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PASSIVE COOLING OF BUILDINGS BY NIGHT-TIME VENTILATION

Schlussbericht

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Zusammenfassung

In Europa wurde in den letzten Jahren ein deutlicher Anstieg des Energiebedarfs für die Gebäudekühlung beobachtet, obwohl z.B. durch passive Kühlung mit Nachtlüftung der Einsatz aktiver Gebäudekühlung in vielen Fällen vermieden werden könnte. Aufgrund von Unsicherheiten in der Prognose der thermischen Komfortbedingungen, werden passive Kühlkonzepte aber nur mit Zurückhaltung eingesetzt. Da die Effektivität der Nachtkühlung stark von den klimatischen Randbedingungen abhängt, wurde eine Methode zur Quantifizierung des klimatischen Kühlpotenzials entwickelt und der Einfluss des Klimawandels untersucht. Obwohl ein deutlicher Rückgang festgestellt wurde, bleibt voraussichtlich doch ein erhebliches Potenzial erhalten, insbesondere, wenn die Nachtlüftung in Kombination mit anderen Kühlmethoden angewendet wird. Gebäudesimulationen zeigten, dass auch der Wärmeübergang an den Raumoberflächen die Effektivität der Nachtlüftung deutlich beeinflussen kann, wobei die Auskühlung für Wärmeübergangskoeffizienten unter ca. 4 W/m^2K stark eingeschränkt ist. Der Wärmeübergang während der Nachtkühlung mit Mischlüftung und Verdrängungslüftung wurde in einem Testraum an der Universität Aalborg untersucht. Dabei wurde die Temperatureffizienz der Lüftung bestimmt. Aus den gewonnenen Erkenntnissen wurde eine Methode entwickelt, die eine Abschätzung der möglichen Anwendung von Nachtlüftung schon während einer frühen Planungsphase ermöglicht.

Abstract

Due to an overall trend towards an increasing cooling energy demand in buildings in many European countries over the last few decades, passive cooling by night-time ventilation is seen as a promising concept. However, because of uncertainties in thermal comfort predictions, architects and engineers are still hesitant to apply passive cooling techniques. As night-time ventilation is highly dependent on climatic conditions, a method for quantifying the climatic cooling potential was developed and the impact of climate warming was investigated. Although a clear decrease was found, significant potential will remain, especially if night-time ventilation is applied in combination with other cooling methods. Building energy simulations showed that the performance of night-time ventilation is also affected by the heat transfer at internal room surfaces, as the cooling effect is very limited for heat transfer coefficients below about 4 W/m^2K. Heat transfer during night-time ventilation in case of mixing and displacement ventilation was investigated in a full scale test room at Aalborg University. In the experiments the temperature efficiency of the ventilation was determined. Based on the previous results a method for estimating the potential for cooling by night-time ventilation during an early stage of design was developed.

Executive summary

In modern, extensively glazed office buildings, due to high solar and internal loads and increased comfort expectations, air conditioning is increasingly applied even in moderate and cold climates, like in Central and Northern Europe. Particularly in these cases, night-time ventilation is often seen as a promising passive cooling concept. Many successful examples of passively cooled buildings demonstrate the possibility of providing good thermal comfort conditions without the need for energy-intensive air conditioning systems. However, due to uncertainties in the prediction of thermal comfort, architects and engineers are still hesitant to apply passive cooling techniques.

The basic concept of night-time ventilation involves cooling the building structure overnight in order to provide a heat sink during the occupancy period. As this requires a sufficiently high temperature difference between the ambient air and the building structure, the efficiency of night cooling is highly sensitive to climatic conditions and hence also to climate warming. In the first part of this PhD study, the potential for passive cooling of buildings by night-time ventilation was evaluated by analysing climatic data, without considering any building-specific parameters. A method for quantifying the climatic cooling potential (CCP) was developed based on degree-hours of the difference between building and external air temperature. Applying this method to climatic data of 259 stations shows very high night cooling potential over the whole of Northern Europe and still significant potential in Central, Eastern and even some regions of Southern Europe. However, due to the inherent stochastic properties of weather patterns, series of warmer nights can occur at some locations, where passive cooling by night-time ventilation alone might not be sufficient to guarantee thermal comfort.

Possible time-dependent changes in CCP were assessed for the period 1990-2100, with particular emphasis on the *Intergovernmental Panel on Climate Change* (IPCC) “A2” and “B2” scenarios for future emissions of greenhouse gases and aerosols. The study was based on 30 Regional Climate Model (RCM) simulated datasets, as obtained from the European *PRUDENCE* project. Under both emissions scenarios and across all locations and seasons, CCP was found to decrease substantially by the end of the 21st century, so that night-time cooling will cease to be sufficient to assure thermal comfort in many Southern and Central European buildings. In Northern Europe, a significant passive cooling potential is likely to remain, at least for the next few decades.

Because heat gains and night ventilation periods typically do not coincide in time, heat storage is essential for effective night cooling, and thus a sufficient amount of thermal mass is needed in the building. In order to assess the impact of different parameters, such as slab thickness, material properties and the surface heat transfer, the dynamic heat storage capacity of building elements was quantified based on an analytical solution of one-dimensional heat conduction in a slab with convective boundary condition. The potential of increasing thermal mass by using phase change materials (PCM) was also estimated. The results show a significant impact of the heat transfer coefficient on heat storage capacity, especially for thick, thermally heavy elements. For thin, light elements a significant increase in heat capacity due to the use of PCMs was found to be possible.

In order to identify the most important parameters affecting night ventilation performance, a typical office room was modelled using a building energy simulation program (*HELIOS*), and the effect of different parameters such as building construction, heat gains, air change rates, heat transfer coefficients and climatic conditions on the number of overheating degree hours (operative room temperature >26 °C) was evaluated. Besides climatic conditions, the air flow rate during night-time ventilation was found to have the largest effect. However, thermal mass and internal heat gains also have a significant impact on the achievable level of thermal comfort. A significant sensitivity to the surface heat transfer was found for total heat transfer coefficients below about 4 W/m^2K.

The convective heat transfer at internal room surfaces is highly affected by the indoor air temperature distribution and the near-surface velocities both of which can vary significantly depending on the air flow pattern in the room. Increased convection is expected due to high air flow rates and the possibility of a cold air jet flowing along the ceiling, but the magnitude of these effects is hard to predict. Heat transfer during night-time ventilation in case of mixing and displacement ventilation has been investigated in a full scale test room. The performance of night time cooling was evaluated based on the temperature efficiency of the ventilation. The results show that for low air flow rates displacement ventilation is more efficient than mixing ventilation. For higher airflow rates the air jet flowing along the ceiling has a significant effect, and mixing ventilation becomes more efficient.

Combining the results of the previous steps, a practicable method for the estimation of the potential for cooling by night-time ventilation during an early stage of design is proposed. In order to assure thermal comfort two criteria need to be satisfied, i.e. (i) the thermal capacity of the building needs to be sufficient to accumulate the daily heat gains within an acceptable temperature variation and (ii) the climatic cooling potential and the effective air flow rate need to be sufficient to discharge the stored heat during the night. The estimation of the necessary amount of thermal mass in the building is based on the dynamic heat storage capacity. The air flow rate needed to discharge the stored heat at a certain climatic cooling potential is assessed based on the temperature efficiency of the ventilation.

1. Introduction

1.1. Increasing cooling energy demand

During the last few decades, a trend towards increasing cooling demand in buildings has been observed in many European countries. Due to high internal and solar heat gains commercial buildings with extensive glazing tend to be overheated in summertime, even in moderate and cold climates like in Central and Northern Europe.

In Switzerland, 46.7 % of the total energy is used in Buildings, i.e. for space heating (35.1 %), domestic hot water (5.5 %), lighting (3.4 %), and for air conditioning and ventilation systems (2.7 %). In private homes until now only 0.1 % of the electric energy is used for air conditioning systems. However, from 2000 till 2006 this number increased by 158 %. In the service sector, the share of the electric energy applied for air conditioning and ventilation systems amounts to 29.1 % already today and increased by 10.6 % during the same period [1].

A study on the energy efficiency of air conditioning systems [2] shows a similar situation for most European countries. Until 2020 the demand for cooled floor area is expected to more than double compared to 2000. Again, office and commercial buildings have the largest share (Figure 1).

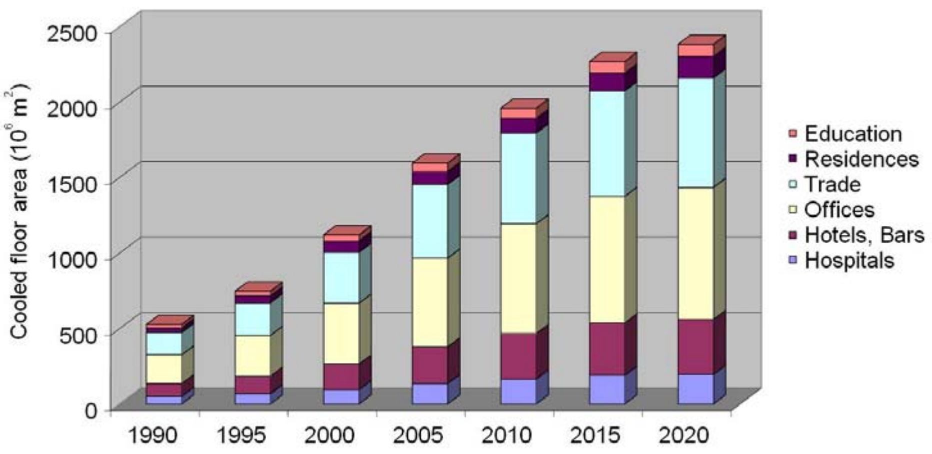


Figure 1. *Evolution of cooled floor area in the EU 15 countries by economic sectors from 1990 to 2020 [2].*

While the heating requirement can be effectively reduced by installing thermal insulation, cooling plays an increasingly significant role in the overall energy demand of buildings. This trend is enhanced by changes in climatic conditions. In mild winters less energy is needed for heating, but more frequent and longer hot spells result in a significant increase in cooling energy demand. For Switzerland, Christenson et al [3] found a clear increase in cooling degree days during the last century, especially during the last 20 years. Although future climate models are subject to high uncertainties, a continuous increase in cooling degree days is believed to be very likely. Using building energy simulation, Frank [4] observed an increase in cooling energy demand in Swiss office buildings by 223-1050 % for the period 2050-2100 compared to 1961-1990.

Additionally, the urban heat island effect contributes to an increasing cooling demand [5]. In densely populated urban areas, changes in the thermal properties of surface materials (e.g. asphalt or concrete), the lack of evaporation from vegetation and a high density of waste heat sources, result in significantly higher temperatures compared to surrounding rural areas. As cities tend to become larger and more densely populated, this effect is expected to increase further.

1.2. Passive cooling of buildings

Basically, active cooling can be avoided in well-designed buildings in most Central and Northern European locations. The first step to achieving this is the reduction of internal and solar heat gains. Solar heat gains can be limited by a moderate glazing ratio and effective, preferably exterior, solar shading devices. Also people and electric equipment contribute to the total heat gains in a building. Therefore, next to the direct energy savings, the application of energy efficient office equipment and using daylight instead of electric light helps to reduce the cooling energy demand and to increase thermal comfort.

Besides the reduction of heat gains, passive cooling is based on the utilisation of natural heat sinks including the sky, the atmosphere and the earth. Ventilation to the atmosphere is the most elementary practise of heat removal from buildings and the most widely used passive cooling method. Ventilative cooling includes the direct removal of heat gains during the day and the cooling of the building structure during the night. Heat transfer to the sky, or rather the outer space beyond the atmosphere, is exclusively by radiation. Radiative cooling is most efficient at night under cloudless skies, and therefore in regions having low atmospheric humidity. Evaporative cooling applies to all processes in which the sensible heat in an air stream is exchanged for the latent heat of water droplets or wetted surfaces. Direct evaporative cooling denotes systems where this takes place in the supply air stream. In indirect systems a heat exchanger is used in order to prevent humidification of the supply air. Generally evaporative cooling is most suitable for dry climates. Also the earth can be utilised as a heat sink in different ways. A building can be coupled directly to the earth, or the supply air can be cooled by means of an earth-to-air heat exchanger. These methods are widely applicable, as at most locations earth temperatures fall below or within the comfort range throughout the entire year. Additionally, ground water and also water from the sea, lakes or rivers can be utilised as natural heat sinks. Detailed information on different passive cooling techniques can be found in [6] and [7].

Many of these passive cooling systems have a very long tradition, especially in hot and arid climates, like in Iran [8]. E.g. dome-shaped roofs with air vents were incorporated in buildings as early as 3000 BC. Also very massive constructions were traditionally used in order to minimise daily internal temperature swings. For increased ventilation different types of wind towers were developed in Egypt (malqaf), the Persian Gulf states (badgir) and Pakistan (wind scoops) [9]. Wind towers were also used in combination with underground tunnels and evaporative cooling from damp walls, fountains or underground water streams. Combining different methods yielded very effective cooling systems.

In modern applications the combination of different passive, or passive and active cooling systems is called hybrid cooling. When passive and active systems are combined the passive system is used as long as the natural heat sink is sufficient and therefore the energy consumption of the active system can be reduced significantly. Whenever passive means are not sufficient to provide acceptable thermal comfort conditions, the application of renewable energy sources should be considered for the operation of active air conditioning systems. E.g. an absorption chiller can be powered by using concentrated solar radiation as an energy source.

1.3. Night-time ventilation

During night-time ventilation the relatively cold outdoor air is used as a heat sink to cool the building. Whenever, the outdoor air temperature is below the building temperature, the thermal mass of the building can be cooled by ventilation. During the following day, heat gains are accumulated by the thermal mass which prevents extensive overheating of the building. The stored heat is then discharged by ventilation during the next night (Figure 2).

Typically solar and internal heat gains occur primarily during the day, but the highest potential for ventilative cooling is available during the night when the outdoor temperature is lowest. Therefore heat storage is an essential requirement for night-time ventilative cooling and a high thermal capacity of building elements (e.g. ceiling, floor, walls) is needed. Furthermore, for an efficient utilisation of the thermal capacity a sufficient heat transfer at the surface and sufficient conduction in the material are needed. An alternative to heavy building elements is the application of phase change materials for increasing the thermal capacity of light-weight structures [10].

Ventilation can be driven by natural forces (thermal buoyancy and wind), or by fans. In case of natural ventilation windows can be operated manually or automatically by a central building management system. As natural ventilation may cause a high variability in the air change rate, hybrid ventilation systems are often used in order to ensure a certain air flow rate. In hybrid ventilation mechanical and natural forces are combined in a two-mode system where the operating mode varies according to the season, and within individual days [11].

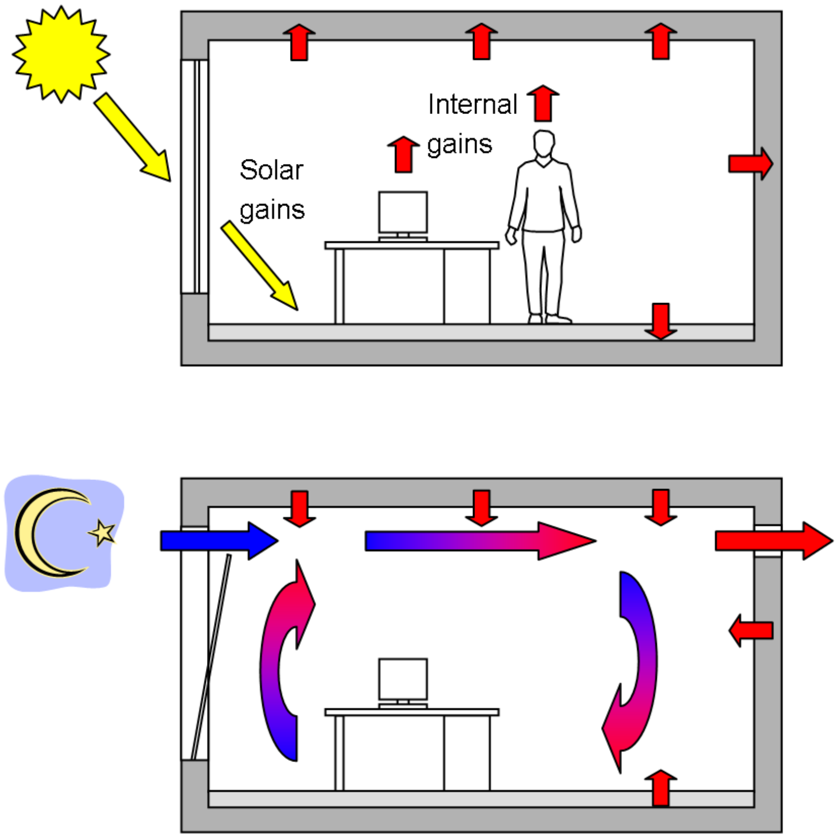


Figure 2. *Basic principle of cooling by night-time ventilation.*

1.4. Examples of buildings cooled by night-time ventilation

On completion in 2003 the *MIVA* building (*Christophorus Haus*) in Stadl-Paura, Austria [12] was one of the most innovative *Passivhaus* buildings in Europe. The cooling concept includes natural night-time ventilation combined with a ground source heat pump (hybrid cooling). 100 tonnes of thermal mass were included in the building in order to accumulate heat gains and decrease daily temperature variations.

In Switzerland, *Forum Chriesbach* [13] was the first office building applying the *Zero Energy House* standard. In this building passive cooling is achieved by combining natural night-time ventilation with an earth-to-air heat exchanger (hybrid cooling). During night-time ventilation cold outdoor air enters the building through windows in the façade, flows to the central atrium and is exhausted at roof level.

A similar concept is applied in the passively cooled *KfW* office building in Frankfurt, Germany. Night-time ventilation is driven by the stack effect in the atrium, but it is supported by mechanical fans when needed (hybrid ventilation). A monitoring study conducted in this building by Wagner et al. [14] showed, that even under extreme climate conditions acceptable thermal comfort conditions can be reached with passive cooling.

The energy consumption and the internal temperatures and CO_2 levels in the naturally ventilated *Lanchester Library* at Coventry University, UK were recorded by Krausse et al. [15]. Due to the exposed thermal mass and the night ventilation strategy the building meets thermal comfort criteria even during prolonged hot spells, using 51 % less energy than a typical air-conditioned office.

1.5. State-of-the-art and research topics

The international research program *PASCOOL* [16] aimed to develop techniques, tools and design guidelines to promote passive cooling applications in buildings. Within the framework of *PASCOOL* the user-friendly computer code *LESOCOOL* [17] was developed to predict the cooling power and fluctuations of internal temperature during night-time ventilation based on simple models. This tool was validated and a good agreement was found between simulated and experimental indoor air temperature. However, there are some limitations. Firstly, only one single air flow path without branches can be modelled. Secondly, the thermal model is only valid for infinitely thick walls. And thirdly, a constant heat transfer, h = 6 W/m^2K is applied for all internal surfaces. Radiative heat transfer between different surfaces is not considered.

IEA ECBCS Annex 28 dealt with low energy cooling systems. A selection chart was provided to help to identify which of the considered cooling technologies are likely to be suitable for a particular application on the basis of key building parameters. This is supported by summary sheets for each of the technologies giving a brief description and key information. For night-time ventilation in cool climates and heavy-weight constructions a possible offset of heat gains in the range of 20 - 30 W/m^2 and a corresponding peak space temperature reduction of 2 - 3 K is given [18].

Often natural or hybrid ventilation is applied for night-time cooling. The *NatVent* project [19] aimed to provide solutions for natural ventilation and low-energy cooling in office-type buildings in countries with moderate and cold climates. A design handbook [20] discusses the basic principles of natural ventilation and provides an overview of prediction methods. *IEA ECBCS* Annex 35 [11] aimed to develop control strategies for hybrid ventilation systems and methods to predict ventilation performance in hybrid ventilated office and educational buildings. An overview of available design methods like simple analytical and empirical methods, single-zone and multi-zone methods, and computational fluid dynamics (CFD) methods is given in [21].

Despite many successful examples and several research projects showing a high potential for cooling by night-time ventilation, architects and engineers continue to be hesitant to apply this technique. This is mainly because of uncertainties in thermal comfort predictions and the lack of simple design tools applicable in an early design stage, when many parameters affecting the performance of night cooling are not known yet. Assumptions made by the user in the input for building simulation leads to uncertainties in comfort predictions [22]. Additionally, the effectiveness of passive and low energy cooling systems is, by their very nature, much more sensitive to climate than is the performance of refrigeration-based systems [23]. For night-time ventilation a sufficiently high temperature difference between ambient air and the building structure is needed to achieve effective cooling of the building mass. The very high variability of climatic conditions and the uncertain development of global climate warming, therefore might cause discrepancies in thermal comfort conditions in real buildings compared to predictions based on standard climate data.

Some general guidelines for the applicability of ventilative night cooling are given by Givoni [24], [25], [26]. Night-time cooling is recommended mainly in arid and desert regions with a summer diurnal temperature fluctuation of 15 to 20 K and night-time temperatures below about 20 °C. For the maximum daytime temperature a range between 30 and 36 °C is given.

A method to evaluate the climate suitability of a given location for direct ventilative cooling and complimentary night-time ventilative cooling was presented by Axley et al. [27]. The method was applied to four different locations in different climatic zones of the United States. Cooling by natural ventilation was found to be feasible and effective in the cooler locations for moderate to high specific internal gains, but not for hot and humid climates, as for example in Miami, FL, where relatively high night-time ventilation rates would be needed to offset moderate specific internal gains. Estimates of the internal gains that may be offset by night-time cooling are based on the assumption that the building has, essentially, infinite thermal mass thus, as stated by the authors, these results may significantly overestimate the benefit of night-time cooling.

Eicker et al. [28] provided experimental evidence for the limitations of night-time ventilation under climatic conditions currently regarded as extreme. They monitored for 3 years an advanced low-energy office building in Weilheim (Germany), constructed in 1999, that was also equipped with an earth to air heat exchanger. While the building performed excellently during typical German summer conditions (2001, 2002), in 2003, with average summer temperatures more than 3 K higher than usual, nearly 10 % of all office hours were above 26 °C. As the temperatures observed during the exceptionally hot summer of 2003 might correspond to those of a typical summer at the end of this century [29], the findings of Eicker et al. clearly demonstrate that in decades to come cooling by night-time ventilation might cease to work in buildings designed for current climatic conditions.

The impact of climate change on different passive cooling techniques has been analysed by Roaf et al. [23]. For the applicability of night-time ventilation, a monthly mean daily maximum temperature of 31°C was assumed to be the limiting criterion. Based on this and other threshold values, they delineated the regions where different cooling techniques could cease to be viable by 2050. These were found to be extensive enough to conclude that the effects of global warming should be taken into account by designers. Although their approach was suitable to detect first-order effects, the study by Roaf et al. suffers from several shortcomings. Their threshold temperature of 31°C only applies to well-shaded buildings with relatively low internal loads (e.g. residential buildings), as the authors state. The method is also not suitable to quantify gradual changes in cooling potential, e.g. across spatial gradients or over time. Additionally, uncertainties associated with any regional climate projections [30], [31] were not taken into account.

Predictions of thermal comfort in buildings applying night-time ventilation are not only affected by variable climatic conditions, but also by uncertainties in other parameters, like the air change rate, heat gains, effective thermal mass and the heat transfer at internal surfaces. Several studies have been undertaken to investigate the effect of different parameters on the efficiency of night-time cooling.

Using an hourly simulation model, Shaviv et al. [32] analysed the maximum indoor temperature in a residential building in the hot humid climate of Israel as a function of night ventilation air change rate, thermal mass and daily temperature difference. In a heavy mass building, the maximum indoor temperature was found to be reduced by 3 - 6 °C compared to the outdoor maximum.

Finn et al. [33] investigated the effect of design and operational parameters on the performance of a night ventilated library building in the moderate maritime climate of Ireland. Increasing thermal mass by changing construction materials (from 887 kg/m^2 to 1567 kg/m^2, per unit floor area) was observed to lower peak daily temperature by up to 3 °C. Internal gains (20 to 40 W/m^2) and ventilation rates up to 10 ACH were also found to have a significant effect on internal comfort, with a change in peak temperature of up to 1.0 °C. However, increasing ventilation rates beyond 10 ACH did not lead to significant improvement.

Breesch [34] developed a methodology to predict the performance of natural night ventilation with building simulation (*TRNSYS* [35] coupled with *COMIS* [36]) taking into account the uncertainties in the input. Next to internal heat gains and the air tightness, internal convective heat transfer coefficients were found to have the most important impact on the predicted thermal comfort. It should be noted that increased convection due to high air change rates was not considered in this study. Further investigation of the convective heat transfer in case of natural night ventilation was recommended.

In most of these studies the effect of convective heat transfer at internal surfaces was not considered at all or only in a small range of heat transfer coefficients, even though other studies showed the importance of this parameter for thermal energy storage in building elements. Akbari et al. [37] used an analytical model and numerical simulations to evaluate the effectiveness of massive interior walls. The effectiveness – defined as the ratio of the wall's total diurnal heat storage capacity for a given convective heat transfer coefficient to the maximum storage capacity of the same wall when the coefficient is infinite – of a 0.305 m thick concrete wall was found to be almost doubled by increasing the convective heat transfer coefficient from 2.84 to 5.68 W/m^{2}K. Several authors point out the importance of this finding for the efficiency of night-time ventilation.

Depending on the direction of the heat flow, standard heat transfer coefficients for combined heat transfer (convection and radiation) are in the range from 5.9 to 10 W/m^{2}K [38]. However, during night-time ventilation radiation does not contribute to the heat transfer from room surfaces to the air (as air is virtually transparent for infrared radiation), but in fact transfers heat from one surface to another. For convective heat transfer standard coefficients are 2.5 W/m^{2}K for vertical walls, 5.0 W/m^{2}K for upward heat flow and 0.7 W/m^{2}K for downward heat flow [39]. This means that, especially at the ceiling – a concrete ceiling often represents a significant share of the thermal mass of a room – the convective heat transfer can be very limited (downward heat flow during night-time ventilation).

On the other hand a higher convective heat transfer is expected due to the increased air flow rate and the possibility of a cold air jet flowing along the ceiling [26], [40], [41]. However, Blondeau et al. [42] did not observe any significant difference in predicted indoor air temperature due to various increased convective coefficients during the night-time in their simulation study.

Several studies deal with the heat transfer at internal room surfaces. Different correlations were proposed for natural (e.g. Alamdari and Hammond [43], Khalifa and Marshall [44], Awbi and Hatton [45]) and mixed convection (e.g. Chandra and Kerestecioglu [40], Spitler et al. [46], Awbi and Hatton [47]) from horizontal and vertical surfaces. Based on such empirical correlations Beausoleil-Morrison developed an adaptive algorithm for the simulation of the convective heat transfer at internal building surfaces [48]. However, many of these correlations are based on experiments on small heated plates. A review comparing natural convective heat transfer at isolated surfaces and surfaces in enclosures revealed clear discrepancies [49], [50]. This demonstrates the necessity of considering a room as a whole.

Additionally, the heat transfer obviously depends on the air flow pattern in a room. The effect of different flow patterns on the storage efficiency during night-time ventilation has been investigated by Salmerón et al. [51] using a 2-dimensional computational fluid dynamics model. A variation by a factor of 6 was found between different configurations of air in- and outlet openings. However, radiation between internal room surfaces was not considered in this study. Furthermore, the impact of the air flow rate on the storage efficiency was not investigated.

The aim of this dissertation is to contribute to the improvement of the design methods for cooling by night-time ventilation. It is mainly focused on the effects of climatic conditions and the heat transfer at internal room surfaces.

1.6. Project phases

In the first part of the project the possibilities and limitations of night-time ventilation under different climatic conditions and the impact of climate warming are evaluated. A degree-hour method for quantifying the climatic potential for night-time ventilation is developed, verified and applied to present and future climate data.

As heat storage is vital for night-time cooling, the impact of different parameters on the dynamic heat storage capacity of building elements is investigated based on the analytical solution of one-dimensional heat conduction.

Subsequently, building energy simulations have been performed to analyse the sensitivity of night cooling performance. The effects of climate, thermal mass, heat gains, air change rate and heat transfer coefficients on thermal comfort conditions in an office building are discussed.

Additionally, the heat transfer in case of mixing and displacement ventilation has been investigated in a full scale test room at Aalborg University. This study provides a detailed analysis of convection and radiation during night-time ventilation depending on the air flow rate and the initial temperature difference between the inflowing air and the room.

Combining the results of the previous steps a new method to evaluate the potential for night-time ventilation during the concept design phase is proposed.

2. Climatic potential for passive cooling of buildings

Night-time ventilation is highly dependent on climatic conditions, as a sufficiently high temperature difference between ambient air and the building structure is needed during the night to achieve efficient convective cooling of the building mass. The purpose of this study was to evaluate the climatic potential for the passive cooling of buildings by night-time ventilation in present and future climates in Europe [52], [53]. A method was developed which is basically suitable for all building types, regardless of building-specific parameters. This was achieved by basing the approach solely on a building temperature variable within a temperature band given by summertime thermal comfort.

2.1. Definition of the climatic cooling potential

Degree-days or degree-hours methods are often used to characterise a climate's impact on the thermal behaviour of a building. The daily climatic cooling potential, CCP_d, was defined as degree-hours for the difference between building temperature, T_b and external air temperature, T_e (Figure 3):

$$CCP_d = \sum_{t=t_i}^{t_f} m_{d,t}\left(T_{b(d,t)} - T_{e(d,t)}\right) \qquad \begin{cases} \mathrm{m} = 1\mathrm{h} \ \text{ if } \ T_b - T_e \geq \Delta T_{crit} \\ \mathrm{m} = 0 \ \text{ if } \ T_b - T_e < \Delta T_{crit} \end{cases} \tag{1}$$

where t stands for the time of day, with $t \in \{0, ..., 24h\}$; t_i and t_f denote the initial and the final time of night-time ventilation, and ΔT_{crit} is the threshold value of the temperature difference, when night-time ventilation is applied. In the numerical analysis, it was assumed that night-time ventilation starts at t_i = 19 h and ends at t_f = 7 h. As a certain temperature difference is needed for effective convection, night ventilation is only applied if the difference between building temperature and external temperature is greater than 3 K.

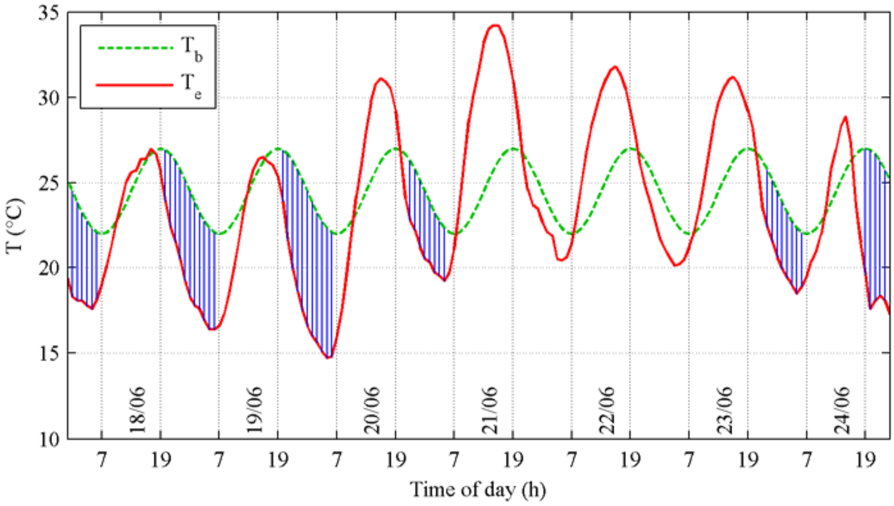


Figure 3. *Building temperature, T_b and external air temperature, T_e during one week in summer 2003 for Zurich SMA (ANETZ data). Shaded areas illustrate graphically the climatic cooling potential, CCP.*

As heat gains and night-time ventilation are not simultaneous, energy storage is an integral part of the concept. In the case of sensible energy storage, this is associated with a variable temperature of the building structure. This aspect is included in the model by defining the building temperature as a harmonic oscillation around 24.5°C with an amplitude of 2.5 K:

$$T_{b(t)} = 24.5 + 2.5\cos\left(2\pi\frac{t-t_i}{24}\right) \tag{2}$$

The maximum building temperature occurs at the starting time of night ventilation, and given a ventilation time of 12 hours, the minimum building temperature occurs at the end time (Figure 3). The temperature range T_b = 24.5 °C ± 2.5 °C corresponds to that recommended for thermal comfort in offices [54].

2.2. Practical Significance of CCP

To discuss the practical significance of the calculated degree-hours, an example shall be given. It is assumed that the thermal capacity of the building mass is sufficiently high and therefore does not limit the heat storage process. If the building is in the same state after each 24 h cycle, the daily heat gains Q_d (Wh) stored to the thermal mass, equal the heat which is discharged by night ventilation:

$$Q_d = \dot{m}\cdot c_p \cdot CCP_d \tag{3}$$

The effective mass flow rate $\dot{m}$ is written as $\dot{m} = A_{Floor}\cdot H\cdot\eta\cdot ACR\cdot\rho$, where A_{Floor} is the floor area and H the height of the room, ACR the air change rate and η a temperature efficiency, which is defined as $\eta = (T_{out} - T_e)/(T_b - T_e)$ and takes into account the fact that the temperature of the outflowing air T_{out} is lower than the building temperature T_b. The density and the specific heat of the air are taken as $\rho = 1.2\,\mathrm{kg/m^3}$ and $c_p = 1000\,\mathrm{J/(kg\,K)}$. Assuming a room height of H = 2.5 m and a constant effective air change rate of $\eta\cdot ACR = 6\,\mathrm{h^{-1}}$ yields:

$$\frac{Q_d}{A_{Floor}} = H\cdot\eta\cdot ACR\cdot\rho\cdot c_p\cdot CCP_d = \frac{2.5\,\mathrm{m}\cdot 6\,\mathrm{h^{-1}}\cdot 1.2\,\mathrm{kg/m^3}\cdot 1000\,\mathrm{J/kgK}}{3600\,\mathrm{s/h}}CCP_d = 5\frac{W}{m^2K}CCP_d \tag{4}$$

For the climatic cooling potential needed to discharge internal heat gains of 20 W/m^{2}K and solar gains of 30 W/m^{2}K during an occupancy time of 8 h follows:

$$CCP_d = \frac{Q_d}{A_{Floor}}\Big/ 5\frac{W}{m^2K} = \frac{(20+30)\cdot 8}{5}\mathrm{Kh} = 80\,\mathrm{Kh} \tag{5}$$

This example should be seen as a rough estimation only, as solar and internal gains of an office room can vary substantially depending on the type of building use, local climate, and the solar energy transmittance and orientation of the façade.

2.3. Climatic cooling potential in Europe

The degree-hour method was applied for a systematic analysis of the potential for night time cooling in different climatic zones of Europe. Semi-synthetic climate data [55] from 259 weather stations was used to map the climatic cooling potential (Figure 4). Additionally the cumulative frequency distribution of CCP was plotted for 20 European locations (Figure 5). These charts show the number of nights per year when CCP exceeds a certain value.

In the whole of Northern Europe (including the British Isles) a very significant climatic cooling potential was found, and therefore passive cooling of buildings by night-time ventilation seems to be applicable in most cases. In Central, Eastern and even in some regions of Southern Europe, the climatic cooling potential is still significant, but due to the inherent stochastic properties of weather patterns, series of warmer nights can occur at some locations, where passive cooling by night-time ventilation might not be sufficient to guarantee thermal comfort. If lower thermal comfort levels are not accepted during short periods of time, additional cooling systems are required. In regions such as southern Spain, Italy and Greece climatic cooling potential is limited and night cooling alone might not be sufficient to provide good thermal comfort during all the year. Nevertheless, night-time ventilation can be used in hybrid cooling systems during spring and fall.

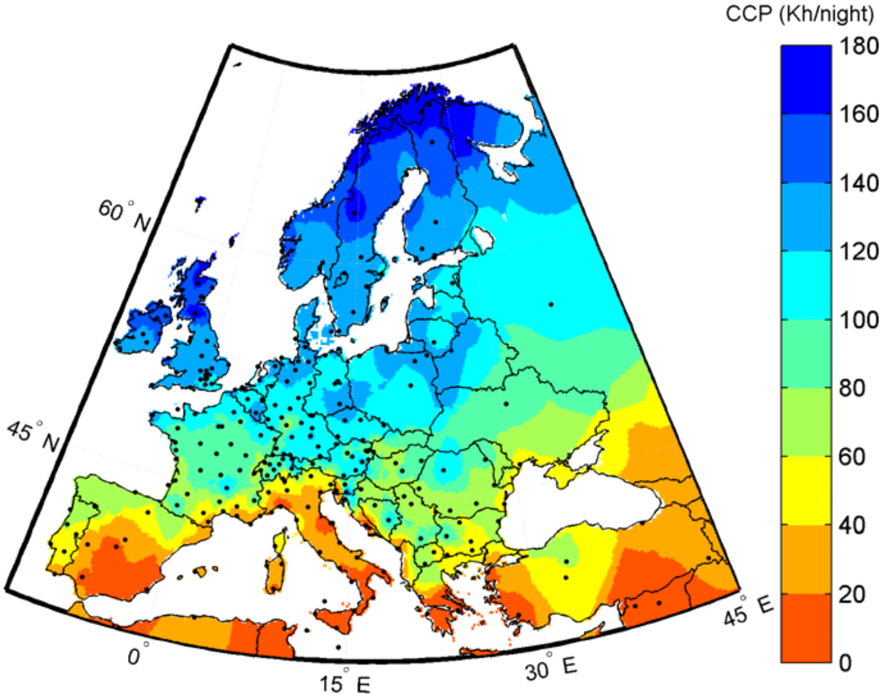
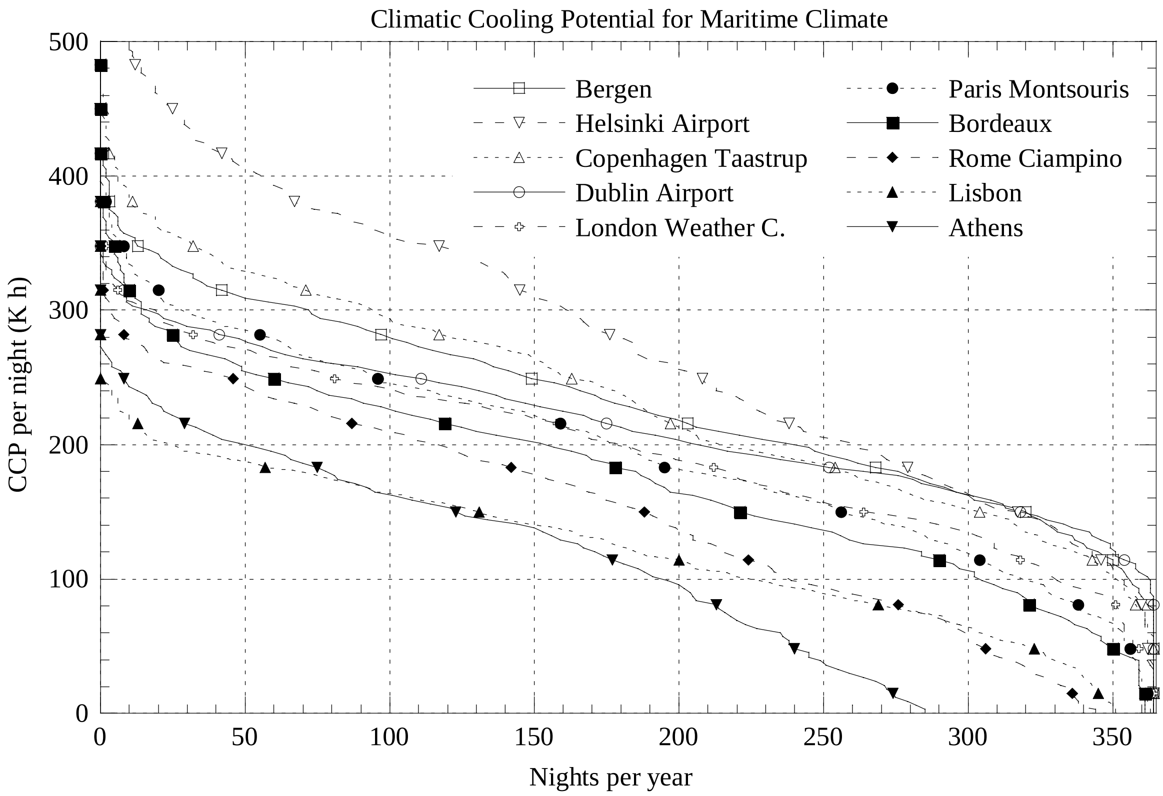


Figure 4. *Map of mean climatic cooling potential (Kh/night) in July based on Meteonorm data [55].*



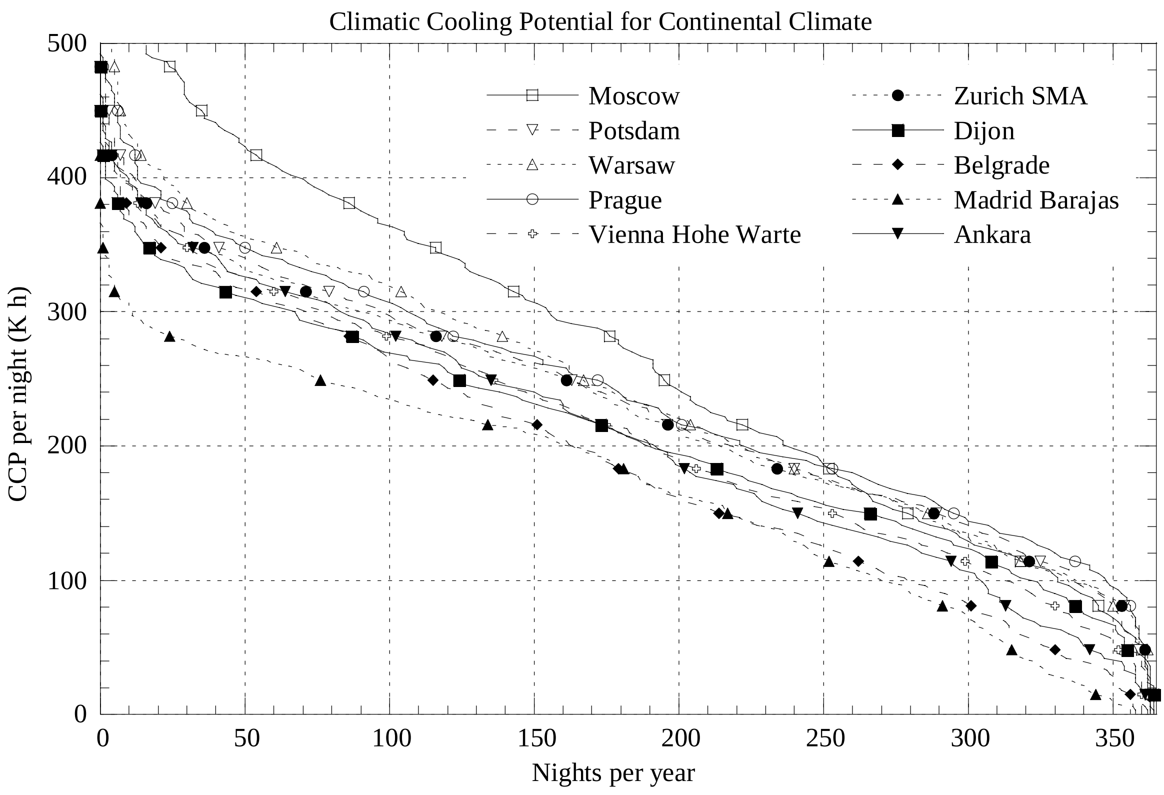


Figure 5. *Cumulative frequency distribution of CCP for maritime (top) and continental (bottom) locations.*

2.4. Impact of climate warming on climatic cooling potential

In order to quantify the impact of climate warming on the potential for night cooling, linear regression models were developed to estimate the daily climatic cooling potential (CCP_d) from the minimum daily air temperature, T_{min}. For eight case study locations representing different climatic zones across a

North-South transect in Europe, CCP was computed for present conditions (1961 - 1990) using measured T_{min} data from the *European Climate Assessment* (ECA) database. Possible future changes in CCP were assessed for the period 2071 - 2100 under the *Intergovernmental Panel on Climate Change* (IPCC) “A2” and “B2” scenarios for future emissions of greenhouse gases and aerosols defined in the *Special Report on Emission Scenarios* (SRES, [56]). The “A2” storyline and scenario family describes a very heterogeneous world. The underlying theme is self-reliance and preservation of local identities. Fertility patterns across regions converge very slowly, which results in continuously increasing population. Economic development is primarily regionally oriented and per capita economic growth and technological change more fragmented and slower than in other storylines. The “B2” storyline and scenario family describes a world in which the emphasis is on local solutions to economic, social and environmental sustainability. It is a world with continuously increasing global population, at a rate lower than “A2”, intermediate levels of economic development, and less rapid and more diverse technological change than in the “A1” and “B1” storylines. While the scenario is also oriented towards environmental protection and social equity, it focuses on local and regional levels. The SRES scenarios do not include additional climate initiatives, which means that no scenarios are included that explicitly assume implementation of the United Nations Framework Convention on Climate Change or the emissions targets of the Kyoto Protocol.

The analysis of climate change impacts was based on 30 *Regional Climate Model* (RCM) data sets obtained from the European *PRUDENCE* project [57]. This project represents an attempt to integrate European climate projections of different institutions, and its website provides a large database of RCM simulation results for Europe. These were based on boundary conditions from 6 global simulations with two *Atmosphere-Ocean General Circulation Models* (AOGCM), Arpege/OPA and ECHAM4/OPYC, plus three atmosphere-only *Global Climate Models* (GCM), ECHAM5, HadAM3H and HadAM3P that were driven with sea-surface temperature and sea-ice boundary conditions taken from simulations with the HadCM3 AOGCM. More information about the climate simulation models can be found on the PRUDENCE website.

For Zurich and Madrid Figure 6 shows significant changes in the percentage of nights per season when the daily cooling potential, CCP_d exceeds a certain value. For Zurich, under current climate conditions CCP_d is higher than 80 Kh (roughly necessary to discharge heat gains of 50 W/m^2, see section 2.2) throughout most of the year, except for about 10 % of summer nights. Under the “A2” scenario CCP_d was found to fall below 80 Kh in more than 50 % (“B2”: 45 %) of summer nights.

For the studied locations in Southern Europe CCP_d values under present climatic conditions were found to be below 80 Kh throughout almost the entire summer, but a considerable cooling potential was revealed in the transition seasons. For the whole year the percentage of nights when CCP_d exceeds 80 Kh in Madrid was found to decrease from 70 % under present conditions to 52 % under “A2” conditions.

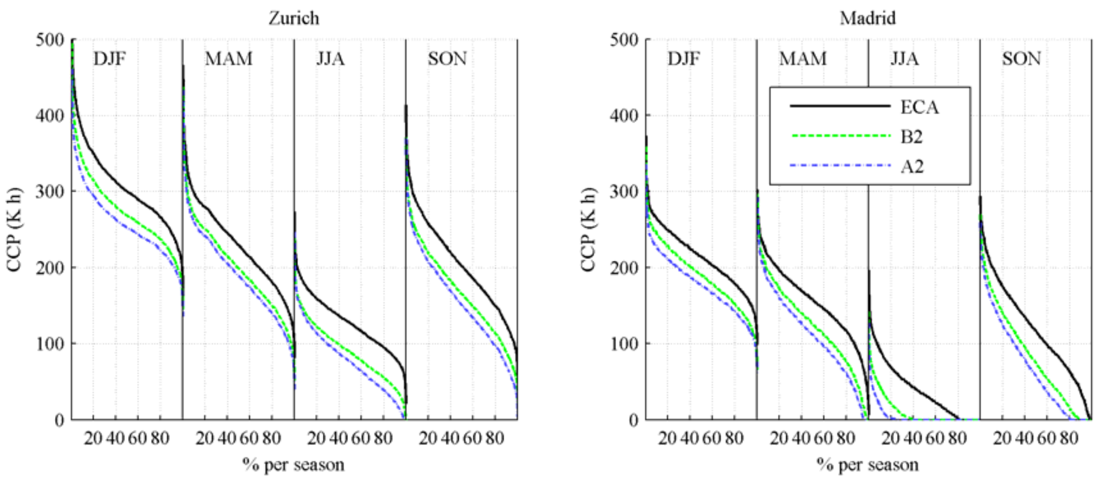


Figure 6. *Seasonal cumulative distributions of CCP_d in Zurich (left) and Madrid (right) for current climate (ECA) and averages for forcing scenarios “A2” and “B2”.*

Transient scenarios were based on upper and lower bounds for the change in global temperature as provided by Cubash et al. [58] (their Figure 9.14, “several models, all SRES envelope”). These bounds accounted for a much broader range of radiative forcing scenarios and possible global temperature responses to a given forcing (climate sensitivities) than the PRUDENCE scenarios. However, it should be noted that these do still not account for the full range of possible future development.

Figure 7 shows a range of the possible development of the mean climatic cooling potential, *CCP* during the summer (June July August) from 1990 to 2100 for three European locations. The lower limit indicates a rapid decrease in night cooling potential, especially after 2030. In the case of Madrid the slope of the lower limit tails out as it approaches zero. The upper limit shows a flatter slope and levels to a constant value at the end of the 21st century.

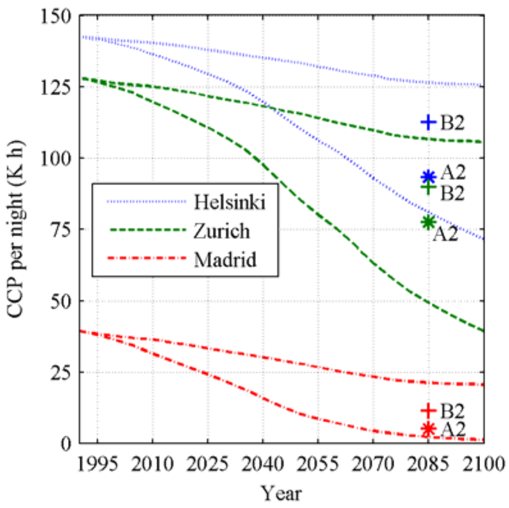


Figure 7. *Time-dependent change in mean climatic cooling potential during summer (JJA); upper and lower scenario based on mean global temperature scenarios ([58] Figure 9.14, "several models, all SRES envelope") and mean values of selected PRUDENCE models for "A2" and "B2".*

The decreases found in mean cooling potential have regionally varying implications. In Northern Europe the risk of thermal discomfort for buildings that use exclusively ventilative night cooling is expected to steadily increase up to possibly critical levels in the second half of the 21st century. In Central Europe extended periods with very low night cooling potential – where thermal comfort cannot be assured based on night-time ventilation only – could already become more frequent in the next few decades, if a strong warming scenario became real. For Southern Europe the potential for ventilative night cooling will sooner or later become negligible during summer and will decrease to critical levels in the transition seasons.

2.5. Concluding remarks

Additionally to climate warming the heat island effect causes a decrease in the potential for night-time ventilation in urban areas compared to surrounding rural areas. Williamson and Erell [59] applied the CCP concept to assess the implications of heat islands for building ventilation. For London they found a reduction in CCP during summer of about 9 %. Even larger effects were found for Adelaide, Australia (up to 26 %) and Sde Boqer, Israel (up to 61 %).

In order to avoid heavily increased energy consumption by mechanical cooling systems there is a great need for additional passive cooling techniques, such as radiant or evaporative cooling, and/or hybrid approaches. However, it should be noted that although cooling by night-time ventilation is expected to become increasingly ineffective during summer, it is likely to remain an attractive option in the transition seasons. This will be even more the case, if it is considered that under general warming the cooling season will tend to start earlier in spring and end later in autumn. In fact, the decreasing cooling potential and the simultaneously increasing cooling demand result in a shift of possible applications of night-time ventilation in Europe from South to North and from summer to the transition seasons.

Any assessment of possible changes in future climate is subject to large uncertainties. Nevertheless, the extent and rate of the expected climatic changes and the long service life of buildings imply the need for designing buildings capable of providing comfortable thermal conditions under more extreme climatic conditions.

3. Dynamic heat storage capacity of building elements

As heat gains and night ventilation periods do normally not coincide in time, the energy of daily heat gains needs to be stored until it can be discharged by ventilation during the following night. A sufficient amount of thermal mass is therefore needed for a successful application of night-time ventilation. For effective utilisation of the thermal mass both a sufficient heat transfer to the surface and sufficient conduction within the element are needed. The purpose of this study was to evaluate the impact of different parameters such as material properties, slab thickness and heat transfer coefficient on the heat storage capacity of building elements [60].

3.1 Model of a building element

A building element was represented by a homogeneous slab with half-thickness d. One surface of the slab was exposed to a varying temperature, while the other surface was considered adiabatic. The analytical solution to the heat transfer problem in a slab with convective boundary condition and sinusoidally varying air temperature ([37], [61]) was used to determine the spatial and temporal temperature profile in the element. Integrating the positive (charging) or negative (discharging) heat flow at the surface over one periodic cycle yields the dynamic heat storage capacity c_{dyn} of the element. This corresponds to the dynamic heat storage capacity as defined in the European standard EN ISO 13786 [62]. A method to calculate the dynamic heat storage capacity of an element composed of layers with different thermal properties is also presented in EN ISO 13786. The dynamic heat storage capacity depends on the time period of the temperature variation. As the performance of night-time ventilation mostly depends on the diurnal heat storage, the capacity was calculated based on a 24 h temperature variation.

3.2. Impact of different Parameters

Figure 8 shows the diurnal heat storage capacity, c_{dyn} of different building elements depending on the heat transfer coefficient h. The properties of the materials are given in Table 1. The impact of the heat transfer coefficient depends greatly on slab thickness and the thermal properties of the material. For thin slabs (d = 15 mm) the heat storage capacity is almost constant for heat transfer coefficients higher than $h = 3\ W/m^2K$. In contrast, for thick slabs increasing the heat transfer coefficient up to $h = 30\ W/m^2K$ significantly increases the diurnal heat storage capacity. Generally, the storage capacity of thin elements, such as gypsum boards used for light-weight wall constructions or medium density fibreboards (MDF) used for furniture is rather small compared to thick and heavy elements such as a concrete ceiling or sand-lime brick walls (bricks made from sand and slaked lime). However, especially at a low heat transfer coefficient and in consideration of its large surface area, furniture might still make a notable contribution to the total heat storage capacity of a room.

Increasing the slab thickness clearly raises the diurnal heat storage capacity until a maximum is reached. Beyond the maximum the capacity decreases slightly and converges to a constant value as the thickness approaches infinity (Figure 9). This somewhat surprising effect has been described previously (e.g. [63]) and is explained by the superposition of an incident wave and a reflected wave. With increasing heat transfer coefficient the maximum becomes more distinct. Additionally the optimum thickness of a concrete slab increases from about d = 90 mm to 140 mm if the heat transfer coefficient increases from $h = 5\ W/m^2K$ to $h = 30\ W/m^2K$.

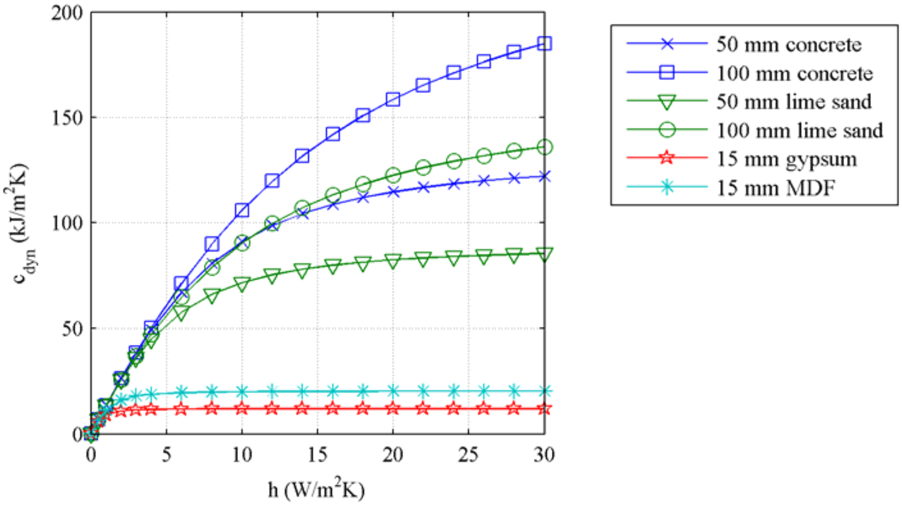


Figure 8. *Diurnal heat storage capacity, c_{dyn} of different building elements depending on the heat transfer coefficient h.*

Table 1. *Material properties.*

	λ (W/m²K)	ρ (kg/m³)	c (kJ/kgK)
Concrete	1.80	2400	1.1
Sand-lime	1.10	2000	0.9
Gypsum	0.40	1000	0.8
MDF	0.18	800	1.7
Gypsum, 20% PCM	0.36	960	9.0*
Gypsum, 40% PCM	0.32	920	17.3*

* In the melting temperature range.

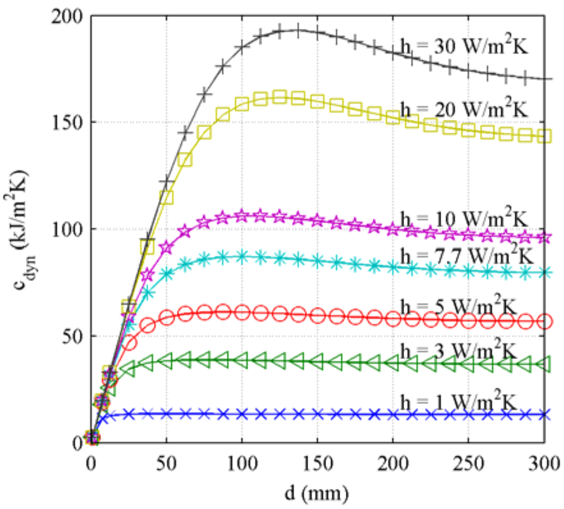


Figure 9. *Diurnal heat storage capacity, c_{dyn} of a concrete slab depending on the thickness, d for different heat transfer coefficients, h.*

Analysing the impact of the thermal conductivity, λ of the slab material showed that in most cases the storage capacity increases only slightly for conductivities above 1.8 W/mK (concrete). For thin slabs (d = 15 mm) there is almost no impact of the conductivity in the range from λ = 0.05 W/mK to λ = 50 W/mK. Only in the case of a very thick slab (d = 100 mm) in combination with a high heat transfer coefficient (h = 20 W/m^{2}K) does the storage capacity increase with conductivities up to 50 W/mK (Figure 10).

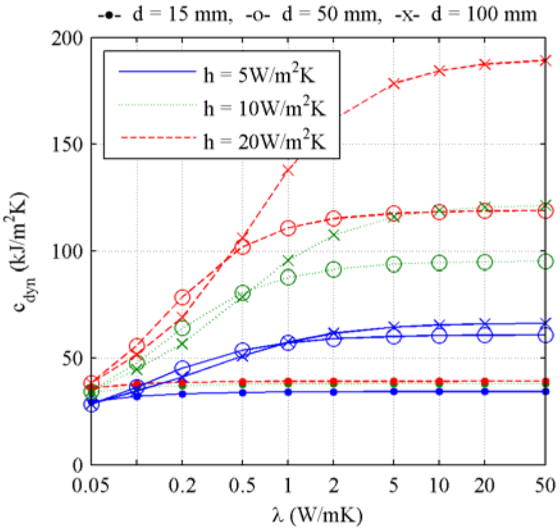


Figure 10. *Diurnal heat storage capacity, c_{dyn} depending on the thermal conductivity, λ for different heat transfer coefficients, h and slab thicknesses, d; ρc = 2.6 MJ/m^{3}K.*

The impact of the volumetric heat capacity, ρc is displayed in Figure 11. The heat storage capacity of very light materials such as insulation materials with ρc < 0.1 MJ/m^{3}K is generally very small. Even for a slab with half-thickness d = 100 mm and a high heat transfer coefficient, h = 20 W/m^{2}K, the heat storage capacity is only 10 kJ/m^{2}K. Increasing the thermal capacity to the value of concrete (ρc = 2.6 MJ/m^{3}K) significantly improves the storage capacity, especially at high heat transfer coefficients (h = 10 to 20 W/m^{2}K) to maximum 158 kJ/m^{2}K. Further improvement for capacities above ρc = 2.6 MJ/m^{3}K is only achieved for thin slabs (d = 15 mm) or at very high heat transfer coefficients (h = 20 W/m^{2}K).

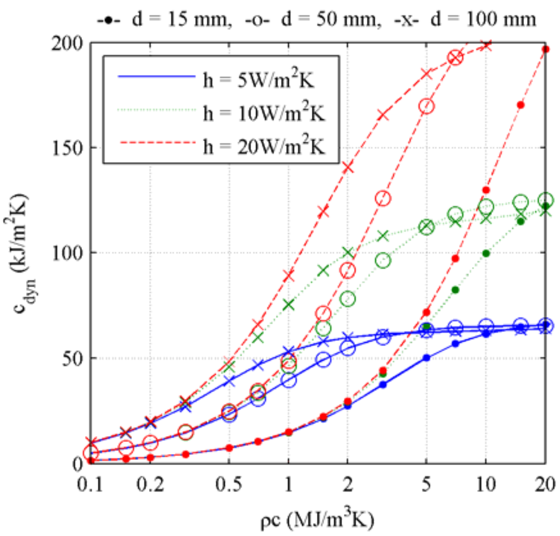


Figure 11. *Diurnal heat storage capacity, c_{dyn} depending on the heat capacity, ρc for different heat transfer coefficients, h and slab thicknesses, d; λ = 1.8 W/mK.*

A possibility for increasing the thermal heat capacity is the integration of micro-encapsulated phase change materials (PCM) into gypsum plaster boards or plaster [10]. For a rough estimation of the effect of PCMs on the dynamic heat storage capacity a very simple model, assuming a constant heat capacity in the melting temperature range, was used in this study. The estimated thermal properties are given in Table 1. Applying this model, the dynamic heat capacity of 15 mm thick gypsum plaster boards with 20 % and 40 % PCM content was found to be similar to a 50 mm and 100 mm thick concrete slab, respectively (Figure 12).

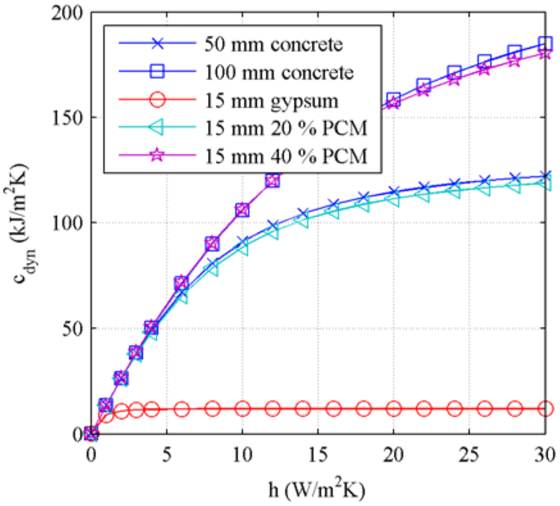


Figure 12. *Effect of integrated PCM on the diurnal heat storage capacity, c_{dyn} depending on the heat transfer coefficient, h. Gypsum plaster board with different PCM contents compared to concrete slabs.*

3.3 Conclusion

The dynamic heat storage capacity of building elements is affected by many parameters, like material properties, thickness and surface heat transfer coefficient. However, the sensitivity to each parameter depends on other parameters. In many cases the capacity is limited by one parameter and changing the limiting parameter has the largest effect. For example, the dynamic heat storage capacity of thin, light-weight elements, like gypsum plaster boards, is limited by the total heat capacity. In this case, the integration of phase change materials can significantly improve the dynamic heat storage capacity.

On the other hand, for massive elements, like a concrete floor slab or sand-lime brick walls, the thermal properties of common building materials are sufficient, but the dynamic heat storage capacity is mainly limited by the heat transfer at the surface. In this case the penetration depth of the temperature variation is limited and the total heat capacity of the element is only partially utilised. Therefore, for massive elements the dynamic heat storage capacity highly depends on the heat transfer coefficient.

4. Simulation study on performance of night-time ventilation

The heat storage capacity of building elements is an important precondition for the application of night cooling. However, the effectiveness of night-time ventilation also depends on other parameters such as internal and solar heat gains, climatic conditions (outdoor air temperature) and ventilation air change rate. Additionally, several building elements with different properties are typically present in a real room. The impact of different parameters and the interaction of different building elements were investigated in detail by building energy simulation [64].

4.1 Simulation model

A typical office room with 20 m^2 floor area (Table 2) was modelled using the building energy simulation programme *HELIOS* [65]. Starting from a base case, different parameters were varied to assess their effect on night ventilation performance. In order to assess the impact of thermal mass, three different building types, representing to a light-weight (suspended ceiling, gypsum board walls), medium-weight (exposed concrete ceiling, gypsum board walls), and heavy-weight (exposed concrete ceiling, solid sand-lime brick walls) construction. The detailed composition of the building elements and the thermal properties of the building materials are given in Table 3. The resulting dynamic heat storage capacity per unit floor area is illustrated in Figure 13.

Table 2. *Main parameters of the modelled office room.*

Floor area	20	m^2
Room height	2.6	m
Volume	52	m^3
Facade area	10.4	m^2
Facade area/volume ratio	0.2	m^{-1}
Internal surface area	86.8	m^2
Ceiling area	20	m^2
Internal wall area	36.4	m^2
External wall area	4.8	m^2
Window area	5.6	m^2
Glazed area (aperture)	4.05	m^2
Glazed area/facade area ratio	38.9	%

Table 3. *Composition of building elements and thermal properties of building materials*

Element	Material	d (m)	λ (W/mK)	ρ (kg/m^3)	c (kJ/kgK)
Floor slab (light-, medium-, heavy-weight)	Carpet	0.005	0.05	80	930
	Plaster Floor	0.08	1.5	2200	1080
	Sound insulation	0.04	0.04	30	1404
	Concrete	0.18	1.8	2400	1080
Suspended ceiling (light-weight)	Air gap	0.25	R = 0.160 m^{2}K/W		
	Acoustic panel	0.02	0.21	800	900
Internal wall (light-, medium-weight)	Gypsum board	0.025	0.4	1000	792
	Mineral wool	0.07	0.036	90	612
	Gypsum board	0.025	0.4	1000	792
Internal wall (heavy-weight)	Plaster	0.015	0.7	1400	936
	Sand-lime brick *	0.15	1.1	2000	936
	Plaster	0.015	0.7	1400	936
External wall (light-, medium-weight)	Concrete	0.18	1.8	2400	1080
	Exp. polystyrene	0.12	0.035	40	1200
	Gypsum board	0.025	0.4	1000	792
External wall (heavy-weight)	Plaster ext.	0.02	0.87	1600	1000
	Exp. polystyrene	0.12	0.035	40	1200
	Sand-lime brick *	0.15	1.1	2000	936
	Plaster	0.015	0.7	1400	936

* Sand-lime brick: Bricks made from sand and slaked lime.

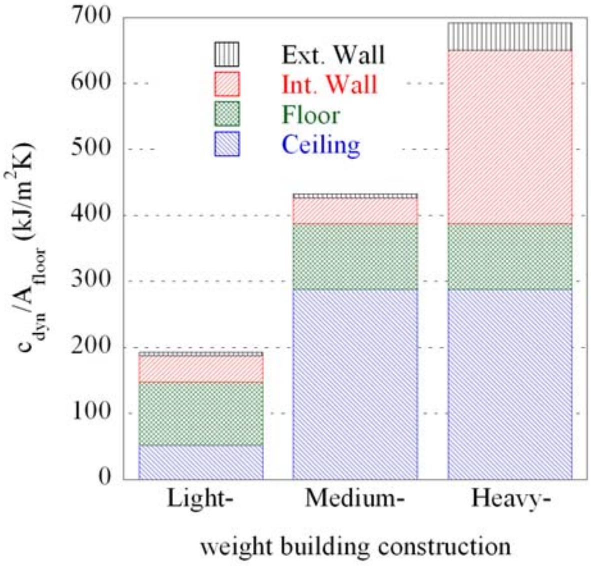


Figure 13. *Dynamic heat storage capacity of building elements for three different levels of thermal mass (EN ISO 13786 [62]; excluding surface resistance).*

Additionally three different levels of internal heat gains (low: 159 Wh/m^{2}d, medium: 229 Wh/m^{2}d and high: 313 Wh/m^{2}d) were defined to show their impact on night ventilation performance. In all cases the office was occupied by two people, with a heat gain of 126 W/person (60 % conv. / 40 % rad.). Heat gains from equipment (80 % conv. / 20 % rad.) were assumed to be 50 W/person (low) or 150 W/person (medium and high). Lighting power was 10 W/m^2 (40 % conv. / 60 % rad.) with different schedules representing an office with (low and medium) and without (high) daylight utilisation (Figure 14). No heat gains were assumed during weekends.

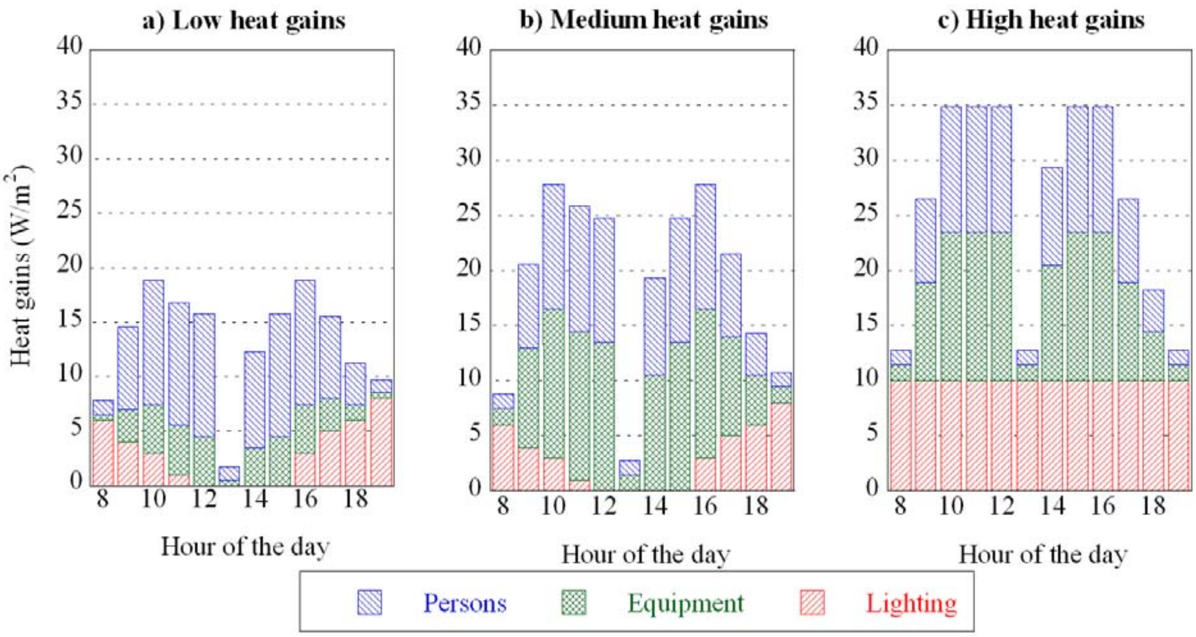


Figure 14. *Low (159 Wh/m^{2}d), medium (229 Wh/m^{2}d) and high (313 Wh/m^{2}d) internal heat gains over the course of a day.*

Night-time ventilation was assumed from 7 pm to 7 am at a fixed air change rate (base case: 6 ACH). The increased air change rate was applied if the external temperature was at least ΔT_{crit} = 3 K below the average room surface temperature, $T_{Surface}$. To prevent over-cooling, night ventilation was terminated as soon as the average room surface temperature, $T_{Surface}$ fell below 20 °C.

Different climatic data sets were applied to assess the impact of local and annual climatic variability. In addition to semi-synthetic data for different locations (*Meteonorm* and *Design Reference Year*, DRY), long-term measured data for Zurich (*ANETZ*) were used.

In order to evaluate the impact of the heat transfer between building elements and the room air, a wide range of surface heat transfer coefficients was investigated. In *HELIOS* convective and radiative heat transfer is modelled using combined heat transfer coefficients. In the base case, all combined heat transfer coefficients were set to h = 7.7 W/m^{2}K in accordance with DIN 4701 [66].

Night-time ventilation performance was rated by evaluating overheating degree hours of the operative room temperature above 26 °C. Different standards (e.g. [54], [67], [68]) give limits of acceptable overheating corresponding to a range from about 100 to 400 Kh above 26 °C per year. The limit to be applied for a certain application mainly depends on the building usage. Generally a wider temperature range will be accepted in residential buildings than in commercial buildings. Furthermore, the tolerated extent of overheating also depends on the occupants' possibilities to adapt themselves, (e.g. by changing clothing) and control their environment (operation of windows, access to building controls, etc.). In order to account for the ability to adaptation the concept of adaptive thermal comfort has been proposed which relates the comfort temperature to the external temperature [69], [70]. In this study 100 Kh above 26 °C was considered good thermal comfort, while up to 400 Kh was still considered acceptable.

4.2. Impact of different parameters

Figure 15 shows the overheating degree hours above 26 °C (ODH_{26}) in a medium thermal mass office with high internal heat gains based on different climatic data for Zurich SMA. In the period from 1981 to 2005 (excluding 2003), overheating degree hours exceeding 26 °C varied in the range from 20 to 400 Kh if measured weather data (ANETZ) was used. In 2003, exceptionally high summer temperatures resulted in nearly 900 Kh above 26 °C, while the mean value from 1981 to 2005 was 142 Kh/a.

Simulations based on Meteonorm or DRY data resulted in a far lower extent of overheating. Overheating degree hours amounted to 2.3 Kh/a if the Meteonorm standard data were applied, and 79 Kh/a when Meteonorm data based on 10-year maximum monthly mean values were used. Applying DRY data (based on measurements from 1981-1990) resulted in 25 Kh/a, while ANETZ data averaged over the same period yielded 82 Kh/a. For the following simulations ANETZ data from the period 1996 to 2005 were applied.

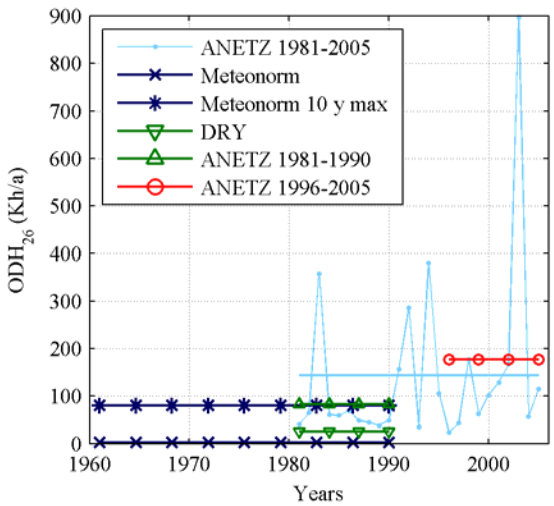


Figure 15. *Overheating degree hours above 26 °C (ODH_{26}) based on different climatic data for Zurich SMA: ANETZ 1981-2005, Meteonorm (standard and 10-year maximum), DRY, ANETZ 1981-1990 (mean) and ANETZ 1996-2005 (mean); medium thermal mass, high level of internal heat gains, air change rate 6 ACH, all heat transfer coefficients h = 7.7 W/m^{2}K.*

Significant effects of both different building constructions (thermal mass) and different levels of internal heat gains are shown in Figure 16. While in most cases thermal conditions were quite comfortable with less than 200 Kh/a, hardly acceptable overheating (ODH_{26} = 416 Kh/a) resulted for the extreme case of a light weight office with high internal loads. A 180 mm thick concrete ceiling in direct contact with the room air reduced overheating by a factor of two compared to a suspended ceiling. The additional application of solid sand-lime brick walls instead of gypsum board walls reduced overheating by a factor of three.

Internal heat gains had an effect of similar magnitude. Both increasing the heat gains of office equipment from 50 W to 150 W per person and using artificial light instead of daylight, respectively, increased overheating degree hours by a factor of 2 to 2.5. These factors depended on thermal mass, with a smaller sensitivity in heavier buildings. Especially in light-weight buildings, heat gains should be reduced as far as possible by applying energy-efficient office equipment, daylight utilisation, and the installation of effective solar shading devices.

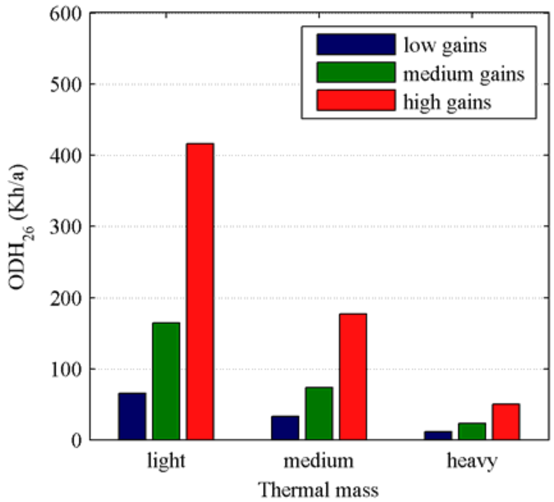


Figure 16. *Overheating degree hours above 26 °C for different building constructions (light, medium and heavy mass) and different levels of internal heat gains (low, medium and high), Zurich (ANETZ 1996-2005), 6 ACH, all heat transfer coefficients h = 7.7 W/m²K.*

Figure 17 (left) shows the overheating degree hours above 26 °C as a function of air change rate during night-time ventilation (0.5 - 32 ACH) for high heat gains and different building constructions. In case of a medium- and heavy weight building, a relatively low air change rate below 4 ACH was sufficient to maintain the limit of thermal comfort (prEN 15251 [54], category III; about 400 Kh). In the light-weight building an air change rate of about 7 ACH is needed to keep this limit.

Increasing the night ventilation air change rate improved thermal comfort significantly. For the example of a heavy-weight building with high internal heat gains, overheating was reduced from 419 to 32 Kh/a when the air change rate was increased from 2 to 8 ACH. However, applying a higher air change rate than 8 ACH did not improve thermal comfort significantly. The critical air change rate beyond which no significant improvement in absolute ODH_{26} values occurred, increased with decreasing thermal mass.

Variation of the combined heat transfer coefficients (convection and radiation) in the range from 5.9 to 10 W/m^{2}K (EN ISO 6946 [38]) generally had a minor effect on overheating degree hours (Figure 17, right). The most significant effect was found for a light-weight office, where overheating degree hours were reduced from 464 to 377 Kh/a. Only little improvement was found for increased heat transfer coefficients (HTC) in the range from 10 to 22 W/m^{2}K. However, night-time ventilation becomes increasingly inefficient for HTCs below about h = 4 W/m^{2}K.

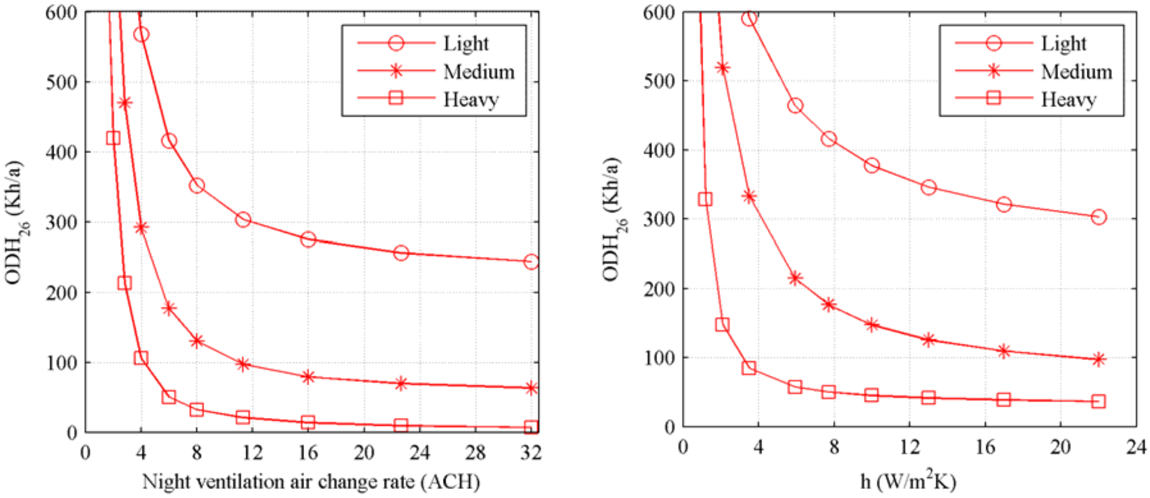


Figure 17. *Overheating degree hours above 26 °C (ODH_{26}) for high internal heat gains and different building constructions; Zurich, ANETZ 1996-2005; left: depending on the air change rate, h = 7.7 W/m²K; right: depending on the heat transfer coefficients (all surfaces, day and night), air change rate: 6 ACH.*

Changing the air flow rate during the day mainly affected thermal comfort in cases where night-time ventilation is not very effective, i.e. in light or medium-weight buildings together with a high or medium level of internal heat gains (Figure 18). In such cases, heat gains were partly discharged during the day and overheating decreased with increasing air change rate. In contrast, overheating increased slightly in cases with very good thermal comfort (*ODH_{26}* < 50 Kh/a), when the air flow rate during the day was increased. In cases with good night ventilation performance with daytime indoor air temperatures below the ambient temperature, an increased air flow rate during the day causes additional heat gains.

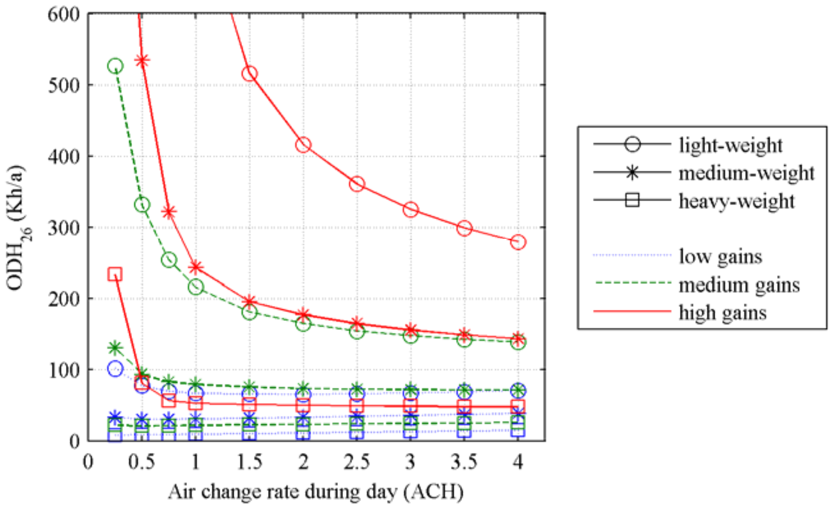


Figure 18. *Overheating degree hours (ODH) above 26 °C as a function of air change rate during the day for different building constructions and different levels of internal heat gains; 6 ACH; heat transfer coefficients h = 7.7 W/m²K (all surfaces, day and night); Zurich, ANETZ 1996-2005.*

4.3. Discussion and conclusions

Night-time ventilation performance was found to be highly sensitive to climatic conditions, which demonstrates the significance of high-quality climatic data for building energy simulations. Furthermore, simulations based on commonly-used semi-synthetic one year data sets such as Meteonorm or DRY data tend to underestimate the extent of overheating compared to weather data measured in recent years. Therefore, for reliable thermal comfort predictions the application of up-to-date climatic data from long-term measurements including extreme weather conditions is recommended. Because of the gradually warming of the global climate, continuous updating of weather data for building simulation is needed.

Simulations with different night ventilation air change rates clearly demonstrated the effectiveness of night-time ventilation, as increased flow rates significantly reduced overheating degree hours. However, increasing the air flow rate above a certain value – a critical air flow rate depending on building construction and heat gains – did not lead to further improvement worth mentioning. On the other hand, the very high sensitivity to night ventilation air change rates up to about 4 ACH makes predictions of thermal comfort very uncertain. This is especially true for buildings using natural night ventilation, where the air change rate depends on ambient temperature and wind conditions. If natural ventilation is driven by buoyancy forces, low air flow rates coincide with a high external temperature, and the cooling effect is smallest during warm periods. In cases where air change rates in a range of high sensitivity are to be expected, the application of a hybrid ventilation system should be considered. A shortfall of the critical air flow rate can thus be prevented and the risk of overheating can be reduced.

Due to high air change rates and high air flow velocities, increased heat transfer is expected during night-time ventilation. However, only little improvement of thermal comfort was found for heat transfer coefficients in the range from 10 up to 22 W/m^{2}K. On the other hand, if all surfaces have a similar temperature which is higher than the room temperature – a typical situation during night-time ventilation – a combined heat transfer model overestimates the radiative heat flow. Heat transfer coefficients for convection alone, however, can be as low as 0.7 W/m^{2}K (EN ISO 13791 [39]; downward heat-flow at horizontal surfaces). Especially at the ceiling (downward heat-flow during night-time ventilation) free convection might be very limited and the reduced heat transfer could impair the efficiency of night-time ventilation. The simulation study showed a very high sensitivity for heat transfer coefficients below about h = 4 W/m^{2}K. The wide range of possible heat transfer coefficients and the partially high sensitivity can cause major uncertainties in thermal comfort predictions. Therefore, convective and radiative heat transfer during night-time ventilation needs to be investigated in more detail.

5. Experiments on heat transfer during night-time ventilation

The convective heat transfer at internal room surfaces is highly affected by the indoor air temperature distribution and the near-surface velocities both of which can vary significantly depending on the air flow pattern in the room. Additionally, if the surfaces of a room differ in thermal capacity or heat transfer, their temperature will also differ and radiation becomes significant. This study provides a detailed analysis of convection and radiation during night-time ventilation depending on the air flow rate and the initial temperature difference between the inflowing air and the room. Heat transfer in case of mixing and displacement ventilation has been investigated in a full scale test room [71].

5.1. Setup of the test room

A test room at Aalborg University – a wooden construction insulated with 100 mm rock wool – was rebuilt for the experimental investigation of the heat transfer during night-time ventilation. For increased thermal mass a heavy ceiling element consisting of 7 layers of 12.5 mm gypsum boards was installed [72]. The walls and the floor were insulated with 160 mm (floor: 230 mm) expanded polystyrene (EPS). After installation of the insulation the internal dimensions were 2.64 m x 3.17 m x 2.93 m (width x length x height) resulting in a volume of 24.52 m^3. A detailed description of the test room can be found in [73].

A mechanical ventilation system was installed to supply air at a defined temperature to the test room, providing an air flow rate ranging from 2.3 to 13 air changes per hour (ACH). Two different air distribution principles representing mixing and displacement ventilation were investigated (Figure 19).

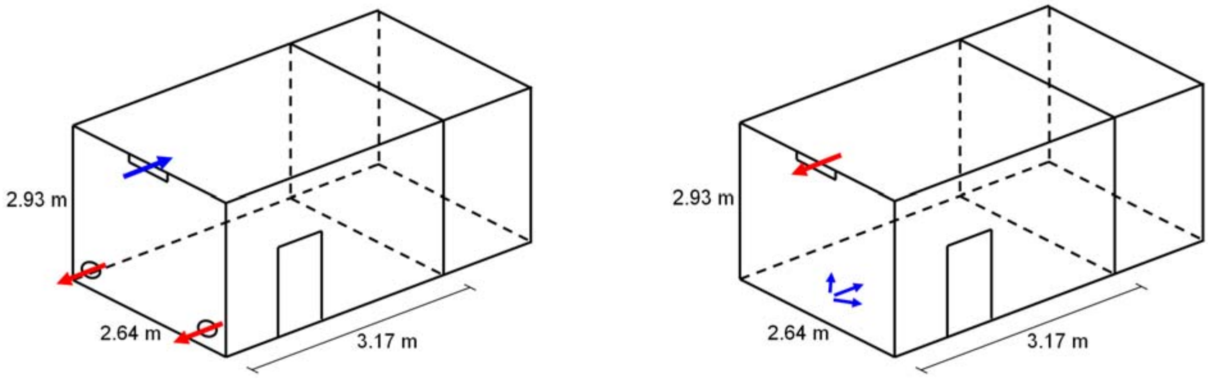


Figure 19. *Configurations of the air in- and outlet openings of the test room for mixing (left) and displacement ventilation (right).*

In total 110 thermocouples were installed in the ceiling element to measure the temperature at 22 positions and 5 different layers (including the internal surface). 18 hot sphere anemometers were installed to measure the air flow velocity close to the ceiling. At the wall and floor surfaces temperatures were measured at 3 positions each. Additional thermocouples were installed to measure air temperatures close to the surfaces and in the symmetric plane of the room. Also the in- and outflowing air temperatures were measured by thermocouples. The air flow rate was measured using an orifice and a micro-manometer.

5.2. Procedure for experiments and data evaluation

In each experiment the response of the test room to a step in the air flow rate (inflow temperature below room temperature) was measured for at least 12 hours. In total 16 experiments with different air distribution modes, air change rates (*ACR*) and initial temperature differences (ΔT_0) were conducted (Table 4).

Table 4. *List of experiments.*

No	Air distribution mode	*ACR* (ACH)	ΔT_0 (K)
1	Mixing ventilation	2.3	7.9
2	Mixing ventilation	3.3	4.3
3	Mixing ventilation	3.3	10.2
4	Mixing ventilation	6.7	2.9
5	Mixing ventilation	6.8	6.1
6	Mixing ventilation	6.6	8.9
7	Mixing ventilation	13.1	2.9
8	Mixing ventilation	13.2	4.0
9	Mixing ventilation	13.1	5.3
10	Mixing ventilation	13.3	9.2
11	Displacement ventilation	3.1	10.1
12	Displacement ventilation	6.7	5.8
13	Displacement ventilation	6.7	11.3
14	Displacement ventilation	12.6	3.6
15	Displacement ventilation	12.6	6.0
16	Displacement ventilation	12.7	12.7

By way of example, Figure 20 shows temperature profiles measured during experiment no. 4, with mixing ventilation with 6.7 ACH and an initial temperature difference of ΔT_0 = 2.9 K. Profiles A to E correspond to different layers of the ceiling element, where A is the internal surface and E is the external surface of the gypsum boards.

For the evaluation of the heat transfer at the internal room surfaces, first the total surface heat flow (conduction in the material) for each section (22 at the ceiling and 3 at the other surfaces) was calculated. The conductive surface heat flow was determined by applying the measured boundary conditions (A and E in Figure 20) to a transient 1-dimensional finite difference model using an explicit scheme. Running the model resulted in the spatial temperature profile for each time step. For each section, *i*, the conductive heat flux, $\dot{q}_{cond,i}$ at the surface was calculated from the spatial temperature gradient. The radiative heat flux, $\dot{q}_{rad,i}$ between the surfaces was determined from the measured surface temperatures and the view factors. The difference between conduction and radiation then yielded the convective heat flux, $\dot{q}_{conv,i}$ for each section. The total convective heat flow, $\dot{q}_{conv,tot}$ was then yielded from integrating $\dot{q}_{conv,i}$ over all surfaces.

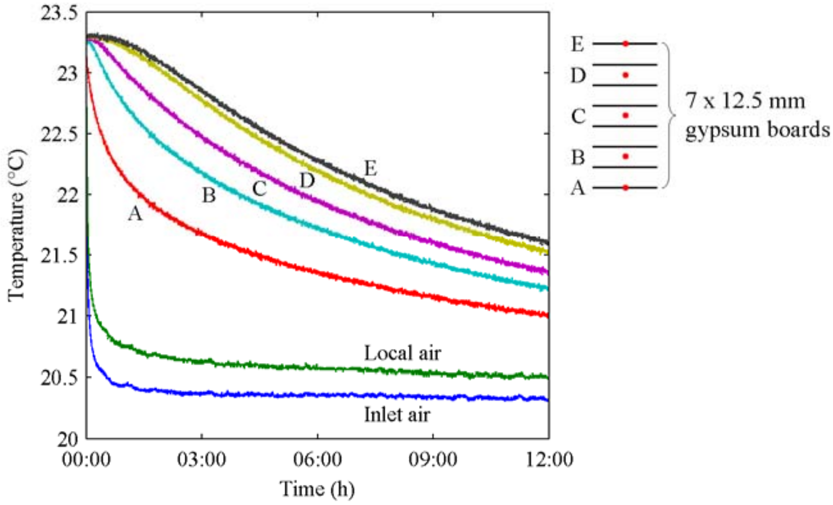


Figure 20. *Temperatures measured at the ceiling during the experiment no. 4 with mixing ventilation with 6.7 ACH and an initial temperature difference of ΔT_0 = 2.9 K (A-E: different layers of the ceiling, A: internal surface, E: external surface).*

5.3. Experimental results

To compare temperature profiles in different experiments the dimensionless Temperature, θ was defined as,

$$\theta = \frac{T - T_{Inlet}}{\Delta T_0}, \tag{6}$$

so that θ = 0 corresponds to the inlet air temperature and θ = 1 to the initial room temperature.

To characterises the flow pattern of the inflowing air jet the 'dimensional' Archimedes number, *Ar'* was defined as,

$$Ar' = \left(\overline{T}_{Surface} - T_{Inlet}\right) / \dot{V}^2, \qquad \left(Ks^2/m^6\right). \tag{7}$$

In mixing ventilation with a small Archimedes number (small temperature difference, high flow rate) the inlet air jet is attached to the ceiling, while for a high Archimedes number (large temperature difference, low flow rate) the air tends to drop down when entering the room.

Figure 21 shows vertical temperature profiles and horizontal temperature profiles 30 mm below the ceiling in experiments with mixing and displacement ventilation, and each with a low and high Archimedes number. For small Archimedes numbers, due to the higher air flow rate the dimensionless room air temperature is generally lower, i.e. closer to the inlet air temperature.

In the experiment with mixing ventilation and low Archimedes number (experiment no. 7) the temperature profiles clearly show the cold air jet close to the ceiling. Along the centre line the temperature increases almost linearly (Figure 21, c)). In the lower part of the room the air is mixed well, i.e. the temperature distribution is homogeneous.

In case of mixing ventilation and a high Archimedes number (experiment no. 3) the cold inflowing air drops down. Except for a small area close to the inlet opening, the air temperature close to the ceiling is very high and homogeneously distributed.

In the experiments with displacement ventilation (experiments no. 11 and 14) the temperature profiles are similar, showing a clear stratification with the largest gradient close to the ceiling. Along the ceiling the air temperature is distributed homogeneously with a slight decrease towards the outlet opening.

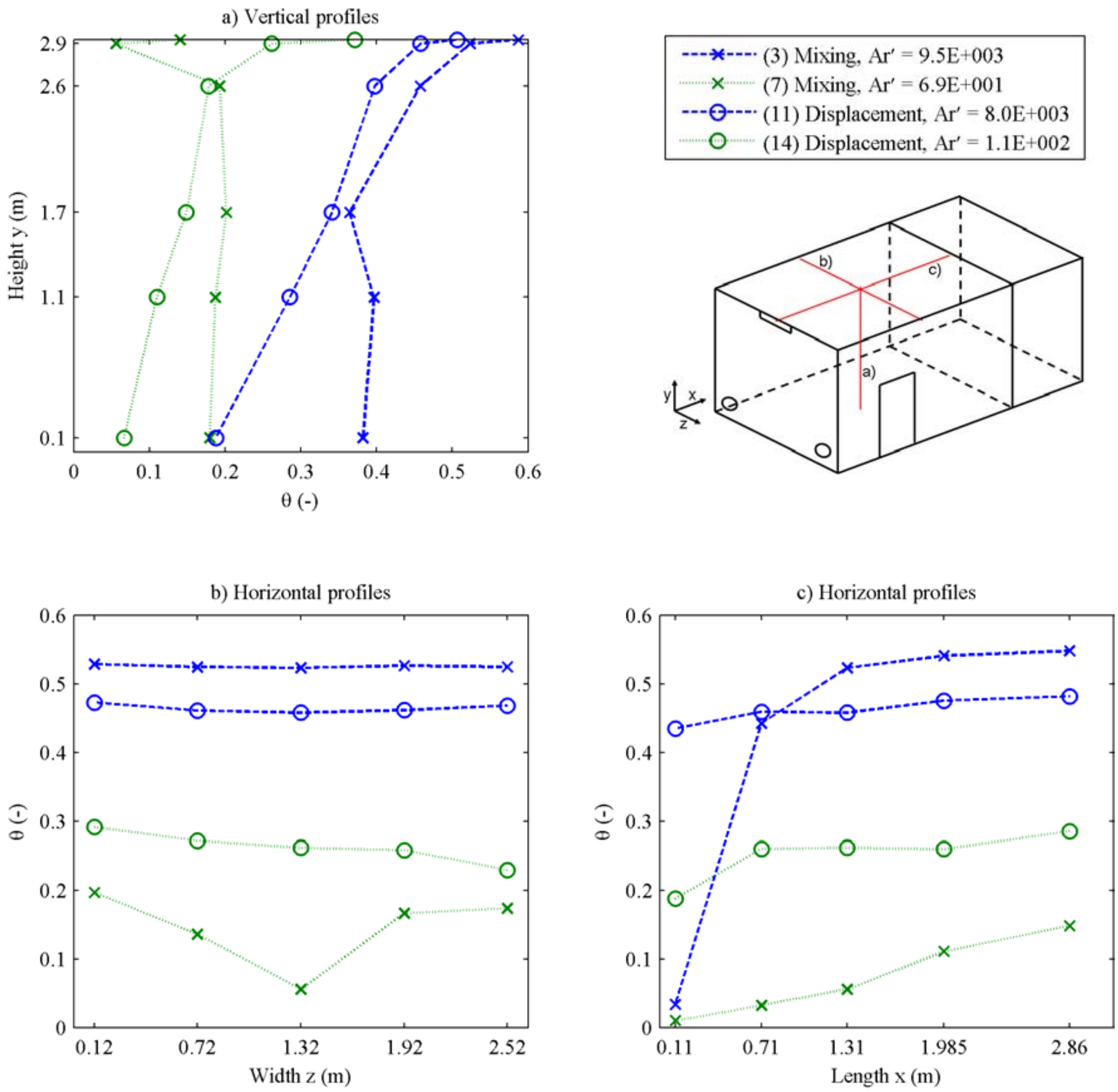


Figure 21. *Temperature profiles (dimensionless temperature θ) comparing experiments no. 3, 7, 11 and 14; a) Vertical profiles (x = 1.31 m, z = 1.32 m); b), c) Horizontal profiles close to the ceiling (y = 2.9 m, b) x = 1.31 m, c) z = 1.32 m); mean values of the 12th hour.*

The different air flow pattern and the development of the air jet also affect the heat transfer at the ceiling. To demonstrate this, the ratio of the convective and the total heat flow from the ceiling was defined as,

$$\gamma = \frac{\dot{Q}_{conv,Ceiling}}{\dot{Q}_{cond,Ceiling}}. \tag{8}$$

Figure 22 shows the convection ratio, γ depending on the 'dimensional' Archimedes number, *Ar'*. During experiments with mixing ventilation, for small Archimedes numbers the inlet air jet is attached to the ceiling and the convection ratio is large. For higher Archimedes numbers the jet tends to drop down. In this case a smaller proportion of the total heat flow is due to convection and radiation becomes dominant. In displacement ventilation, the air flow pattern does not change depending on buoyancy effects and the impact of *Ar'* is small. In all experiments with displacement ventilation less than 32 % of the heat flow from the ceiling is due to convection.

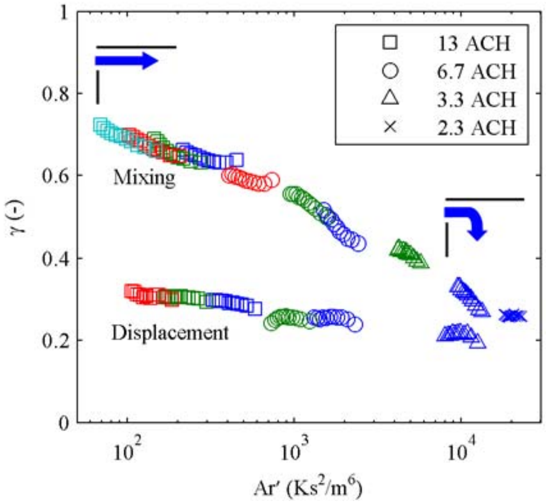


Figure 22. *Ratio γ of convective to total heat flow from the ceiling depending on Ar' for mixing and displacement ventilation; hourly values, first hour excluded. Uncertainty estimate for γ: ± 0.11 [73].*

Comparing different experiments shows that the mean convective heat flux from all internal room surfaces scales linearly with the difference between the mean surface temperature and the inflowing air temperature. This means, that the average heat transfer coefficient, $h' = \dot{q}_{conv,tot} / (\overline{T}_{Surface} - T_{Inlet})$, defined using the inlet air temperature instead of the room air temperature, is almost constant during each experiment and for experiments with different inflowing air temperatures (Figure 23).

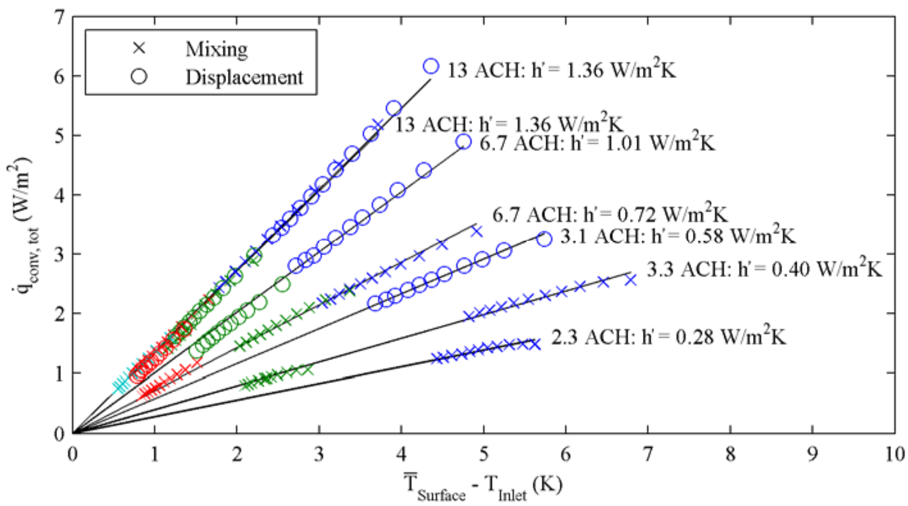


Figure 23. *Mean convective heat flux from all surfaces depending on the difference between the mean surface temperature and the inlet air temperature; hourly values, first hour excluded; different colours relate to different experiments. Uncertainty estimate for $\dot{Q}_{conv,tot}$: ± 16 % [73].*

The performance of night-time ventilation can be described by the temperature efficiency of the ventilation:

$$\eta = \frac{T_{Outlet} - T_{Inlet}}{\overline{T}_{Surface} - T_{Inlet}} \tag{9}$$

The temperature efficiency yielded from the measurements mainly depends on the air distribution mode and the air change rate (Figure 24). During each experiment (excluding the first hour) and for different inlet air temperatures the efficiency is almost constant.

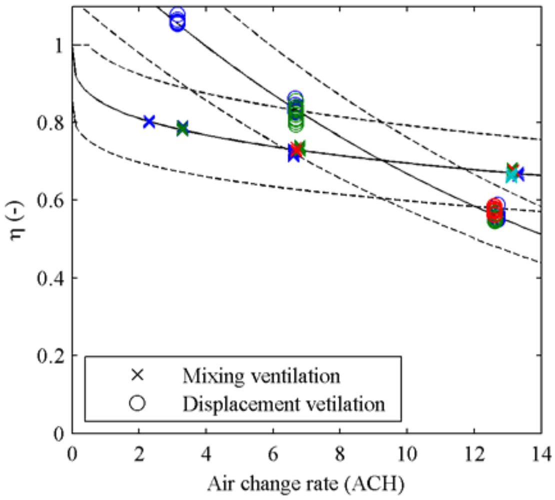


Figure 24. *Temperature efficiency η depending on the air change rate for mixing and displacement ventilation; hourly values, first hour excluded. Fitted curves with estimated uncertainty bands (± 14 % [73])*

In a perfectly mixed room, during night-time cooling ($T_{Inlet} < \overline{T}_{Surface}$ and without internal heat sources) the temperature efficiency is limited to 1. The values found in experiments with mixing ventilation decrease slightly with increasing air change rate. In displacement ventilation at small air flow rates, the temperature stratification can result in an efficiency exceeding 1. For higher air change rates, the decrease in the efficiency is more distinct for displacement ventilation than for mixing ventilation.

5.4. Conclusion

The experimental results clearly demonstrated the interaction of convective and radiative heat flows contributing to the total heat flow removed from a room during night-time ventilation. For mixing ventilation, different flow characteristics significantly affect the ratio of the convective to the total heat flow from the ceiling. In cases with low convective heat transfer at the ceiling (mixing ventilation at high Archimedes number or displacement ventilation) large differences in surface temperatures cause higher radiative heat flows from the ceiling to the floor.

Nonetheless, it is beneficial to prevent warm air from accumulating below the ceiling. In displacement ventilation this is achieved through the location of the outlet opening close to the ceiling. At low air flow rates, the temperature stratification and the high location of the outlet opening results in a very high temperature efficiency ($\eta > 1$). In mixing ventilation warm air should be removed from the ceiling by the inflowing air jet. However, only at a relatively high air change rate (above about 10 ACH) the effect of the air jet flowing along the ceiling becomes significant and mixing ventilation is more efficient. Therefore, if a low air flow rate is expected, the outlet opening should be placed as close to the ceiling as possible.

6. Design method for concept design phase

The findings from the previously described studies shall be combined to develop a practicable method for the estimation of the potential for cooling by night-time ventilation during an early stage of design.

6.1. Design method

The climatic conditions can be characterised by the climatic cooling potential, *CCP* (Kh), the thermal mass of the building can be estimated by the dynamic heat storage capacity, c_{dyn} (kJ/m^{2}K), and the air flow pattern including the resulting heat transfer can be described by the temperature efficiency of the ventilation, η.

In order to assure thermal comfort two criteria need to be satisfied:

1. The thermal capacity of the building needs to be sufficient to accumulate the daily heat gains within an acceptable temperature variation.
2. The climatic cooling potential and the effective air flow rate need to be sufficient to discharge the stored heat during the night.

If both criteria are satisfied for each day of the year, the building temperature will stay within the comfort range which was defined to assess the amount of thermal mass and the climatic cooling potential. As the heat gains and the climatic cooling potential are highly variable, it might not be possible to satisfy both criteria for every day of the year. In this case a certain extent of overheating will occur especially during hot periods, when high heat gains coincide with a low cooling potential. Assessing the extent of overheating during hot spells requires a dynamic model considering all parameters jointly. However, for a first estimation during the concept design phase, the two criteria can be considered separately. Furthermore each day/night cycle is considered separately, and therefore dynamic effects with time periods longer than 24 hours are not taken into account.

To assess the necessary amount of thermal mass, the daily heat gains, Q_d (Wh) need to be estimated and the maximum acceptable daily temperature variation, ΔT (K) needs to be determined. For the internal heat gains, the heat from people, equipment and lighting need to be considered. The solar heat gains can be estimated from the solar irradiation, the glazed area and the total solar energy transmittance (g-value) of the facade. If the outdoor temperature is higher than the indoor temperature, additional heat gains arise from thermal transmission and ventilation during the day.

The total amount of dynamic heat storage capacity, C_{dyn} (J/K) can then be estimated as,

$$C_{dyn} = \sum_i c_{dyn,i} \cdot A_i > \frac{Q_d \cdot 3600\, s/h}{\Delta T} \tag{10}$$

In the calculation of the heat storage capacity, c_{dyn} of the building elements in the room, the surface heat transfer during the day needs to be considered (see Figure 8). In the European Standard prEN 15251 [54] ranges for thermal comfort are given in three categories, A: 2 K, B: 3 K and C: 5 K. According to the ASHRAE Standard 55 [70] the thermal comfort range is 5 K (PPD: 10 %) to 7 K (PPD: 20 %). Additionally, the maximum temperature drift for a 4 h period is given as 3.3 K. Generally broader temperature ranges are acceptable in non-air-conditioned buildings [74].

The estimation of the air change rate and daily cooling potential, CCP_d necessary to discharge daily heat gains, Q_d is based on the conservation of energy during one daily cycle of charging and discharging the thermal mass:

$$Q_d = \eta \cdot A_{Floor} \cdot H \cdot \frac{ACR}{3600\, s/h} \cdot \rho_{Air} \cdot c_{p,Air} \cdot CCP_d \tag{11}$$

where η is the temperature efficiency of the ventilation, A_{Floor} the floor area, H the room height, ρ_{Air} the density of air and $c_{p,Air}$ the heat capacity of air. The daily cooling potential, CCP_d available at a certain location can be determined from cumulative distributions as presented in Figure 5.

Rearranging equation 11 yields the daily heat gains per unit floor area and CCP_d (Wh/m^{2}Kh), which can be discharged at a given air change rate (ACH) depending on the temperature efficiency, η:

$$\frac{Q_d}{A_{Floor} \cdot CCP_d} = \eta \cdot H \cdot \frac{ACR}{3600\, s/h} \cdot \rho_{Air} \cdot c_{p,Air} \tag{12}$$

For the temperature efficiency, found in the experiments with mixing and displacement ventilation (section 5) this correlation is shown in Figure 25. This diagram provides a descriptive illustration of night-time ventilation performance. Additionally, such a diagram can be used as design chart to estimate the air flow rate, necessary to discharge daily heat gains at a given CCP_d. However, in order to make this method more generally applicable, further work is needed to determine the temperature efficiency for various room geometries, constructions (amount and location of thermal mass) and air distribution principles (location of inlet and outlet openings). A catalogue of design charts for various cases will be of immediate use for estimating the performance of night-time ventilation during the concept design phase.

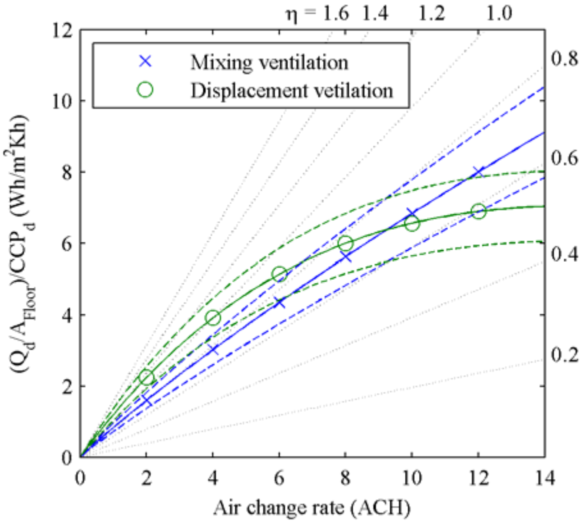


Figure 25. *Heat discharged during night-time ventilation per unit floor area and daily CCP depending on the air change rate; based on experimental results for mixing and displacement ventilation (including range of uncertainty [73]) and lines of constant efficiency η; room height H = 2.93 m.*

6.2. Example of application

An example shall be given to illustrate the application of the previously described design method for the office room defined for the simulation study (see section 4) located in Zurich:

1. Heat gains

Three different levels of internal heat gains, low (159 Wh/m^2), medium (229 Wh/m^2) and high (313 Wh/m^2) were defined for the parameter study.

Additionally, solar heat gains need to be considered. In the simulation model, the office room had two windows (total glazed area, A_{Glass} = 4.05 m^2) on the South façade. With the solar shading device closed, the total solar energy transmittance, g was 16 %. Assuming solar irradiation on a sunny day in July of I = 2000 Wh/m^2 yields:

$$\frac{Q_{d,Solar}}{A_{Floor}} = \frac{A_{Glass}}{A_{Floor}} \cdot g \cdot I = \frac{4.05\ m^2}{20\ m^2} \cdot 0.16 \cdot 2000 \frac{Wh}{m^2} = 65 \frac{Wh}{m^2} \tag{13}$$

For the total daily heat gains follows:

- Low gains: Q_d / A_{Floor} = 224 Wh/m2
- Medium gains: Q_d / A_{Floor} = 294 Wh/m2
- High gains: Q_d / A_{Floor} = 378 Wh/m2

2. Thermal mass

For the calculation of the dynamic heat storage capacity of the building elements the heat transfer coefficient was assumed as h = 7.7 W/m^{2}K. For the three cases defined for the simulation study the dynamic heat storage capacity per unit floor area amounts to [62]:

- Light-weight: C_{dyn} / A_{Floor} = 131 kJ/m2K
- Medium-weight: C_{dyn} / A_{Floor} = 180 kJ/m2K
- Heavy-weight: C_{dyn} / A_{Floor} = 287 kJ/m2K

Applying equation 10, the maximum daily temperature variation, ΔT resulting for the different cases can be determined (Table 5). Aiming a maximum daily temperature variation of 5 to 6 K leads to the following conclusions:

- Low gains: Light- or medium-weight construction
- Medium gains: Medium-weight construction
- High gains: Heavy-weight construction

Table 5. *Maximum daily temperature variation ΔT (K) depending on building construction and daily heat gains.*

	Low gains (224 Wh/m^2)	Medium gains (294 Wh/m^2)	High gains (378 Wh/m^2)
Light-weight (131 kJ/m^{2}K)	6.2 K	8.1 K	10.4 K
Medium-weight (180 kJ/m^{2}K)	4.5 K	5.9 K	7.6 K
Heavy-weight (287 kJ/m^{2}K)	2.8 K	3.7 K	4.7 K

3. Temperature efficiency

For this example the temperature efficiency found in the experiments (section 5) during mixing ventilation was assumed. The equation for the fitted curve is

$$\eta = 1 - 0.1538 \cdot ACR^{0.2966} . \qquad (14)$$

4. Climatic cooling potential

The climatic cooling potential is computed according to the definition in section 2.1. (equation 1) based on ANETZ data measured at Zurich SMA for the 10 years period 1996 to 2005. Figure 26 shows the cumulative frequency of CCP during the 50 warmest nights per year. In 15 nights per year CCP is below 60 Kh and there are 2 - 3 nights per year without any cooling potential.

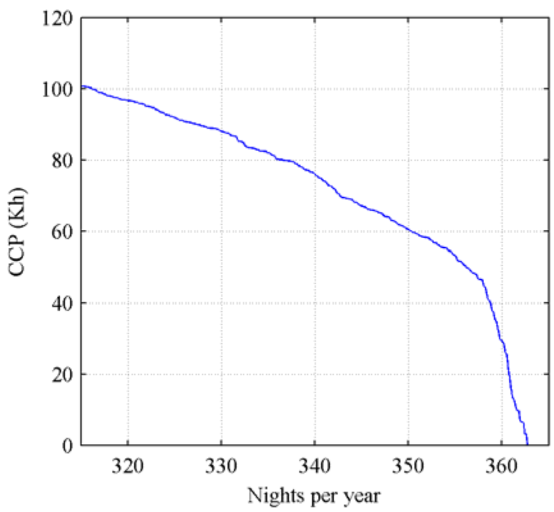


Figure 26. *Climatic cooling potential, CCP for the period 1996-2005 at Zurich SMA (based on ANETZ data).*

The *CCP* needed to discharge the daily heat gains is assessed as (cf. equation 11):

$$CCP = \frac{Q_d}{A_{Floor}} \cdot \frac{3600\,\mathrm{s/h}}{\eta \cdot ACR \cdot H \cdot \rho_{Air} \cdot c_{p,\,Air}} , \qquad (15)$$

with room height H = 2.6 m and $\rho_{Air} \cdot c_{p,\,Air} = 1142\ \mathrm{J/kgK}$.

5. Night ventilation air change rate

Depending on the air change rate the number of days per year with insufficient *CCP* can be assessed from the cumulative frequency distribution. For the three levels of heat gains Figure 27 shows the risk for overheating depending on the air change rate. In all cases the risk for overheating is reduced rapidly with increasing air change rate to about 7 to 8 days per year. However, even at 30 ACH a risk of overheating on 4 to 5 days per year remains.

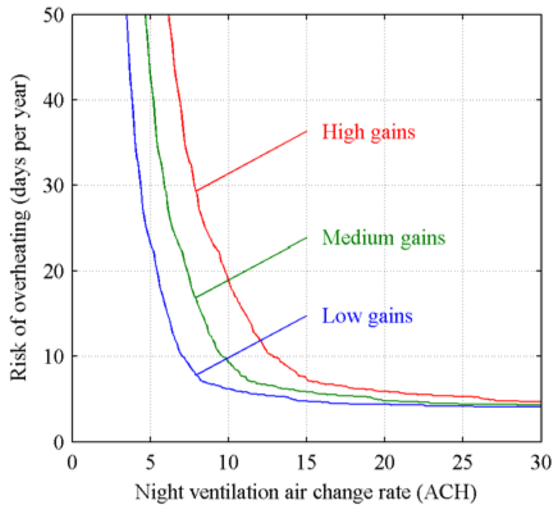


Figure 27. *Number of days per year with a risk of overheating depending on the air change rate for low, medium and high heat gains.*

For the three levels of heat gains, the air change rates needed to limit the risk of overheating to 7 days per year are:

- Low gains: *ACR* = 9 ACH
- Medium gains: *ACR* = 12 ACH
- High gains: *ACR* = 16 ACH

6.3. Comparison with simulation results

In order to verify the design method, the different case examples were simulated with the suggested air change rates (low gains: 9 ACH, medium gains: 12 ACH, high gains: 16 ACH). Table 6 gives the number of days when the operative room temperature exceeds 27 °C which was the maximum of the building temperature in the definition of *CCP*. In the recommended cases the operative room temperature exceeds 27 °C on 3.8 (low gains/light-weight), 2.7 (medium gains/medium weight) and 0.7 (high gains/heavy-weight) days per year, while the design method predicted a risk of overheating on 7 days per year in all cases.

Table 6. *Number of overheating days with operative room temperature exceeding 27 °C (days per year) based on simulations.*

	Low gains (224 Wh/m^2) 9 ACH	Medium gains (294 Wh/m^2) 12 ACH	High gains (378 Wh/m^2) 16 ACH
Light-weight (131 kJ/m^{2}K)	3.8 d/y*	8.6 d/y	22.0 d/y
Medium-weight (180 kJ/m^{2}K)	1.5 d/y	2.7 d/y*	5.8 d/y
Heavy-weight (287 kJ/m^{2}K)	0.5 d/y	0.7 d/y	0.7 d/y*

* Cases recommended based on the design method

This comparison suggests that the design method overestimates the risk for overheating. There are various possible reasons to explain this:

- As discussed in [52], in some cases the assumption of a harmonically oscillating building temperature results in an underestimation of *CCP*. Due to its definition the building temperature starts to decrease at 7 pm even if the external temperature is not low enough for effective night-time ventilation. This results in *CCP* = 0, although there might be a cooling potential during the late night hours if the building temperature would remain at a constant level until ventilation starts (cf. Figure 3).
- In the simulations ventilation with 2 ACH was considered during the day to ensure indoor air quality. In the mild climate of Zurich the external temperature is mostly below the indoor temperature, and therefore the heat gains are partially discharged by ventilation during the day. To account for this effect it could be assumed, that only a certain fraction of the heat gains needs to be discharged by night-time ventilation. However, the discharge of heat during the day highly depends on climatic conditions. To be on the safe side, it is recommended to design night-time ventilation for the total daily heat gains. Additionally in the simulation no heat gains were assumed during weekends.
- In the example the temperature efficiency measured in the experiments during mixing ventilation was applied, while in the simulations the coefficient for combined heat transfer was assumed as 7.7 W/m^2K. As discussed in section 4.2 a combined heat transfer model might overestimate radiation during night-time ventilation.
- In the simulation night-time ventilation was terminated when the mean surface temperature fell below 20 °C, i.e. the maximum daily temperature variation without exceeding 27 °C was 7 K. In the suggested case examples the temperature variation was in the range from 4.7 to 6.2 K. Comparing the case examples shows that the overestimation of overheating by the design method is highest in cases with a small daily temperature variation. This means that increasing the thermal mass does not only decrease the daily temperature variation but also gives additional safety in the ventilation rate.
- In case of oversized thermal mass (daily temperature variation smaller than the acceptable thermal comfort range), insufficient cooling potential in one single night will not cause overheating on the next day. Only after several consecutive nights of insufficient CCP the temperature of the thermal mass will gradually increase and finally the room temperature will exceed the limit for thermal comfort. In the design method based on daily CCP this dynamic effect can not be accounted for. Therefore, in the detailed design phase a dynamic building simulation should be performed.

6.4. Concluding remarks

Although there are some limitations, the example showed that the proposed design method gives valuable results for the estimation of the thermal capacity and the airflow rate needed to ensure thermal comfort by night-time cooling. This method is recommended for a first estimation during the concept design phase and is not meant to replace a detailed analysis by means of dynamic building energy simulations.

The model most commonly used in building energy simulation assumes a homogeneous room air temperature and the outflowing air temperature being equal to the room air temperature. During displacement ventilation, due to the temperature stratification, some of the internal room surfaces (e.g. the floor) can be colder than the outflowing air. As the temperature of this surface, $T_{Surface,i}$ is still higher than the air close to it, the surface releases heat to the air ($\dot{q}_{conv,i} > 0$). For a one-air-node model this results in a negative heat transfer coefficient:

$$h_i = \frac{\dot{q}_{conv,i}}{T_{Surface,i} - T_{Room\ air}} = \frac{\dot{q}_{conv,i}}{T_{Surface,i} - T_{Outlet}} < 0 \qquad (16)$$

An alternative approach could be to use the temperature efficiency, η to calculate the total heat flow discharged from a room in a dynamic model. In this model, the efficiency represents the heat transfer at all room surfaces (including radiation), which supersedes the definition of heat transfer coefficients for single surfaces. However, such a model does not give the heat flows from different room surfaces separately. Therefore the entire thermal mass of the room needs to be modelled as one element with a homogeneous surface temperature. A possible approach for representing the thermal mass of a room (or building) using a virtual sphere model has been proposed by Li and Yam [75]. The applicability of the temperature efficiency and the virtual sphere model in a dynamic building energy simulation, however, needs further investigation.

7. Conclusions

For quantifying the climatic potential for night-time cooling a degree-hours method was developed, evaluating the utilisable difference between the building and the ambient temperature. The method for calculating the climatic cooling potential (CCP) is basically suitable for all building types, regardless of building-specific parameters. This was achieved by basing the approach solely on a building temperature variable within a temperature band given by summertime thermal comfort.

Applying this method to European climate data displayed an obvious gradient in CCP decreasing from North to South. Under current climatic conditions, the potential for passive cooling by night-time ventilation was found to be very significant in Northern Europe (including the British Isles), and still sufficient in most nights of the year in Central and Eastern Europe. However, a risk of thermal discomfort on a few days per year exists, depending on building design and heat gains. In Southern Europe passive night-time cooling might not be sufficient the whole year round, but can still be used in hybrid cooling systems.

A detailed analysis of data from future climate projections until the end of the 21st century showed that night-time cooling potential will cease to be sufficient to assure thermal comfort in many Southern and Central European buildings. In Northern Europe, a significant cooling potential is likely to remain, at least for the next few decades. At the same time the cooling demand increases which results in a shift of possible applications from South to North and from summer to spring and autumn. Due to uncertainties in climate projections, upper and lower bound estimates for future CCP were found to diverge strongly in the course of the 21st century. Therefore, the need for adaptive building design taking into account a wide range of possible climatic boundary conditions becomes evident.

The high sensitivity of night cooling to climatic conditions was confirmed in a parameter study based on building energy simulation (*HELIOS*). Simulations based on commonly-used semi-synthetic one year data sets such as *Design Reference Year* (DRY) or *Meteonorm* [55] data were found to underestimate the extent of overheating compared to measured weather data. Therefore, for reliable summer thermal comfort predictions the application of long term measured climate data including extreme periods is recommended.

The building energy simulation study also revealed a high sensitivity to the surface heat transfer, if heat transfer coefficients were below about 4 W/m^{2}K. Convective heat transfer coefficients at internal room surfaces however vary in a wide range from 0.7 W/m^{2}K given for downward heat flow at horizontal surfaces [39], to values above 20 W/m^{2}K in cases with high air flow velocities. Applying the standard definition of heat transfer coefficients – based on the difference between the surface temperature and the average room air temperature – the temperature stratification in case of displacement ventilation can even result in negative heat transfer coefficients at individual surfaces.

A practicable method to evaluate the potential for night cooling was proposed. In order to assure thermal comfort two criteria need to be satisfied, i.e. (i) the thermal capacity of the building needs to be sufficient to accumulate the daily heat gains within an acceptable temperature variation and (ii) the climatic cooling potential and the effective air flow rate need to be sufficient to discharge the stored heat during the night. The amount of thermal mass needed to accumulate daily heat gains can be assessed based on the dynamic heat capacity of the building elements. Using a design chart, the minimum air flow rate necessary to discharge the stored heat can be estimated based on the climatic cooling potential and the temperature efficiency. This approach might mainly be helpful during the concept design phase of a building. For a more detailed analysis of the summertime transient thermal behaviour, the temperature efficiency can also be applied to determine the total heat flow in a dynamic simulation.

Experimental results showed, that for displacement ventilation at low air flow rates, the temperature stratification and the high location of the outlet opening results in a very high temperature efficiency ($\eta > 1$). In mixing ventilation warm air is supposed to be removed from the ceiling by the inflowing air jet. However, only at a relatively high air change rate (above about 10 ACH) the effect of the air jet flowing along the ceiling becomes significant and mixing ventilation is more efficient. Therefore, if a low air flow rate is expected, the outlet opening should be placed as close to the ceiling as possible.

Considering higher comfort expectations, increasing heat gains and the gradually warming climate, a rapid increase in cooling energy demand is expected if mechanical air conditioning systems are applied. As peak cooling loads occur coincidentally over large areas, a very high peak load capacity would be needed to prevent extensive breakdowns of electric power supply. Additionally, substantial greenhouse gas emissions are related to the increased energy demand, while actually a significant reduction of emissions will be necessary to restrain climate warming to an acceptable limit. This demonstrates the great need for the application of passive and low energy cooling methods. Overall, this PhD study showed that night-time ventilation holds a high potential to reduce cooling energy demand in Central, Eastern and Northern Europe. It is hoped, that the findings of this study will contribute to a more frequent application of passive cooling methods.

8. Recommendations for future work

The method proposed for assessing the heat flow discharged during night-time ventilation is based on the temperature efficiency of the ventilation. In the experiments conducted, the temperature efficiency during mixing and displacement ventilation was determined for a test room with a heavy ceiling and light-weight walls and floor. In order to make this method more generally applicable, further work is needed to determine the temperature efficiency for various room geometries, constructions (amount and location of thermal mass) and ventilation systems (location of inlet and outlet openings).

Additionally the presence of furniture might have a significant impact on the air flow, the heat transfer, and the resulting temperature efficiency. On the one hand, furniture might constrain the heat transfer at the floor and wall surfaces. On the other hand, the large surface area of furniture might increase the total convective heat flow. Even if furniture does not contribute much to the total heat storage capacity of a room, the temperature of its surfaces drops quickly and heat is transferred from heavy-weight building elements by radiation. Furthermore the presence of furniture might completely change the air flow pattern in a room.

A possible approach to investigate a large number of different cases could be the application of computational fluid dynamics (CFD). As the time constant of building elements is typically much longer than the time constant of the room air, a steady state CFD model is believed to be sufficient to determine the temperature efficiency. To induce a temperature gradient between the thermal mass and the supply air a heat source is needed in a steady state model. In order to achieve a realistic surface temperature distribution, a heat source (constant temperature or constant heat flow density) at the external surface of heavy-weight building elements is suggested. Initial CFD simulations using such a model appeared to be promising.

In case of displacement ventilation the temperature stratification can result in negative heat transfer coefficients according to the standard definition based on the difference between the surface temperature and the average room air temperature. In order to avoid a physically meaningless definition of negative heat transfer coefficients, the temperature efficiency of the ventilation could possibly be used to model the total heat flow discharged during night-time ventilation. Furthermore, the temperature efficiency is based on the difference between the average internal surface temperature and the inflowing air temperature, and thus the assumption of a homogeneous room air temperature can be avoided. The applicability of this approach could be investigated by comparing simulation results from different models to experimental data.

Nomenclature

A	Area	(m^2)
ACR	Air change rate	(h^{-1}), (ACH)
Ar'	'Dimensional' Archimedes number	(Ks^2/m^6)
c, c_p	Specific heat capacity	(J/kgK)
C_{dyn}	Dynamic heat storage capacity	(J/K)
c_{dyn}	Dynamic heat storage capacity per unit area	(J/m^2K)
CCP	Climatic cooling potential	(Kh)
d	Thickness	(m)
g	Total solar energy transmittance	(-)
H	Height	(m)
h	Heat transfer coefficient	(W/m^2K)
I	Solar irradiation	(Wh/m^2)
$\dot{m}$	Mass flow rate	(kg/s)
ODH_{26}	Overheating degree hours above 26 °C	(Kh)
Q	Heat	(J), (Wh)
$\dot{Q}$	Heat flow	(W)
$\dot{q}$	Heat flux	(W/m^2)
T	Temperature	(°C)
ΔT	Temperature difference	(K)
ΔT_0	Initial temperature difference	(K)
t	time	(s)
$\dot{V}$	Volume flow rate	(m^3/s)

Greek symbols

γ	Convection ratio	(-)
η	Temperature efficiency	(-)
θ	Dimensionless Temperature	(-)
λ	Thermal conductivity	(W/mK)
ρ	Density	(kg/m^3)

Subscripts

b	Building	*f*	Final
cond	Conduction	*i*	Initial
conv	Convection	*min*	Minimum
crit	*Critical*	*max*	Maximum
d	Day	*rad*	Radiation
e	External	*tot*	Total

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