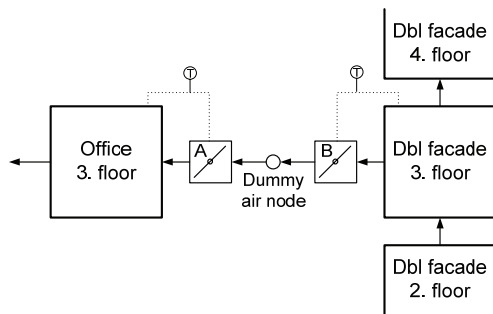


Figure 5. Principal sketch of the window control. Damper A represents the window and is controlled by the indoor temperature. B is controlled by the temperature in the double-skin façade.

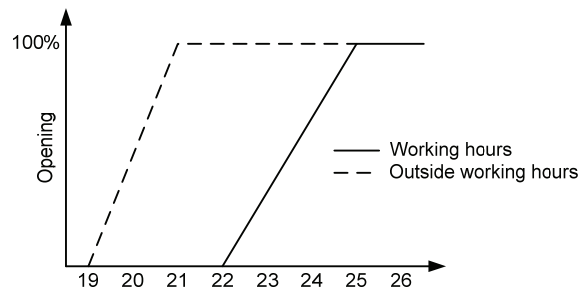


Figure 6. Control strategy for the windows and the hatches. The dashed line yields outside working hours, while the solid line yields from 6 am – 8 pm.

SIMULATION RESULTS

As mentioned previously, two cases are compared:

1. The planned solution, where the only possibility to ventilate naturally is via a fire hatch over the entrance to the reception and in the ceiling on warm and sunny days. The hatches are closed outside working hours. The mechanical ventilation system operates outside working hours if needed.
2. Natural ventilation is more actively used. The windows in the office wing and the fire hatches are controlled, also during the night. In addition to the planned fire hatches, some new hatches are placed higher up on the south façade.

For the worst case scenarios, considering the warm week August 13-19, the effect of wind is ignored for both Case 1 and Case 2, resulting in natural ventilation being driven by thermal buoyancy only. Additional simulations have been carried out for the period outside the heating season, May 1 to October 1. For these simulations the effect of wind is taken into account. The weekly simulation uses a time step of 3 minutes, while the seasonal simulations were performed with 15 minutes time steps. Beneath, thermal conditions during the warm week in August are compared for Case 1 and Case 2, followed by the comparison of energy use from May 1 to October 1.

Thermal conditions

One concern for the projecting team was that high temperatures could occur in the upper parts of the atrium. Figure 7 shows the operative temperatures at the highest point of occupation in the atrium. Case 2 achieves a slightly better thermal comfort in the atrium, even though both cases will have some hours with unsatisfactory thermal conditions on the warmest days.

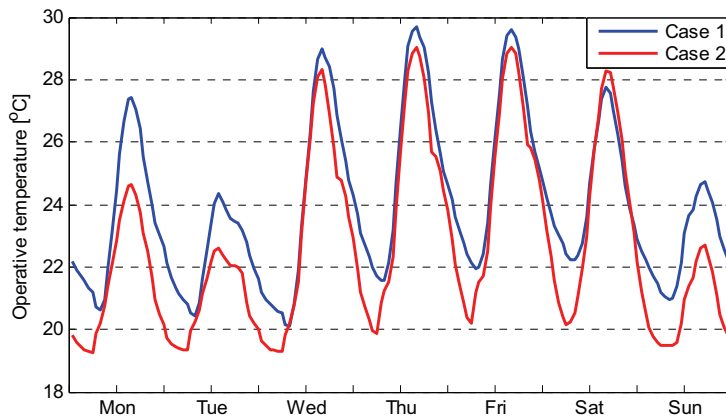


Figure 7. Operative temperature in the upper part of the atrium for the two cases. The temperature is at it's highest between 2 pm and 4 pm.

The most critical zone in the office wing is the 4th floor, since it is the one that is most exposed to the sun, and the effect of thermal buoyancy is less than the floors below. In addition, the temperature in the cavity of the double skin facade is highest outside the 4th floor, which will also affect the efficiency of the natural ventilation. Figure 8 shows that both cases will achieve satisfactory thermal conditions within the working hours. However, the temperature raises almost 2 degrees as the mechanical ventilation and cooling shuts down at 8 pm in Case 2.

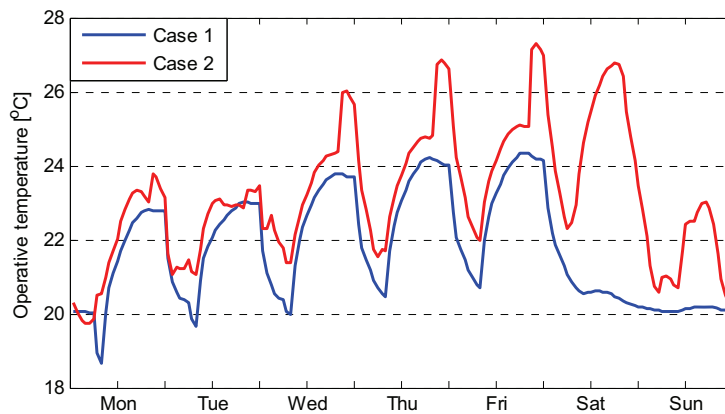


Figure 8. Operative temperature in the 4th floor of the office wing

Figure 9 shows the ventilation rate for the two cases during the case week. While the ventilation system in Case 2 rarely runs outside working hours, the mechanical ventilation must be in operation practically day and night in order to cool the building mass and achieve the thermal condition shown in previous figures.

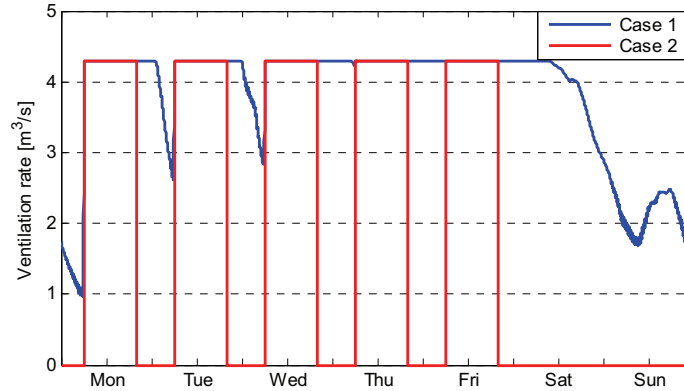


Figure 9. Total ventilation rates for the two cases.

Case 1 has the possibility to open fire hatches on warm days in order to ventilate naturally. However, as Figure 10 shows, the hatch in the reception only opens for short periods, and never fully open as this occurs at 25°C. Natural ventilation will thus be very limited even though the hatches in the ceiling are fully open. This is the reason for proposing some additional hatches higher on the south façade in the design aiming to utilize natural ventilation more actively.

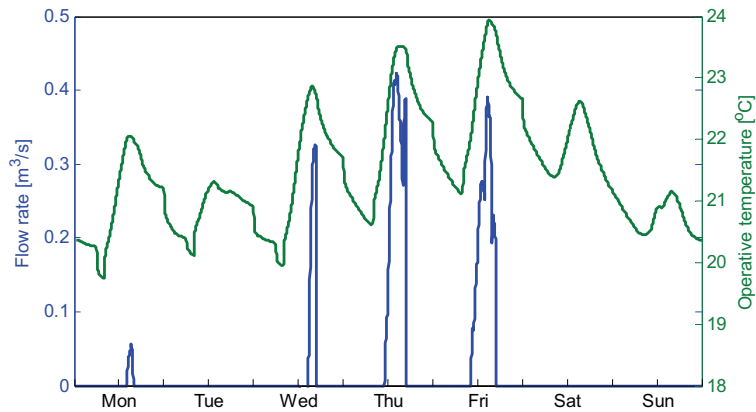


Figure 10. Temperature in the reception and the flow rate through the fire hatch for case 1.

Energy use

The seasonal energy use is estimated for the two cases. The energy use associated with the mechanical ventilation system is electric power to fans and to cooling machinery, which includes power to the compressor, pumps and dry coolers. Assuming that the efficiency of the ventilation system η_{tot} is constant for all air flow rates, the relation $P \sim Q^3$ can be used, where P is the total of all fan power measured as power input to the fan engine (kW) and Q is the total mechanical air flow (m^3/s). Thus, the following relation can be obtained:

$$\frac{P}{Q^3} = \frac{P_{nom}}{Q_{nom}^3} \Rightarrow P = P_{nom} \cdot \left(\frac{Q}{Q_{nom}} \right)^3 \quad (1)$$

where the subscript *nom* indicates nominal air flow rate.

By definition, the specific fan power (*SFP*) is (Mysen 1999):

$$SFP = \frac{P_{nom}}{Q_{nom}} \quad (2)$$

Combining (1) and (2), and assuming that the *SFP* does not vary significantly with the air flow rate, the following approximation for the fan power can be obtained:

$$P = Q_{nom} \cdot SFP \cdot \left(\frac{Q}{Q_{nom}} \right)^3 \quad (3)$$

In Table 4, the energy calculations are summarized. To calculate the electric fan power need, equation (3) is used with the simulated flow rates for each time step and integrated over the entire time period.

Table 4. Summary of the seasonal energy use for the two cases.

	Cooling energy need ¹ [kWh]	Res. Electric energy use for cooling ² [kWh]	Electric fan energy need to ³ [kWh]	Total electric energy need [kWh]	Relative energy need to Case 1
Case 1	21 300	7100	30 400	37 500	0
Case 2	12 600	4200	13 800	18 000	-52 %

¹Energy needed to cool the ambient air to the supply temperature
²Assumed COP = 3.0 (pumps, compressors and cooling fans included)
³Utilizing eq.(3) and assumed SFP = 2.5

DISCUSSION AND CONCLUSIONS

This paper has shown how an office building with an atrium and a double-skin façade can be modeled in a building simulation program. The paper also gives an example on how advanced controlling of windows and hatches can be implemented to utilize natural ventilation, even though such a controller functions in principle do not exist in the simulation program.

Although there are several uncertainties attached to such a model, it can be very helpful in finding alternative solutions in the design process. Moreover, simulations can reveal design mistakes. An example of this is demonstrated in Figure 10, which clearly shows that the planned solution will not work as intended.

The simulations also show that when dealing with a building with high thermal mass, it is crucial to empty the thermal storages during nighttime. If not, heat will accumulate over warm periods and create a considerable thermal comfort problem.

The energy calculations are approximate. Both the mechanical ventilation efficiency and the *SFP* are dependent on the volume flow rate. For Case 2 the *SFP* is constant, since the mechanical ventilation is either on or off. This is also correct for Case 1 during working hours, but when the fans are running on part load, the *SFP* will change. However, the purpose of the energy calculations is to provide a rough

estimate of the potential energy savings from utilizing natural ventilation. Despite of the uncertainties, the study clearly indicates that important savings can be realized through relatively simple measures.

Conclusions

- The simulations show that both cases will achieve satisfactory thermal conditions during working hours of a warm and sunny week, except for some hours in the most exposed areas in the upper part of the atrium.
- The thermal conditions are not depreciated by switching from mechanical ventilation to natural ventilation at night. In fact, in this particular case, the thermal conditions are somewhat improved in some parts of the building when using natural ventilation.
- The simulation indicates that there is a major potential in saving energy by combining mechanical ventilation with utilization of natural ventilation.
- The electrical energy saving potential for the case building, located in Trondheim, Norway, is estimated at approximately 50% during the period from May 1 to October 1.
- The study reveals an important potential for improving building and HVAC system design, which underlines the importance of using simulation tools in the design process.

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**Paper IV: Building Simulation as an assisting tool in decision making Case study:
With or without a double-skin façade?**

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Abstract

Implementation of double-skin façades in buildings has been an object of broad application in the recent years. In this presentation, a planned office building in the city centre of Trondheim, Norway, is used as a case for considering whether a double-skin should be applied to the east façade in order to reduce the heating demand, thus making the double-skin façade a profitable investment. The building is modeled both with and without a double-skin façade with the building energy simulation program ESP-r. This paper describes how a double-skin façade with controllable windows and hatches for natural ventilation can be implemented in the simulation program. The simulation results indicate that the energy demand for heating is about 20 % higher for the single-skin façade with the basic window solution compared to the double-skin alternative. However, by switching to windows with an improved U-value in the single-skin alternative, the difference in energy demand is almost evened out. The number of hours with excessive temperatures is, in contrast to other studies on the subject, not significantly higher for the double-skin alternative. However, the predicted energy savings are not sufficient to make the application of a double-skin façade profitable.



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Building simulation as an assisting tool in decision making Case study: With or without a double-skin façade?

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Abstract

Implementation of double-skin façades in buildings has been an object of broad application in the recent years. In this presentation, a planned office building in the city-centre of Trondheim, Norway, is used as a case for considering whether a double-skin should be applied to the east façade in order to reduce the heating demand, thus making the double-skin façade a profitable investment. The building is modeled both with and without a double-skin façade with the building energy simulation program ESP-r. This paper describes how a double-skin façade with controllable windows and hatches for natural ventilation can be implemented in the simulation program. The simulation results indicate that the energy demand for heating is about 20% higher for the single-skin façade with the basic window solution compared to the double-skin alternative. However, by switching to windows with an improved *U*-value in the single-skin alternative, the difference in energy demand is almost evened out. The number of hours with excessive temperatures is, in contrast to other studies on the subject, not significantly higher for the double-skin alternative. However, the predicted energy savings are not sufficient to make the application of a double-skin façade profitable.

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Keywords: Double-skin façade; Natural ventilation; Modeling; ESP-r

1. Introduction

Implementation of double-skin façades in both new and existing buildings has seen broad application in recent years. There are both technical and architectural arguments for investing in a double-skin façade. The architectural arguments are that double-skin façades provide open and transparent façades as well as a homogenous façade expression (mono material) [1]. The technical arguments relate to the practical functions of a double façade, both concerning thermal performance and as shelter against outdoor climate. The double façade may reduce the heating demand by functioning as a thermal buffer and as a pre-heater of ventilation air. It may also improve the thermal comfort during cold winter days since the inner window surfaces will be warmer, thus increasing the operative temperature. In addition, the double-skin façade protects sun-shading devices from dirt and adverse weather

conditions, and it may allow natural ventilation in places, this generally would not be possible due to high outdoor noise levels [2].

Unfortunately, there are several disadvantages related to double-skin façades. The investment costs are considerably higher than for a traditional single-façade (60–80%), even though the inner façade may have a simpler construction and finish [3]. The additional glass layer reduces the daylight illumination levels indoor, and gives a larger glass area to keep clean. Thermally, the risk of overheating on warm sunny days is evident and may lead to a higher cooling demand. However, with well-dimensioned openings, an optimized space between the façades, and well-positioned shading devices, it is possible to reduce the overheating risk to a minimum [2].

In literature, several papers describe how double-skin facades should work to improve a building's energy efficiency. However, Gertis [4] underlines that only few simulations have been made and few measurements are available to support the claimed benefits of double-skin façades. An example of measurements performed on buildings with double-skin facades is Pasquay's study [5] on three buildings with

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double-façades in Germany (Nordrhein-Westfalen). He concludes that energy may be saved in specific locations compared to conventional solutions with full air-conditioning. However, the author points out that double-skin façades are not the best choice for every building in every location, and that every building must be considered independently. Stec and van Paassen [8] have compared the performance of nine different façade systems for dutch climate and conclude that the double-skin systems are competitive in energy performance, and underline the importance of treating the double-skin as an integrated part of the HVAC system. Saelens et al. [9] claim that most double-skin facades are incapable of lowering both the annual heating and cooling demand, and that “only by combining typologies or changing the system settings according to the particular situation, a substantial overall improvement over the traditional insulated glazing unit with exterior shading is possible”.

Guidelines or recommendations for modeling double-skin façades are practically non-existent and, as mentioned, even examples are rare. However, some work on this issue is done by Dickson [6] and Hensen et al. [7]. Both studies point out that the modeling of double-skin façades is particularly complex because of the high degree of interactions between the many heat transfer processes. The study of Saelens et al. [9] concludes with the necessity to perform whole building energy simulations in order to evaluate the performance of double-skin façades.

This paper focus on an office building that is being planned in Trondheim, a city in the middle of Norway. The objective is to compare the building’s energy performance with and without a double-skin on the east façade.

2. Methods

A model of the building design is implemented in the simulation program ESP-r, which is a transient simulation program based on the finite volume technique. ESP-r is capable of modeling the energy and fluid flows within combined building and plant systems when constrained by control actions and subject to dynamically varying boundary conditions [10].

The method used in this paper is to model the building with and without a double-skin on the east façade, and then compare the energy demand and thermal environment for the two alternatives.

2.1. The case building

The new office building will be raised in the city centre of Trondheim, Norway (63°N, 10°E). The building will be integrated with existing brick buildings. The new building consists of an atrium and an office wing, both five storeys tall. The east façade is considered built with a double-skin. This façade is 30 m long and 10 m high, and has a window-to-wall ratio of 80%. The ground floor in the atrium is planned as a reception hall and the first floor is a common area. Further up the atrium there are balconies with working places and space for group activities. The office wing has both cell offices and

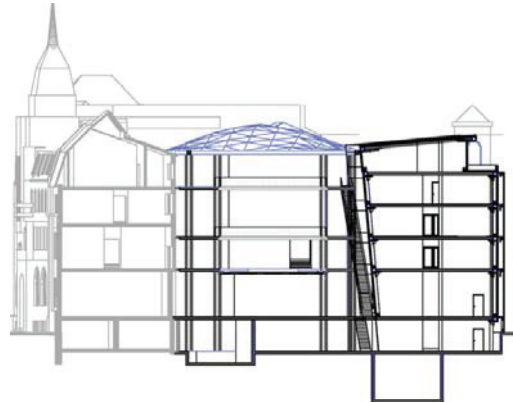


Fig. 1. Cross-section of the case building viewed from the south with the double façade drawn in. Light gray lines represent existing buildings.

open-plan offices distributed on all five floors. There are no walls significantly obstructing the airflow between the office wing and the atrium. In total there are about 150 and 100 working places in the office wing and the atrium, respectively. Fig. 1 shows the cross-section of the building viewed from the south with the atrium in the middle, and the office wing to the right.

In this study, only the atrium and the second, third, and fourth floor of the office wing are modeled (about 2000 m² of floor surface area in total). The reason is that the remaining part of the building is designed with a separate mechanical ventilation system. Fig. 2 shows how the simulation model appears in the ESP-r cad-window.

2.1.1. Construction properties

Properties of some selected building materials are presented in Table 1. The optical data are calculated by WIS, which is software that determines the thermal and solar characteristics of window systems [11]. The U -values for the windows are given by the manufacturer, and the wall’s U -value is according to the standard building code for Norway [12]. The floors in the office wing are suspended concrete floors with a wooden finish. The

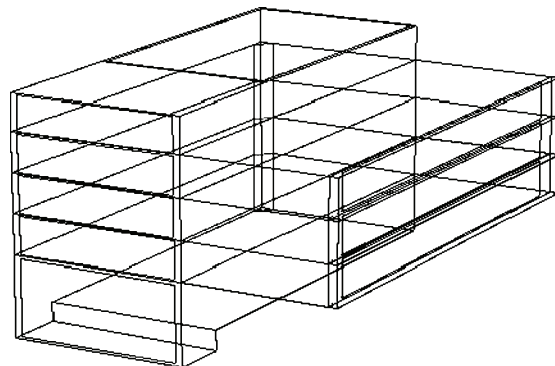


Fig. 2. Wireframe of the simulation model in the ESP-r cad window.

Table 1
Properties for selected building constructions

	Construction	U-value (W/(m ² K))	Transmittance (g-value)
Double façade	Windows facing the cavity in the double façade	1.40	0.578
	External glazing in the double façade without blinds	5.10	0.541
	External glazing in the double façade with blinds angled 45°	3.00	0.320
Single façade	Windows on east façade without blinds	1.40	0.578
	Windows on east façade with blinds angled 45°	1.40	0.177
Common for both	Glass roof in the atrium with semi-transparent canvas	1.40	0.212
	Glass façade in the south façade	1.40	0.304
	External walls	0.22	–

concrete in the ceiling is exposed. The external walls of the existing buildings facing the interior space of the new building are made of brick.

2.1.2. Ventilation

The mechanical ventilation rates are set to meet the Norwegian codes and guidelines [12], which is 7 l/s per person and an additional 2 l/(s m²) for emissions from unknown or high-emitting building materials, and corresponds to IDA 1 (Indoor Air Class 1) in the EN-13779 [13]. The supply air temperature is 19 °C during winter months (November–February), 17 °C during spring/fall (March–April/September–October) and 15 °C in the summer (May–August). The operating time for the mechanical ventilation is from 6 a.m. to 6 p.m.

The building is planned without local cooling equipment. It is assumed that the cooled supply air and the exposed concrete in the ceiling in the office wing and the heavy mass walls in the atrium will absorb the excessive heat during the working day and keep the temperature within acceptable limits. During the night, heat is removed from the building by means of natural ventilation through windows and hatches [14]. In addition, the mechanical ventilation supports the natural pre-cooling of the building in the morning hours. The operation strategy is given in Table 2. Initial simulations indicate that the office air temperatures increase averagely about 4 K in case of a double-skin façade, and about 5 K in case of single façade on warm summer days. This means that in order to avoid temperatures above 26 °C, the operative temperature should not exceed

22 °C in case of a double-skin, and 21 °C in case of a single façade at the start of the working day.

2.1.3. Heat gains

Internal heat gains due to occupancy, lighting and office equipment have been set according to ASHRAE guidelines [15], and the occupancy pattern is based on the user profile for typical office buildings predefined in ESP-r. Fig. 3 shows the internal heat gains. The internal gains are valid weekdays, and turned off during weekends.

2.2. Climate data

In this study, semi-synthetic meteorological data provided by Meteororm [16] for Trondheim in 2003 was used. The annual average temperature of this climate year is 5.7 °C.

2.3. Simulation model

The following assumptions and simplifications have been done in the simulation model:

- The base area is modeled as rectangular for both the atrium and the office wing, although the geometric shape in reality is trapezoidal. However, the floor and external wall surface areas are in accordance with the drawings.

Table 2
Operation set-points for the mechanical and natural ventilation

Time	Mechanical ventilation		Window operation ^a		Hatches ^b
	Double façade	Single façade	Double façade	Single façade	
0–3	Off	Off	20–22	20–22	20–22
3–6	22–23	21–22	20–22	20–22	20–22
6–18	100%	100%	21–24	Shut	21–24
18–24	Off	Off	20–22	20–22	20–22

The intervals are operative temperature in the respective air-conditioned zone, and the openings open 0–100% within the intervals.

^a The operable windows in the office wing.

^b Hatches on the atrium's south façade. Equal strategy for both alternatives.

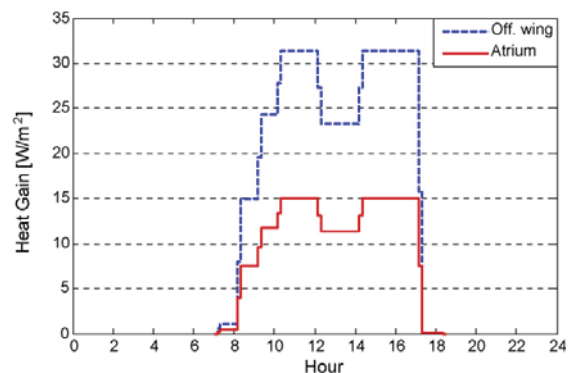


Fig. 3. Internal heat gains used in the simulations.

- The atrium is divided horizontally in three zones. Although this is a rough approximation, it makes it possible to take thermal stratification into account, and improves the accuracy of the airflows through the atrium.
- For the alternative with double-skin façade, venetian blinds are placed within the cavity. When the global radiation on the façade exceeds 200 W/m^2 , the blinds will shut. For the single-skin alternative, there are external blinds with the same control strategy. The blinds are individually controlled for each floor.
- The solar obstruction caused by the high-rise buildings situated to the east is taken into account, while the potential effect from the low-rise buildings situated to the south is ignored.
- Zone set-points for heating are $21 \text{ }^\circ\text{C}$ the year around, with a set-back of 2 K outside working hours.
- The heat recovery efficiency is set to 75%.

2.4. Modeling the double-skin façade

A multi-storey double-skin façade is chosen for this building, i.e. there exist no horizontal or vertical partitioning between outer and inner façade. The cavity is ventilated via openings in the bottom and the top of the façade.

To obtain a tolerably accurate prediction of the façade performance, it is important to ensure appropriate treatment of the solar insolation, the cavity convection regimes, the surface view-factors, airflow resistances, vertical temperature gradient and cavity divisions with fictitious divisions [6]. Fig. 4 shows the heat transfer mechanisms and airflow paths involved.

For this particular case, the double-skin façade is divided into a stack of three zones adjacent to each of the office levels. The zones are divided by fictitious transparent surfaces with high conductivity, negligible thermal mass and high emissivity, and coupled by an airflow network which also includes the inlet opening at the bottom and the top outlet opening at the top of the façade. Since the double-skin façade will be ventilated, the

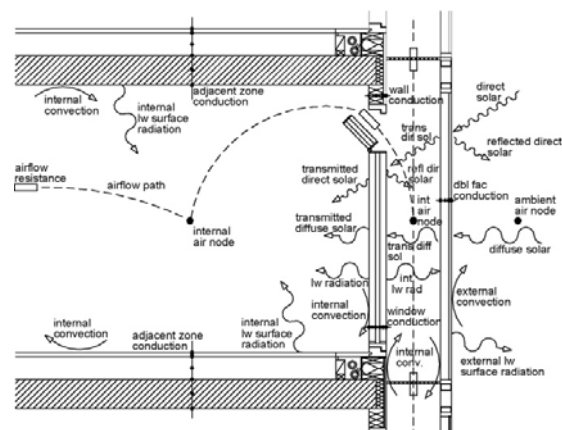


Fig. 4. Cross-section of the double-skin façade and the adjacent office floor with the heat transfers mechanisms and the airflow network.

Bar Cohen & Rohsenow correlation, proposed by Dickson [6], is used to predict the convective heat transfer for the surfaces facing the cavity when it is open. When the cavity is closed, the default Alamdari & Hammond correlation is used.

2.4.1. Operating the natural ventilation and the double-skin façade

The operation of the windows depends on both the temperature in the office and in the cavity of the double-skin façade. That is, if the indoor temperature reaches the low set-point, the window opening area increases proportionally with the indoor temperature until the window is fully open at the high set-point, according to the set-points given in Table 2. If the temperature in the double-skin façade exceeds $25 \text{ }^\circ\text{C}$, the windows close in order to avoid increased cooling demand. This is not the optimal strategy, since the temperature in the cavity may be less than $25 \text{ }^\circ\text{C}$ but higher than the office temperature, thus increasing the cooling load. However, in ESP-r there are currently no control functions that are able to sense two parameters at the same time. Since the operation of the windows depends on the temperature in the office as well as in the cavity of the double-skin façade, it is necessary to introduce a dummy air node in the simulation model. This makes it possible to have two openings between the indoor air node and the node in the double-skin façade. One opening represents the actual window, and the other is an opening with negligible fluid resistance when open. When the temperature in the double-skin façade exceeds $25 \text{ }^\circ\text{C}$, the latter opening shuts in order to prevent the hot air from entering the offices. Fig. 5 illustrates how this is implemented for the window openings on the third floor. Damper A represents the window, and damper B is the fictitious opening. The control strategy is in principle also implemented for the windows in the single-skin alternative and hatches facing the ambient, in which case damper B in Fig. 5 is controlled by the ambient temperature.

The study of Gratia and Herde [2] gives recommendations for how a south-facing double façade should be operated in order to achieve the best energy performance for various weather conditions. In the present case however, the double

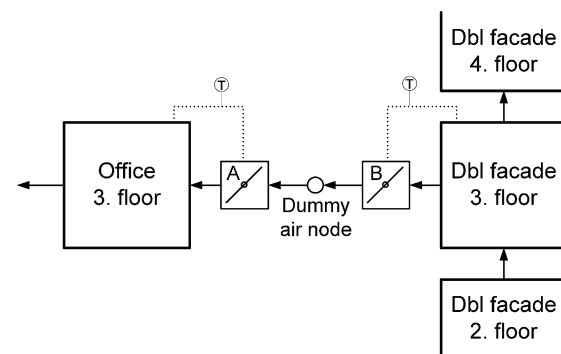


Fig. 5. Principal sketch of the window control. Damper A represents the window and is controlled by the indoor temperature and damper B is controlled by the temperature in the double-skin façade.

Table 3
For comparison, the overall U -values for the east façade for the four alternatives calculated from the method described in ISO 6946 [17]

	Alternatives	U -value ($W/(m^2 K)$)
Double façade	1 ^a	0.94
	2 ^a	0.94
Single façade	3	1.16
	4	0.84

^a Air cavity closed.

façade is facing east, and the sun is often obstructed by neighboring high-rise buildings. Whether the cavity should be ventilated or not, is determined by the air temperature in the double-façade cavity and whether there is a space heating demand. Therefore, the chosen control strategy for the façade openings is rather simple; in the summer time, the openings are always open. Else, the hatches open when the air temperature in the cavity exceeds 20 °C to avoid overheating.

3. Simulations

Four different alternatives for the east façade are considered:

1. Double-skin façade and the double façade's cavity used as a supply air duct for passive pre-heating of the supply air.
2. Double-skin façade without pre-heating of the supply air.
3. Conventional single-skin façade, mechanical ventilation only during working hours.
4. Conventional single-skin but with windows with improved U -value, mechanical ventilation only during working hours.

See Table 3 for comparison of the U -value of the east façade.

4. Results

4.1. Energy performance

Fig. 6 shows the simulated monthly energy use for space heating and pre-heating and pre-cooling of the ventilation air for the four considered alternatives.

An interesting observation is that Alternative 4 has the lowest heating energy demand during the winter months (November–February), but significantly higher during the transitional months. This shows the advantage the double-skin façade has; benefiting of the sun to reduce the heating

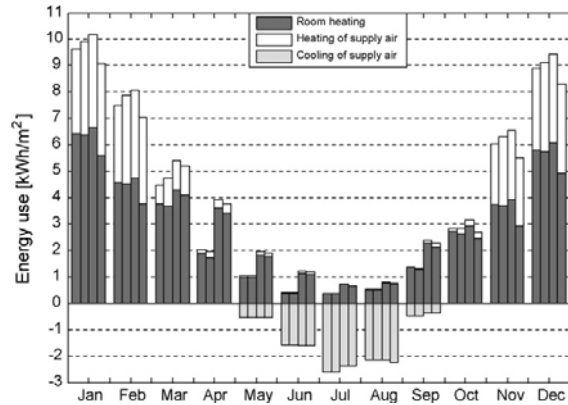


Fig. 6. Room and ventilation heating demand distributed monthly. First bar for each column is Alternative 1 and second is Alternative 2 and so on.

demand. During the winter months, however, the solar gains on the east-facing façade will be very low since the sun then will be severely obstructed by the high-rise building to the east.

Table 4 shows the annual energy use for space heating, and ventilation air heating and cooling. The table shows that there is practically no difference in the alternatives when it comes to cooling, but as expected, Alternative 1 has the lowest overall heating energy demand. Although the cooling energy demand is about the same for the four alternatives, the number of hours with over-heating in the office wing is less for the alternatives with the double-skin façade. Thus, the natural ventilation during working hours will improve the thermal environment slightly, but in practice, the four alternatives are equal in this matter.

4.2. Economy

In the end, it will be the investment and operational costs that will decide which alternative the developers choose for the east façade. The construction and maintenance cost of a double-skin façade system is not very often described in the existing literature, and when it is, the opinions are contradictory in reports from different authors [18]. Undoubtedly, the construction cost of a double-skin façade is higher than for a single-skin one. In Central Europe the constructional costs per square meter façade are in the order of [18]:

Table 4
Simulated annual heating demand and number of hours with excessive temperatures

Alternatives	Space heating energy (kWh/m^2)	Supply air heating energy (kWh/m^2)	Total (kWh/m^2)	Relative difference from alternative 1	Hours $T_{op} > 26$ °C	
					Office fourth floor	Atrium level 4
1	28.4	11.8	40.2	–	0	67
2	27.8	13.3	41.1	2.4	0	67
3	34.2	13.7	47.9	19.1	5	47
4	29.1	13.5	42.7	6.2	11	53

Table 5
Annual costs per floor area

Cost	Cost for each alternative (€/m ²)			
	1	2	3	4
Façade ^a	8.03	7.81	4.46	4.91
Maintenance ^b	4.00	4.00	4.10	4.10
Energy ^c	3.52	3.60	4.19	3.73
Total	15.55	15.41	12.75	12.74

^a From Ref. [18] and assumed interest rate at 7% and life time is set to 35 years.

^b From Ref. [19].

^c Assumed energy price 0.0875 €/kWh.

- Single-skin façade 300–500 €/m²;
- Standard double-skin 600–800 €/m²;
- Double-skin with adjustable air in and outlet 700–1000 €/m²;
- Double-skin with open able exterior hatches 800–1300 €/m².

Table 5 shows the annual cost estimates for the four alternatives for the case building. The investment costs are conservatively chosen from the list above. For the maintenance cost it is assumed that outer façades are cleaned twice a year and the surfaces facing the cavity once a year. The sun shading is cleaned every second year for the single-skin and every fourth year for the blinds in double-skin façade. Even if it would be necessary to clean an external shading every year, this is rarely realized in practice [19]. The period of calculation is 35 years, the energy price is set to 0.0875 €/kWh, and the interest rate is assumed to be 7%.

5. Discussion

An interesting observation from the results is that even though the total demand for heating energy is lower for Alternative 1 with respect to Alternative 2, the energy use for space heating is higher for Alternative 1. When using air from the double-skin cavity the climate buffer is somewhat impaired and the conduction through the inner façade increases. Consequently, the additional heat loss from the room must be taken into account when calculating the achievement by using the double-skin cavity as an air intake duct, especially in months with limited solar radiation.

The difference in energy use for heating the supply air between Alternative 1 and Alternative 2 is somewhat less than one could expect. One reason for this may be the relatively high-efficient heat recovery unit used. With a less efficient heat recovery unit, the difference would be larger. Another reason is that the sun is very low in the horizon during the winter months and the sun is also obstructed by the high-rise buildings in the early mornings. Another interesting observation is that for some months, such as April, the total heating energy demand is actually higher for Alternative 1 compared to Alternative 2. The reason is the fact that heat gain in the cavity is not utilized, since most of the pre-heating energy demand is covered by the heat exchanger. In this case, the intake air should have been bypassed the cavity, since maintaining the thermal buffer is most energy efficient in this case. This implies that

an optimal operation strategy must be considered for this configuration.

This study is conducted on a building with relatively high standard of thermal insulation. Clearly, a building of poorer standard would achieve higher energy savings by adding a double-skin façade.

The study shows, in contrast to other studies on this subject, that by applying a double-skin façade the heating energy demand is decreased significantly (by 16%) compared to a conventional solution, without an increased cooling demand. In fact, the thermal condition in the office wing is slightly improved. However, by replacing the windows in the single-skin alternative with windows with a better *U*-value (Alternative 4), the achieved energy saving is nearly evened out.

The reflective glazing used in the south façade of the atrium has a *g*-value close to 0.3. This glass type could also be used in both the single- and double-skin alternatives and may eliminate the need for external sun shading. The reflective glass type would not remove glare completely as venetian blinds do, but combined with internal blinds or drapes this problem should be minimized. Although glass with reflective coating is slightly more expensive, exterior sun shading may be omitted, thus reducing investment costs, and also reducing cleaning and maintenance cost. However, in this case, using reflective glass also has some disadvantages. Firstly, reflective glazing reduces the solar insolation permanently, thus the possibility to utilize passive solar heat is reduced. This may be significant for buildings located in heating dominated climates. In particular, it is important when the cavity in the double-skin façade is used as a pre-heater of ventilation air (Alternative 1). Secondly, reflective glass coatings may cause increased lighting requirements and consequently an indirect cooling load attributable to the lighting. Based on these latter arguments, reflective glazing was not found to be suitable for this type of building in this climate.

From an economic point of view, it is difficult to recommend the double-skin façade, even with the documented energy savings because of the considerable higher investment cost. The annual cost estimates done in this study are also relatively conservative. That is, the investment costs of the double-skin façade are rather higher than lower in practice. If a double-skin façade is chosen despite these facts, it must be out of architectural and aesthetic reasons.

6. Conclusions

- This study shows, in contrast to some other studies on this subject, that application of a double-skin façade decreases the heating energy demand, without significantly increasing the number of hours with excessive temperatures.
- The simulation results indicate that the heating energy demand is about 20% higher for the single-skin façade with the basic window solution compared to the double-skin alternative.
- By using the cavity of the double-skin as a pre-heater for the supply air, further savings may be achieved. However, the simulations show that optimal strategies must be found in

order to avoid increased heat loss (from the offices), which cannot be utilized in the pre-heating of the supply air.

- By replacing the basic windows with windows with an improved U -value, the difference in heating demand between the double-skin and single-skin alternatives are almost evened out on an annual basis. This alternative is about as cost-efficient as the basic alternative in a 35-year perspective.
- If a double-skin façade is chosen, this must be done for other reasons than economy. From an economic point of view, the energy savings will not defend the additional costs the double-skin façade constitute.

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Paper V: The effect of suspended ceilings on energy performance and thermal comfort

Rasmus Z. Høseggen, Hans M. Mathisen and Sten O. Hanssen

Abstract

The objective of this study is to determine the potential energy savings and thermal comfort benefits of exposing concrete in the ceiling to the indoor air as an alternative to suspended ceilings. The performances of the two were assessed by monitoring the room air and surface temperatures in an office building in operation. There was also simulation of different scenarios with a calibrated building simulation model. In this study, it is shown that ESP-r is capable of simulating an advanced controlled office building in operation with good agreement with the measurements. The results presented in this paper indicate 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. However, by removing the suspended ceiling, only minor annual heating energy savings are achieved.

(Submitted for review)

The effect of suspended ceilings on energy performance and thermal comfort

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Abstract

The objective of this study is to determine the potential energy savings and thermal comfort benefits of exposing concrete in the ceiling to the indoor air as an alternative to suspended ceiling. The performances were assessed through monitoring of room air and surface temperatures in an office building in operation, and simulation of different scenarios with a calibrated building simulation model. In this study, it is shown that ESP-r is capable of simulating an advanced controlled office building in operation with good agreement with the measurements. The results presented in this paper indicate that exposed concrete in the ceiling both reduces the number of hours with excessive temperatures considerably and create a better and more stable thermal environment during the working day. Also, exposed concrete increases the achievements of utilizing night free cooling significantly. However, by removing the suspended ceiling, only minor annual heating energy savings are achieved.

Keywords: energy demand, thermal comfort, thermal mass, exposed concrete, suspended ceiling, measurements, free cooling, ESP-r

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1. Introduction

In the literature, several studies have evaluated the effects of thermal mass on energy use and thermal comfort, both parametrically and experimentally. However, there are only few studies that both assess the potential heating and cooling load benefits of exposed thermal mass in office buildings. Moreover, the experimental studies found are solely on test cells, and they are not taking user behavior and advanced building control systems into account. The objective of this study is to determine the potential energy savings and thermal comfort benefits of exposing the concrete ceiling to the indoor air as an alternative to the original suspended ceiling. This is assessed through monitoring of room air and surface temperatures in an office building in operation and simulation of different scenarios with a calibrated building simulation model.

As a consequence of the Norwegian partnership in the EEC, Norway is obliged to implement the EU Energy Performance of Buildings Directive (EPBD) [1] in the national laws and regulations. Thus, the new building codes and guidelines are also revised. The former regulations only set requirements for U-values and air tightness. The new regulations introduce an *energy frame* for different building categories. If the net energy demand calculated according to the methodology established in the new Norwegian Standard prNS3031 [2] is within the frame, regulations are satisfied. Since the frame is based on net specific energy demand per year, the efficiencies of the energy systems are not taken into account. This means that for example the coefficient of performance of a highly efficient mechanical cooling system is not rewarded. However, passive measures that reduce the net cooling demand will contribute to satisfy the energy frame. This has led to a renewed interest in utilizing passive measures to decrease the total energy use in buildings.

Thermal mass can give a positive contribution to the indoor environment and buildings' energy performance, both summer and winter. In the summer time, excessive heat is absorbed and reduces the need for cooling during the day-time. The absorbed heat will gradually be released when the temperature decreases during the night. Buildings that are unoccupied during the evening and night may be cooled down, in order to empty the thermal storages and heat may be absorbed the following day. In the winter

time, energy from the sun and internal heat gains can be absorbed in the thermal mass during the day, and gradually released to the indoor air at night, thus completely or partially reduce the need for heating.

Several studies have shown that thermal mass together with night ventilation may reduce the indoor maximum temperature and the cooling energy demand. Thermal mass combined with night ventilation may reduce the maximum indoor temperature by 2-6 K (e.g. Givoni [3] and Shaviv [4]), provided that the diurnal outdoor temperature swing is adequate. Dependent on the climate and building type, the cooling energy savings found in the literature span from 5 % to 36 % (e.g. Burch [5] and Ruud [6]). Moreover, some studies (e.g. Kolokotroni [7] and Gratia [8]) conclude that if the heat gains are not too excessive in office buildings, thermal mass and night ventilation should be sufficient in order to cover the cooling demand alone in moderate climates. A recent study by Artmann et al [9] assesses the climatic potential for passive cooling of buildings by night-time ventilation. The study concludes that the whole Northern Europe, and in particular the British Isles and Scandinavia, has a very high potential for night-time ventilative cooling.

Some studies also conclude that thermally heavy buildings have lower space heating energy demand than light buildings. According to Norèn et al [10] and Bellamy et al [11], heavy residential buildings demand about 15 % less space heating energy compared to equivalent light buildings. Ståhl [12] has estimated the energy savings to be about 20 % for offices. Heating energy savings are most significant in the intermediate seasons in cold climates and in climates where the building's balance temperature is close to the mean outdoor temperature [13] [14].

However, there are several obstacles preventing use of thermal mass in office buildings. A survey among 750 corporate managers in the Nordic countries reveals that every third leader plan to reorganize or rebuild the office plan within the next two years [15]. Hence, modern office buildings should have a high degree of adaptability to meet varying requirements. Clearly, use of heavy materials in partition walls may come in severe conflict with the desire of having a flexible building. This, together with building materials' heat capacity, makes external walls, floors, and ceilings the most common alternatives for use and possible exposure of thermal mass.

As Shaviv [4] and Norén [10] conclude on thermal mass' ability to reduce energy demand for cooling and heating respectively – the greatest effect of thermal mass is going from a light structure to a medium heavy structure. The effect is less significant by a further increase of thermal mass. Thus, the focus in this paper is to assess the potential benefits of increasing the thermal mass in an office cell by exposing concrete in the ceiling compared to having a suspended false ceiling.

2. Method

To investigate the effect of exposing concrete in the ceiling, six identical office cells were studied from which three of the suspended ceilings were removed. The purpose was to monitor the room parameters and compare the overall energy and thermal comfort performance. Further, the measurements were used to calibrate a detailed simulation model in ESP-r [16] to compare the energy use for the whole building.

3. Building description

The case used in this study is an office building at the Nord-Trøndelag College (HiNT) campus in Levanger (63.75°N), located 80 km north of Trondheim, Norway. The building, which from this point will be referred to as Røstad, is located in rural surroundings in a coastal climate. The building was ready for occupation in August 2002, and has a common wing with meeting rooms and educational areas. Two other wings are office areas, see Figure 1 and Figure 2. In this paper only one of the office wings (the TKS wing) is studied. The two storey building has no basement, but a culvert for supply of ventilation air is embedded in the ground along the central axis of the wing.

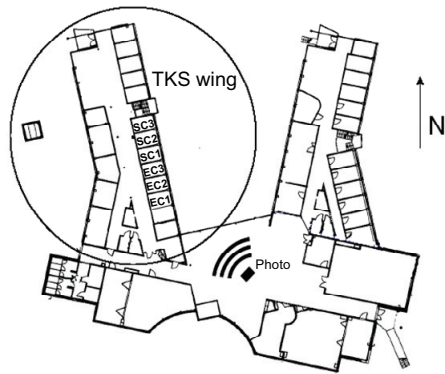


Figure 1 Plan view of the second floor showing the common area and the two office wings. The point labeled “Photo” is from where the photo in Figure 2 is taken.



Figure 2 The TKS wing viewed from south-east, showing the air exhaust tower and the glazed staircase.

3.1 HVAC system and control

The hybrid ventilation is of so called culvert type. In principle it is constructed as shown in Figure 3. The ducts from the culvert to the rooms are buried in the ground beneath the floor. At the façade the ducts turn 90° upwards. The ducts end in a damper placed inside the supply air terminal device. The air diffusers are placed at the floor beneath the windows.

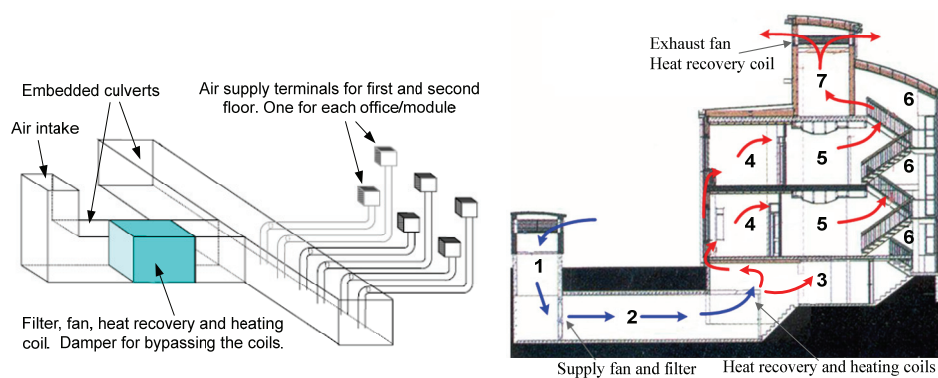


Figure 3 Principle of the culvert and a section through the building. 1: Air intake tower. 2: Air intake culvert. 3: Air distribution culvert. 4: Offices. 5: Corridors. 6: Stairway. 7: Exhaust air tower.

From the offices the air flows through grilles placed close to the ceiling and into the corridor. Exhaust of air takes place through corridors and stairway up to the tower on the roof. The exhaust air tower contains a heat recovery coil and a fan.

Figure 4 shows the principle of the ventilation and controller system. The supply and exhaust fans are controlled by the pressure difference between the culvert and corridor at the second floor, i.e. the fans keep the pressure difference constant. When there is no heating demand for the ventilation air and the outdoor temperature exceeds 15 °C, the bypass dampers opens to reduce the pressure drop.

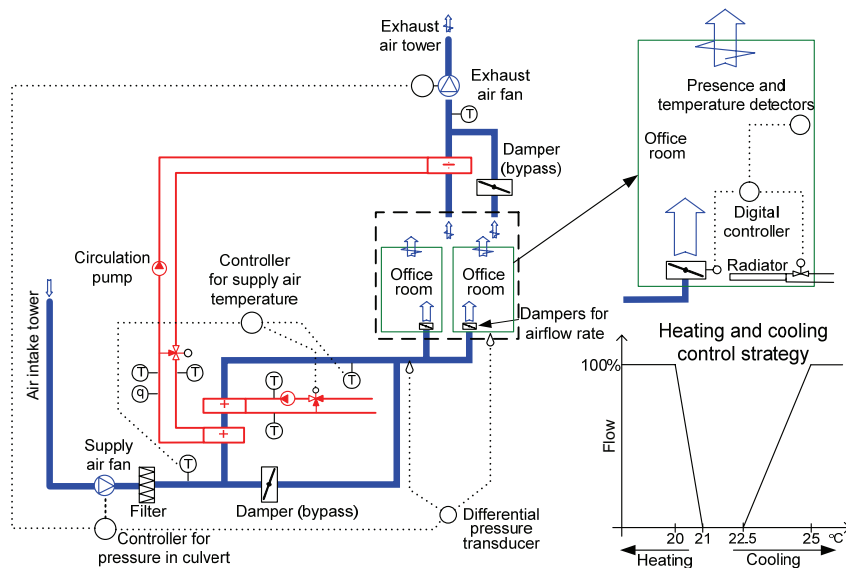


Figure 4 Principle of the ventilation system and the control strategy for the room controller. The set-point in each office cell is individually controlled, and is in this case 21°C.

The dampers in the supply air terminal devices control the air flow rate to each room. They operate as follows: Normally they close at 4pm. At 6am the dampers open to give approximately 25 m³/h (3.3 m³/h m² ~ 1.2 ach) of air. When a person enters the room they open to supply 43 m³/h (5.7 m³/h m² ~ 2 ach). If the room air temperature exceeds the set-point for cooling (SP+1.5K) they open further. The dampers will continue to open proportionally with the temperature until they reach full opening at 25°C,

supplying the rooms with about 200 m³/h (26.7 m³/h m² ~ 9 ach). If the room air temperature is above the cooling set-point, the dampers also open at night, provided that the outdoor temperature is above 15 °C. Further discussion of this control strategy can be found in [17].

3.2 Office rooms

Each office has a presence detector, a temperature sensor and a digital controller unit that controls heating, ventilation and lighting. A room is registered as empty if the detector has not detected any movement for five minutes. To avoid the ventilation and lighting to turn on and off for short absences, a 20 minute time delay is implemented for these controls. This means that a person must be absent more than 20 minutes before the lights switch off, the set-point change and the ventilation tunes down.

The six rooms of special interest in this study (Room EC1 to EC3 and SC1 to SC3, see Figure 1) are located on the second floor at the east-north-east side of the building. From this point on, the rooms with the original suspended ceiling will be referred to as SC and the rooms with exposed concrete in the ceiling as EC. The rooms are 7.5 m² and equally equipped with a laptop, an external LCD-screen, are similarly furnished and have the same heating and lighting equipment. The office rooms have painted plaster internal walls and glazed wall to the hallway. The floors are concrete with a linoleum covering. All rooms originally have suspended ceilings of painted plaster boards with 50 mm insulation above, which for Room EC1 to EC3 are removed.

4. Measurements

The six cell offices involved in the experimental study were monitored during the period from October 1st 2005 to October 15th 2006. Room air temperature, air supply damper positions, radiator valve positions and room set-points were registered every 15 minutes by the building energy management system (BEMS). The occupancy of every room in the building was also logged. In addition, Room EC2 and Room SC2 were equipped with six calibrated thermocouples connected to a logger, to get a more detailed picture of the air temperature stratification, air supply temperature and surface temperatures. The reason why six offices were involved in the study was to make the rooms adjacent to EC2 and SC2 similar, thus minimizing boundary differences.

4.1 Criteria for data evaluation

To be able to compare the performance of Room EC2 and Room SC2 it was crucial that both rooms were approximately similarly used for periods of at least three days in a row in order to take the effect of heat storage into account. This is of great importance since the set-point for heating and ventilation is dependent on whether people are present or not. In addition, heat gains from lighting and people will also influence the room temperature considerably. During the entire year there were only 10 periods that more or less fulfilled these criteria.

4.2 Measurement results

Presented below are the results of the measurements from selected periods for each season where the criteria presented in the previous chapter are fulfilled as close as possible. For every period the room temperatures are plotted for comparison. However, since the HVAC-system always will try to even out any deviation from the set-point, the damper positions, which directly affect the air flows, are also plotted. In addition, the set-point and the ventilation rate are (or may be) dependent on the presence of people. Therefore, the rather complex picture needs four plots for comparison – air temperature, damper position, actual occupancy and the recorded occupancy added 20 minute delay.

Fall

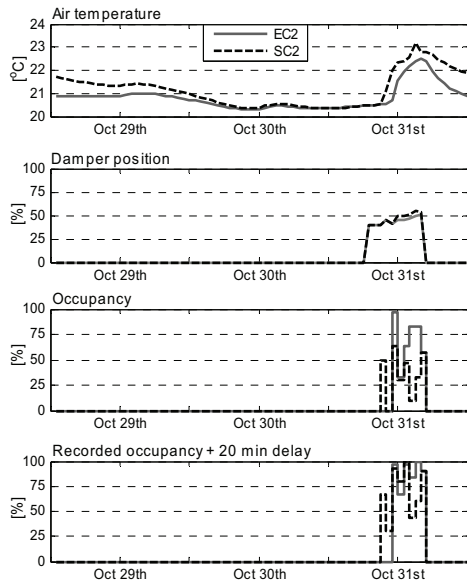


Figure 5 Saturday 29th to Monday 31st October 2005

Winter

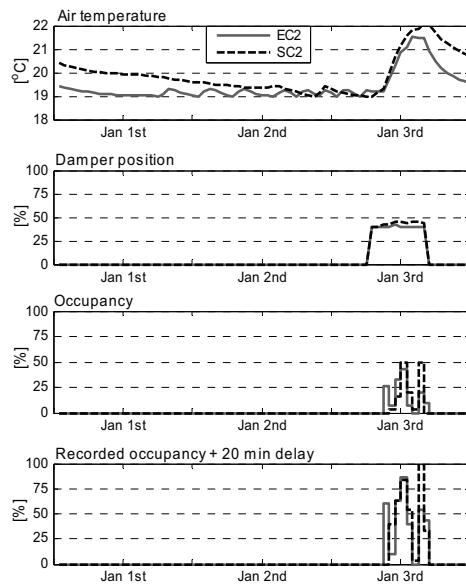


Figure 6 Saturday December 31st 2005 to Monday January 2nd 2006

Spring

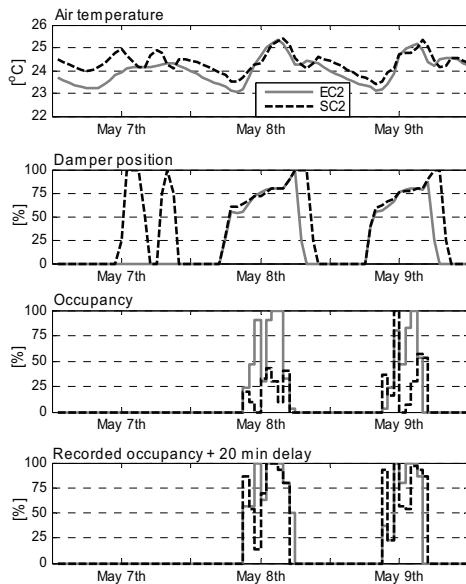


Figure 7 Sunday May 7th to Tuesday 9th of May 2006.

Summer

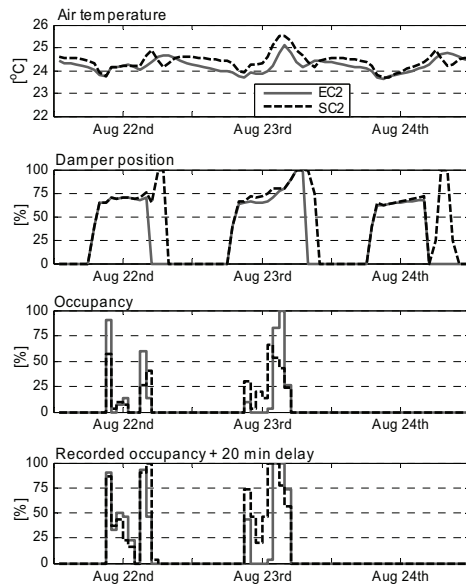


Figure 8 Tuesday 22nd to Thursday 24th of August 2006

In Table 1 some parameters are summarized for the three day periods presented in Figure 5 to Figure 8.

Table 1 Summary of the key parameters from the periods given in Figure 5 to Figure 8.

Season	Mean temp [°C]		Max temp [°C]		Total air flow [m ³]		Occupancy 0/20 [h]	
	EC2	SC2	EC2	SC2	EC2	SC2	EC2	SC2
Fall	20.8	21.2	22.5	23.2	442	530	4.2/5.3	2.9/5.6
Winter	19.4	20.0	21.5	22.1	376	383	6.6/10.2	4.7/11.2
Spring	24.0	24.3	25.3	25.4	3208	5630	9.1/12.3	4.9/12.0
Summer	23.9	24.1	25.1	25.5	4158	5868	4.1/7.0	4.0/9.2

The radiators were operating only during the 3-day winter period, and required 3.5 kWh and 2.6 kWh for EC2 and SC2, respectively.

The following observations can be made from the measurement results:

- Neither the mean temperatures nor the maximum temperatures are significantly different for the two rooms.
- To obtain the thermal conditions, Room SC2 requires considerably higher air volume flow rate, especially during the summer and spring periods.
- Room EC2 is according to the occupancy data used more intensively than Room SC2. The ratio between occupancy without delay and with 20 minute delay is higher for Room EC2, which indicates that the room is used more continuously than Room SC2.
- Seen in connection with the occupancy data, the exposed concrete in Room EC2 does not seem to delay the temperature peak significantly compared to Room SC2.

At first sight, the measurement results indicate that Room EC2 performs best with respect to cooling demand. However, there are some factors that must be taken into consideration before crediting the exposed concrete ceiling. Firstly, the removal of the false ceiling resulted in an increased room volume of about 4 m³ and an increased external wall area of about 1.2 m². Secondly, the false ceiling consists of a 22 mm plaster board insulated with 50 mm mineral wool, which provided additional insulation. This may have had influence on the results, and is also supported by the measurements of the radiator use during the 3-day winter period. The radiator uses about 35 % more heating energy during this period.

The occupancy data have also some degree of uncertainty. As mentioned, room absence was registered when no movement was detected for five minutes. In the post-processing of the data, this delay was subtracted from each time the room was registered as empty. This implies that absences shorter than five minutes were not registered, and affects the results in the direction of a higher registered occupancy than the actual occupancy. There may also have been false detections caused by non-human activity. On the other hand, false registrations may have occurred when a person was present but not detected because the movements were too small, which results in a lower registered occupancy than the actual occupancy.

To be able to compare the performance of the rooms on an equal basis and to quantify the achievements, a simulation model must be used.

5. Simulation model

A model of the building is implemented in the simulation program ESP-r, which is a transient building simulation program based on the finite volume technique. ESP-r is capable of modeling the energy and fluid flows within combined building and plant systems when constrained by control actions and subject to dynamically varying boundary conditions [16].

The model comprises in total 25 thermal zones. Most of the office cells are merged to larger units, while the offices with special interest in this study are modeled in detail. Figure 9 shows how the implemented geometry appears in the ESP-r cad window, with rooms involved in the experimental study enlarged for convenience. Surrounding buildings which influence on the solar irradiation are not shown, but are implemented in the model.

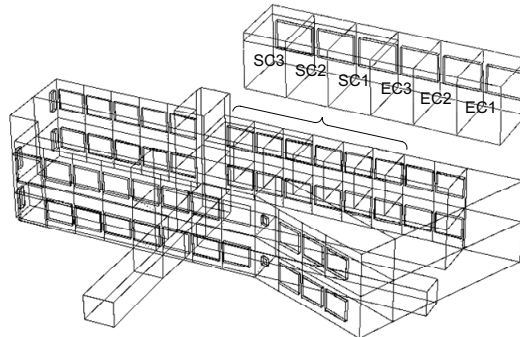


Figure 9 Simulation model as it appears in the ESP-r cad window.

5.1 Building constructions

The modeling of the building elements are done by investigating the construction drawings. In principle, the building envelope is very well insulated, far better than required by the national building code at the time of planning. However, the extensive occurrences of thermal bridges make the U-value increase significantly. Table 2 shows the basic U-values and the thermal bridge corrected U-values calculated in accordance with standards [18].

Table 2 U-values for selected building constructions

Building construction	Basic U-value [W/m ² K]	Thermal bridge corrected U-value [W/m ² K]
External walls	0.18	0.33
Roof	0.12	0.17
Floor on ground	0.15	0.17
Windows	1.31	1.40

The air intake culvert and the air distribution culvert are embedded ground coupled concrete ducts. The floors and the walls are modeled as a concrete layer and an insulation layer. One meter of earth is also included in these constructions. The construction is coupled to a constant temperature in the ground; 10 °C underneath the building and 5 °C outside the building. This is a fairly rough simplification, but gives a good approximation to the complex physics in the culverts [19]. This approximation is also done for the floor constructions with ground coupling.

5.3 Heat gains and ventilation schedule

The measurements show considerable differences in the occupancy with and without the 20 minute delay. This is taken into account in the simulation model. The maximum heat gains from people and equipment is multiplied with the average hourly occupancy without delay, while the lighting and ventilation is set according to the occupancy data with a 20 minute delay.

Table 3 Internal heat gains in the office rooms

Gain	Sensible [W/m ²]	Latent [W/m ²]	Rad comp. * [%]
Person	9	5	58
Equipment	14	0	50
Lighting	13	0	80
Total	36	5	63

* Radiative component of the sensible heat gain

The supply air rate, dependent on both time and temperature to the respective zones is then:

$$Q_a(t, \theta) = Q_{basic}(t) + O(t) \cdot Q_{pres} + Q_{temp}(\theta) \text{ [m}^3/\text{s]} \quad (1)$$

where

Q_a = total ventilation rate supplied to zone

Q_{basic} = basic ventilation supplied weekdays
from 6am to 4pm

O = occupancy (0-1)

Q_{pres} = occupancy dependent air supply

Q_{temp} = temperature dependent air supply

t = time

θ = temperature

5.4 Calibration

Climate data for use in the calibration of the simulation model was collected at the building site during September 2006. Temperature and wind data was monitored by the building energy management system, and the solar data was collected with the use of a solar radiation sensor which measures both global and

diffuse radiation [20]. Figure 11 shows the most influential parameters from September 4th to September 6th, which were the days chosen for the validation of the simulation model.

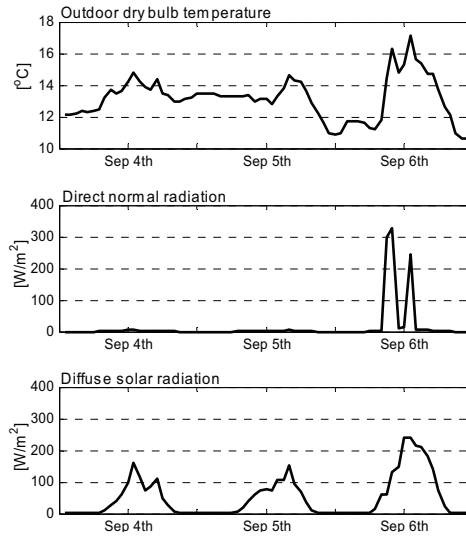


Figure 11 Chosen climate parameters collected at the building site, September 4th to September 6th, 2006.

All zones in the building have their respective measured occupancy profiles implemented for the calibration period. The room geometries are similar to the measurements, i.e. the room height is 50 cm higher for the rooms without suspended ceiling. The greatest uncertainties in the calibration are connected to the infiltration and the ventilation air changes. These two parameters are only known approximately and by trial and error fitted to the measurements. The best fits were found to be 0.12 h⁻¹ infiltration rate and 90 % of the design ventilation air flow rate, respectively. Figure 12 shows the air temperature for Room EC2 and Room SC2 after calibration.

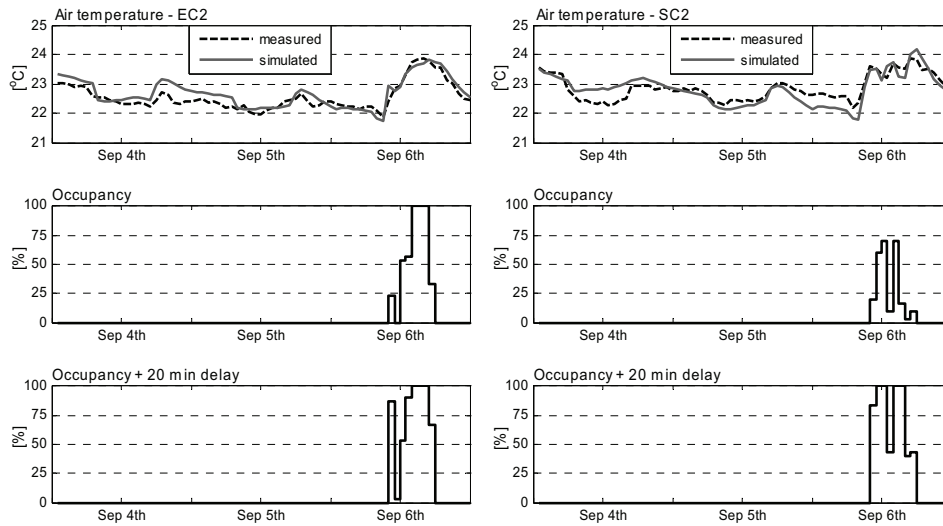


Figure 12 Simulation results for the room with exposed concrete (EC2, left) and the room with suspended ceiling (SC2, right) after calibration.

6. Simulations

Annual and seasonal simulations were carried out to estimate potential energy savings and comfort achievements of exposing the concrete in the ceiling. In the energy simulations, all offices are either with or without the suspended ceiling, respectively. The comfort simulations focus on the same office cells as the measurements (EC2 and SC2). The room heights are similar for all rooms in the simulations.

The occupancy factor, defined as the number of occupied rooms divided by the total number of rooms, is for Røstad relatively low. Therefore, to make the simulations more general, two additional occupancy profiles are added. The second profile is measured occupancy data from Statens Hus, a typical office building in Trondheim [21]. The third one, labeled Fictitious, is an extreme profile where the room is assumed in use continuously the entire working day. The profiles for the working days are showed in Figure 13. The internal heat gains are the same as in Table 3.

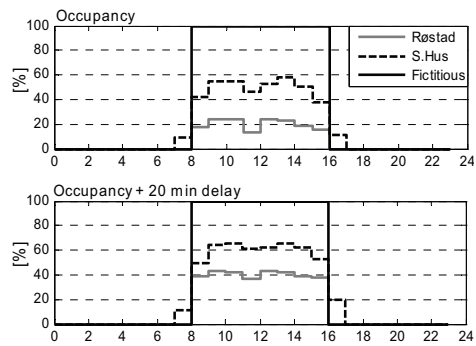


Figure 13 Occupancy profiles used in the simulations

6.1 Heating energy simulations

Annual simulations were run to evaluate the differences in heating energy demand for the whole building with and without suspended ceiling and for different occupancy profiles. The results summarized in Table 4 show that there are only minor differences in total heating energy demand whether the concrete is exposed or not. Even though the space heating energy demand is about 10 % lower at fall and spring, the annual differences are insignificant.

Table 4 Simulated heating demand for the three different occupancy profiles

Case	Occupancy profile		Heating demand [kWh/m ²]					Annual	Rel. [%]*
			Fall	Winter	Spring	Summer			
Suspended ceiling	Røstad	Space	4.9	39.9	8.9	0.2	53.9		
		Vent.	1.9	9.2	2.9	2.5	16.5		
		Total	6.7	49.1	11.8	2.7	70.4		
	Statens Hus	Space	4.0	37.9	7.7	0.2	49.8		
		Vent.	2.2	10.4	3.4	3.0	19.0		
		Total	6.2	48.3	11.2	3.2	68.8		
	Fictitious	Space	3.1	33.7	6.4	0.1	43.4		
		Vent.	2.7	11.0	4.2	4.2	22.1		
		Total	5.8	44.7	10.7	4.4	65.5		
	Exposed concrete	Røstad	Space	4.6	40.7	8.4	0.1	53.9	-0.1
			Vent.	1.9	9.2	3.0	2.7	16.7	1.5
			Total	6.5	49.9	11.4	2.9	70.6	0.3
Statens Hus		Space	3.7	37.7	7.2	0.1	48.7	-2.1	
		Vent.	2.1	10.4	3.4	3.2	19.1	0.8	
		Total	5.9	48.1	10.6	3.3	67.9	-1.3	
Fictitious		Space	2.7	33.5	5.8	0.1	42.1	-3.0	
		Vent.	2.6	11.0	3.9	4.7	22.2	0.5	
		Total	5.3	44.4	9.8	4.8	64.3	-1.8	

*Heating energy demand relative to the case with suspended ceiling

6.2 Thermal comfort

Simulations were done to assess the thermal comfort in the office rooms during the period from May 1st to September 30th. Table 5 summarizes the number of hours the operative temperature exceeds 25 °C and 26 °C for the office with exposed concrete (EC2) and room with suspended ceiling (SC2), respectively.

The simulations indicate that the exposed concrete reduces the hours of excessive temperatures significantly, and that the differences increase with the internal heat load.

Table 5 Number of hours (n) the operative temperature (θ_{op}) exceeds 25 °C and 26°C during working hours.

Occupancy profile	$n \theta_{op} > 25^\circ\text{C}$		$n \theta_{op} > 26^\circ\text{C}$	
	EC2	SC2	EC2	SC2
Røstad	33	43	2	5
Statens Hus	90	123	18	31
Fictitious	194	256	35	68

Figure 14 shows the simulation of a warm week in August with the fictitious user profile. Although the exposed concrete only dampens the maximum temperature about half a degree, it makes the room keep below the limit of 50 hours exceeding 26 °C in the cooling season.

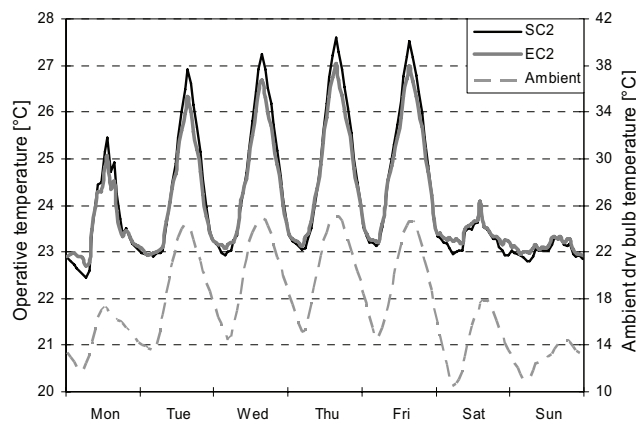


Figure 14 Thermal conditions for the office cells with the fictitious occupancy profile

6.3 Mechanical ventilation

In this section it is assumed that the office building is equipped with a balanced mechanical ventilation system, with 80 % heat recovery efficiency. The ventilation system operates from 6 am to 4 pm. Outside working hours, the ventilation system is supplying the office rooms with conditioned air as long as the operative temperature in the rooms is above 21 °C. In the cooling season, the supply air temperature is fixed at 17°C. It is also assumed that the offices have wall diffusers, thus the forced convection regime must be taken into consideration. Beausoleil-Morrison [22] studied the influence of convection

correlations common in building simulation programs, taking into account location of heating devices and HVAC equipment, and when they were operated. The study showed large differences with the use of correlations taking into account the supplementary information, relative to the default correlation in ESP-r. When the ventilation system is running, the Fischer convection correlation proposed by Beausoleil-Morrison [23] is used. Two cases are considered;

- A. Common design flow rate; 7 l/s per person and 2 l/s per m² floor area (~3.5 ach in the offices)
- B. Minimum required ventilation rate for offices in Norway; 7 l/s per person and 0.7 l/s per m² floor area (~1.9 ach in the offices), provided documented low-emitting building materials

Table 6 summarizes the annual space heating demand for the three different user profiles in Case A and Case B, respectively. Although the space heating energy savings are more evidently compared to the case of hybrid system with displacement ventilation, they are still minor.

Table 6 Space heating demand for the whole building with suspended ceiling and exposed concrete

Case	Occupancy profile	Space heating demand [kWh/m ²]		Rel [%]*
		Exp concrete	Susp ceiling	
A	Røstad	57.8	59.0	-2.0 (-1.7)
	Statens Hus	51.5	53.1	-3.0 (-2.6)
	Fictitious	42.9	45.1	-4.9 (-4.1)
B	Røstad	52.5	54.7	-4.0 (-3.6)
	Statens Hus	46.9	49.0	-4.3 (-3.8)
	Fictitious	38.9	41.7	-6.7 (-5.8)

* Relative space heating demand for exposed concrete. Relative total heating energy saving in parenthesis

Table 7 summarizes the number of hours the operative temperature in the office rooms exceeds 25 °C and 26 °C for the three different user profiles in Case A and Case B, respectively.

Table 7 Number of hours (n) the operative temperature (θ_{op}) exceeds 25 °C and 26°C during working hours.

Case	Occupancy profile	$n \theta_{op} > 25^\circ\text{C}$		$n \theta_{op} > 26^\circ\text{C}$	
		EC2	SC2	EC2	SC2
A	Røstad	0	2	0	0
	Statens Hus	1	15	0	0
	Fictitious	49	226	1	67
B	Røstad	86	155	2	26
	Statens Hus	310	413	55	208
	Fictitious	525	555	373	443

Figure 15 and Figure 16 show the resulting operative temperature for Case A and Case B, respectively. Both figures show that the increased convection reduces the maximum temperature. Also, the room with exposed concrete creates a better and more stable environment during the day. While the operative temperature in the room with suspended ceiling (SC2) rises about 5 K during the warmest days, the temperature in the room with exposed concrete (EC2) rises only 3 K. However, EC2 requires an air flow rate 10-17 % higher outside working hours to achieve this. In comparison, the guidelines to the national building code in Norway [24] recommend a maximum temperature variation of 4 K during a working day.

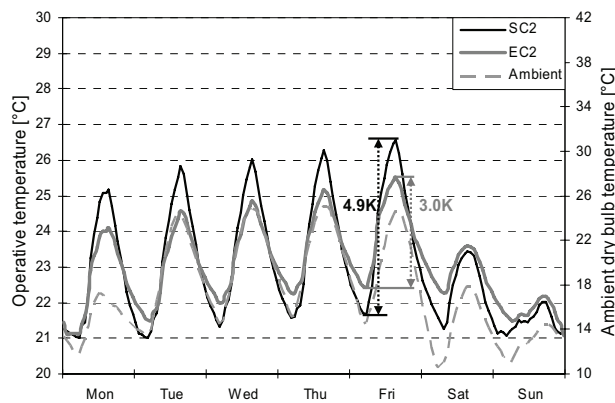


Figure 15 Simulations of a warm week in August with the fictitious user profile for Case A

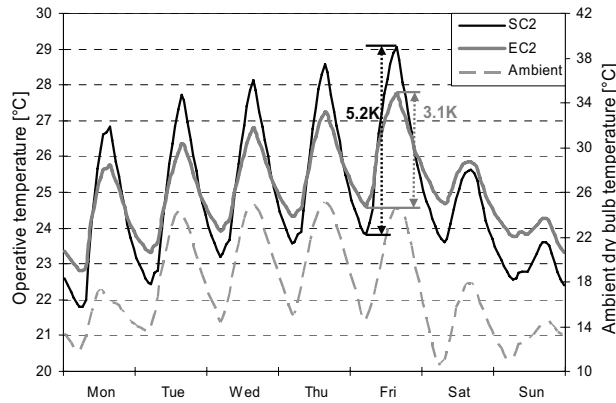


Figure 16 Simulations of a warm week in August with the fictitious user profile for Case B

6.4 Mechanical day ventilation and natural night ventilation

As mentioned in the introduction, Scandinavia has a large potential in utilizing natural ventilation to cool the building during the night hours. This section evaluates the potential to cool the building structure during the night with outdoor air as an alternative to supply the rooms with conditioned air outside working hours. Between 6 am and 4 pm the mechanical ventilation system supplies the rooms with conditioned air. Based on the requirement of maximum 50 hours with operative temperature above 26 °C, Figure 17 shows the maximum day-time internal heat load the rooms can handle as a function of room night-time air changes for Case A and Case B, respectively. In these simulations the fictitious occupancy profile is used.

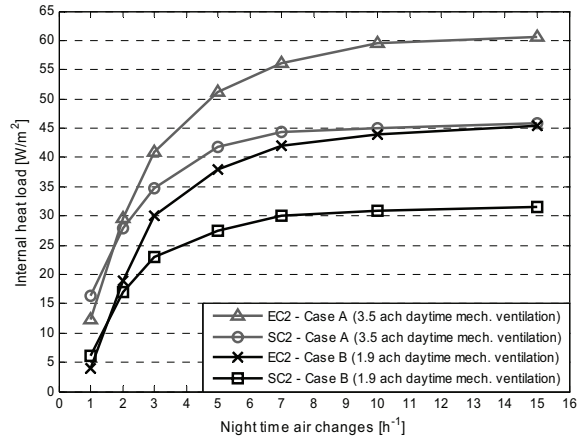


Figure 17 Maximum internal heat load the rooms can handle as function of night time air changes

Figure 17 shows that the rooms can handle a considerably higher internal heat load if they are sufficiently ventilated outside working hours. When the hourly room air change is above two ach, the room with exposed thermal mass can handle a considerably larger internal load compared to the room with suspended ceiling. An air change rate exceeding 10 ach does not improve the performance significantly.

7. Discussion

In the experimental part of this study, emphasis has been on making the conditions in the studied office rooms as similar as practically possible. Still, removal of the suspended ceiling enlarges both the room volume and the area exposed to the exterior. This may have had influence on the measurement results. Also, the many parameters affecting the indoor temperature have made it difficult to use the measurements directly to address and quantify the apparent achievements of exposing the concrete in the ceiling to the indoor air.

Presence of people does not only affect the indoor temperature by virtue of internal heat load, but also ventilation, lighting and room temperature set-points. In addition, the HVAC system will always try to equalize any deviation from the room temperature set-point, thus a quantitative evaluation of the

performance of the office rooms under normal operation directly from the measurements has been difficult. Hence, the most important use of the measurements has been to calibrate the simulation model.

It may be questionable whether hourly averaging of the room occupancy is an adequate approach to approximate the actual use of the rooms. In open plan offices or in rooms where several people work, this approach should be applicable. However, in single offices, persons are either present or absent. In theory, a mass-less building would practically require no time for heating or cooling to the desired set-point, and would have lower overall cooling or heating loads than actual buildings. The question is whether short time absences from the room can be used to cool or heat the room quickly, and be fully “recovered” when the person is back. Sub-hourly studies with different HVAC control strategies should be done to investigate this.

Figure 15 and Figure 16 clearly show the thermal mass effect. The temperature peaks are reduced with more than 1 K the warmest days and the temperature variation during the day is significantly less for the room with exposed concrete. However, to achieve this, the exposed concrete room demands up to 17 % higher air volume flow rate outside working hours compared to the room with suspended ceiling. Thus, if utilization of thermal mass also is to be a significant energy saving measure, heat must be removed from the space by other means than mechanical ventilation. Use of natural night ventilation shows promising results. As shown in Figure 17, the room with exposed concrete can handle an internal heat load about 10-15 W/m² higher than the room with suspended ceiling if sufficient night-time ventilation is provided. Moreover, the room with exposed concrete can handle the internal heat load with just half the mechanical ventilation day-time airflow rate (EC2 Case B compared to SC2 Case A), provided that the night-time air change is about 7 ach or larger.

8. Conclusions

In this paper, it has been shown that ESP-r is capable of simulating an advanced controlled office building in operation with very good agreement with measurements. The results obtained in this study are summarized below.

The simulations supported by measurements indicate that compared to office rooms with suspended ceiling, rooms with exposed concrete;

- do not decrease the annual space heating demand significantly ($< 3\%$),
- decrease the number of hours with excessive temperatures, and
- can handle a larger internal heat load and a more intensive use during the working day.

For mechanical ventilation system with wall mounted diffusers close to the ceiling, compared to office rooms with suspended ceiling, rooms with exposed concrete;

- increase the space heating savings somewhat more than in the case of hybrid system ($< 7\%$),
- decrease the maximum temperature with more than 1 K the warmest days of the year, and
- sustain a steadier thermal environment throughout the working day. While the operative temperature in rooms with suspended ceiling varies about 5 K, the rooms with exposed concrete varies only 3 K on warm days.

Night-time ventilation with outdoor air shows promising results. Compared to rooms with suspended ceiling, rooms with exposed concrete can;

- handle a considerably higher day-time internal heat load, provided that the night-time air change is greater than 2 ach,
- handle the same internal heat load with half the day-time ventilation rate, provided that the night-time ventilation is greater than 7 ach, and
- handle an extra internal heat load of 10-15 W/m² in the offices, provided that the night-time ventilation is 7 ach or greater. Hourly air change beyond 10 ach does not improve the performance particularly for either of the room types.

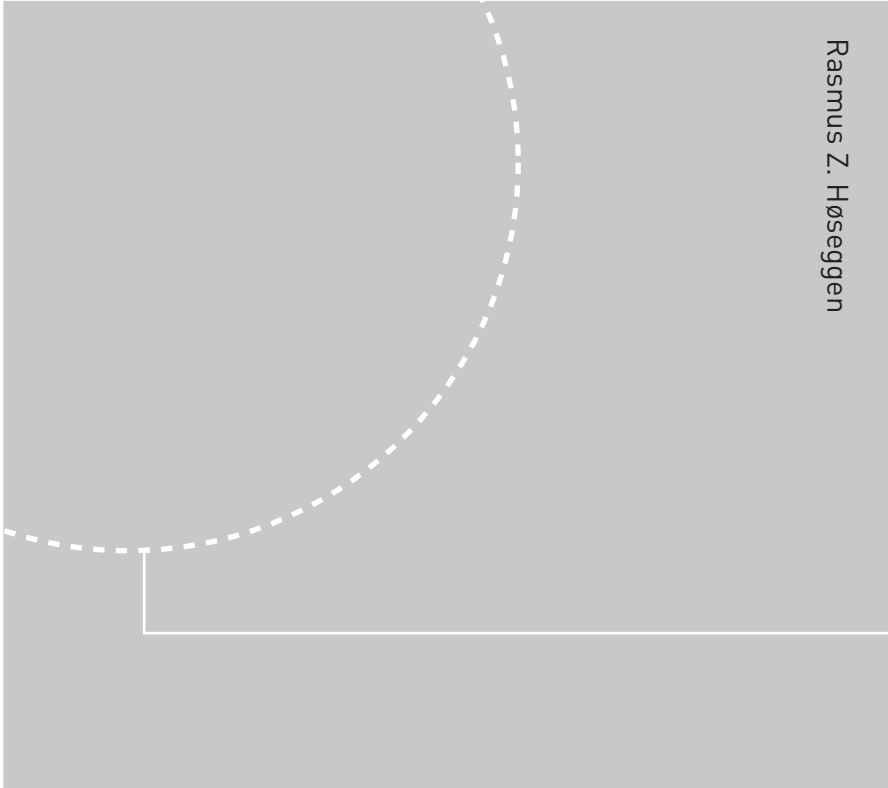
9. Acknowledgment

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