Ventilative cooling:

need, potential, challenges, strategies

A selection of papers
from the Proceedings of the 33rd AIVC- 2nd TightVent Conference:
Optimising Ventilative Cooling & Airtighness for [Nearly] Zero-Energy Buildings, IAQ & Comfort

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the international platform for ventilative cooling

INIVE

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Foreword

In general, but in particular in nearly zero-energy buildings, there is a very strong tendency to drastically reduce the heating demand. One adverse side effect is that in doing so, it often increases the risk of overheating in summer and shoulder seasons. This is in particular, but not only, the case for lightweight constructions.

Experience shows that active cooling is too often considered to begin with, while other options should be prioritized in the building design when relevant. In fact, proper building design strategies including adequate solar control, thermal mass, ventilative cooling—i.e., use of ventilation to cool indoor spaces—can overcome this risk of overheating with no or minimum use of active cooling.

Because there is a growing interest in ventilative cooling to reduce the cooling energy demand and improve thermal comfort in summer and shoulder seasons, one purpose of the '33rd AIVC Conference' was to discuss the potential, challenges and perspectives of this technique in a track of specific sessions.

This conference also inaugurated venticool, the international platform on ventilative cooling, whose aim is to accelerate the uptake of ventilative cooling by raising awareness, sharing experience and steering research and development efforts in the field of ventilative cooling.

Because venticool partners realize the valuable experience and knowledge shared during this conference which should be of interest to many professionals, policy makers, and researchers beyond the conference attendance, venticool has produced this book gathering 16 publications presented during the conference. On behalf of venticool partners, I wish you a pleasant and informative reading.

Peter Wouters

Manager of INIVE EEIG

What is venticool?

venticool is the international ventilative cooling platform launched in October 2012 to accelerate the uptake of ventilative cooling by raising awareness, sharing experience and steering research and development efforts in the field of ventilative cooling. The platform supports better guidance for the appropriate implementation of ventilative cooling strategies as well as adequate credit for such strategies in building regulations. The platform philosophy is pull resources together and to avoid duplicating efforts to maximize the impact of existing and new initiatives. venticool will join forces with organizations with significant experience and/or well identified in the field of ventilation and thermal comfort like AIVC (www.aivc.org) and REHVA (www.rehva.eu).

The platform was officially launched during the 33rd AIVC-2nd TightVent conference in Copenhagen, Oct. 10-11 2012 hosting 4 topical sessions on ventilative cooling. It has been initiated by INIVE EEIG with (International Network for Information on Ventilation and Energy Performance) is at present supported by the following organizations:

- AGORIA naventa (www.agoria.be/WWW.wsc/webextra/prg/izContentWeb?ENewsID=89198&vApplication=Naventa)
- ES-SO (www.es-so.com)
- Eurima (www.eurima.org)
- INIVE (www.inive.org)
- VELUX (www.velux.com)
- Window Master (www.windowmaster.com)

IEA ECBCS Annex 62 on ventilative cooling

The Executive Committee of the International Energy Agency Energy in Buildings and Communities Programme (IEA ECB) accepted the formation of a new IEA EBC Annex on Ventilative Cooling at their last meeting in November 2012. This new Annex 62 is given a one year preparation phase which, if successful, will continue in a four year working and reporting phase from 2014 – 2017. During the preparation phase two workshops will be arranged to define and focus the Annex's objectives, feasibility, methodology, and deliverables in detail. The 1st Annex 62 Preparation Meeting was held March 21 – 22 in the BBRI offices, Brussels, Belgium and the 2nd Preparation meeting in Athens, September 23-24, 2013.

In order to address the cooling challenges of buildings the research focus of the annex will be on development of design methods and compliance tools related to predicting, evaluating and eliminating the cooling need and the risk of overheating in buildings and to develop new attractive energy efficient ventilative cooling solutions.

Annex 62 will be divided in three subtasks. Subtask A "Methods and Tools" will analyse, develop and evaluate suitable design methods and tools for prediction of cooling need, ventilative cooling performance and risk of overheating in buildings. The subtask will also give guidelines for integration of ventilative cooling in energy performance calculation methods and regulation including specification and verification of key performance indicators. Subtask B "Solutions" will investigate the cooling performance of existing mechanical, natural and hybrid ventilation systems and technologies and typical comfort control solutions as a starting point for extending the boundaries for their use. Based upon these investigations the subtask will also develop recommendations for new kinds of flexible and reliable ventilative cooling solutions that can create comfort under a wide range of climatic conditions. Subtask C "Case studies" will demonstrate the performance of ventilative cooling through analysis and evaluation of well-documented case studies.

EBC Annex 62 will include the participation of approximately 15 countries from Europe, Japan and the US, from universities, research centers and manufacturers and suppliers of ventilation equipment. At the first preparation meeting the focus, research objectives and research methodology was determined. The second and final preparation meeting focused on the development of a detailed work plan for the research to be carried out on Ventilative Cooling from 2014-2017.

The operating agent is Professor Per Heiselberg (e-mail ph@civil.aau.dk), from Aalborg University, Denmark.

Cooperation between venticool and IEA Annex 62

venticool provides technical and logistical support to IEA Annex 62, in particular for:

- The organization of events (workshop, webinars, etc.);
- Communication and outreach activities foreseen within the annex work;
- Input on standards and regulations issues.

This collaboration is expected to be mutually beneficial by taking advantage of the complementary expertise and knowledge of the persons and organizations involved.

Table of Contents

Summary of the Ventilative Cooling track at the AIVC Conference October 2012 in Copenhagen

Per Heiselberg

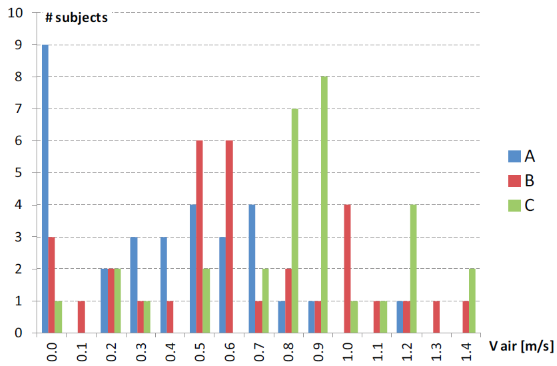
Aalborg University, Denmark

The current development in building energy efficiency towards nearly-zero energy buildings represents a number of new challenges to design and construction of buildings. One of the major new challenges is the increased need for cooling present in these highly insulated and airtight buildings, which is not only present in the summer period but also in the shoulder seasons and in offices even during occupied hours in winter. In most post-occupancy studies of high performance buildings in European countries elevated temperature levels is the most reported problem, especially in residences.

These new challenges were strongly reflected in the programme of the 33rd AIVC conference where about 30 papers and presentations in 6 sessions dealt with different issues related to ventilative cooling.

Performance Criteria

The first question to ask in the design of ventilative cooling systems is what are the performance criteria? What is considered as overheating and how can ventilative cooling be a solution? Research on different strategies to offset of warm sensation in high temperature conditions showed that increased air velocities can compensate and ensure comfortable conditions at higher temperature levels and that air fluctuations and turbulence intensity play an important role. But, the research results also showed significant individual differences in the preferred air velocity, which indicate that personal control is very important.



Condition	t_o [°C]	RH %
A	26	50
B	28	45
C	30	40

Figure 1: Number of subjects choosing a certain velocity [1]

Prediction Methods

Prediction of energy use in residential buildings is often based on simplified monthly methods and is estimated for the residence as a whole. Averaging the need for cooling in both time and space underestimates the need for cooling. Excess heat in spaces exposed to solar radiation is considered to be distributed fully to other spaces and excess solar radiation during daytime is partly distributed to night time. Therefore, the need for cooling to ensure acceptable temperature levels in all spaces will be higher in reality. The analysis of the risk of overheating is often based on the calculated cooling need. Unfortunately, there is no correlation between the calculated cooling need with these simplified methods and the number of hours with elevated temperature levels. So, even if no cooling need is predicted and designers do not expect overheating problems, the number of hours with elevated temperature levels can be considerable. Several presentations dealt with these issues both from a more theoretical approach analyzing the energy balance and heat transfer processes within spaces in buildings but also different methods to predict Ventilative Cooling performance and the risk of overheating was presented.

Case Studies

A number of case studies on the application of ventilative cooling were presented. The case studies demonstrated ventilative cooling solutions both in residences, schools and shopping malls and in many situations considerable energy savings was obtained.

Several of the case studies highlighted the need for development of new technical solutions. One example was in the new Nicosia Townhall, where it was concluded that free cooling by night ventilation was the simplest strategy to keep comfortable temperatures, but using standard openings was not the best solution. In order to ensure better opening possibilities as well as protection from insects, dust and vandalism special façade vents were developed.

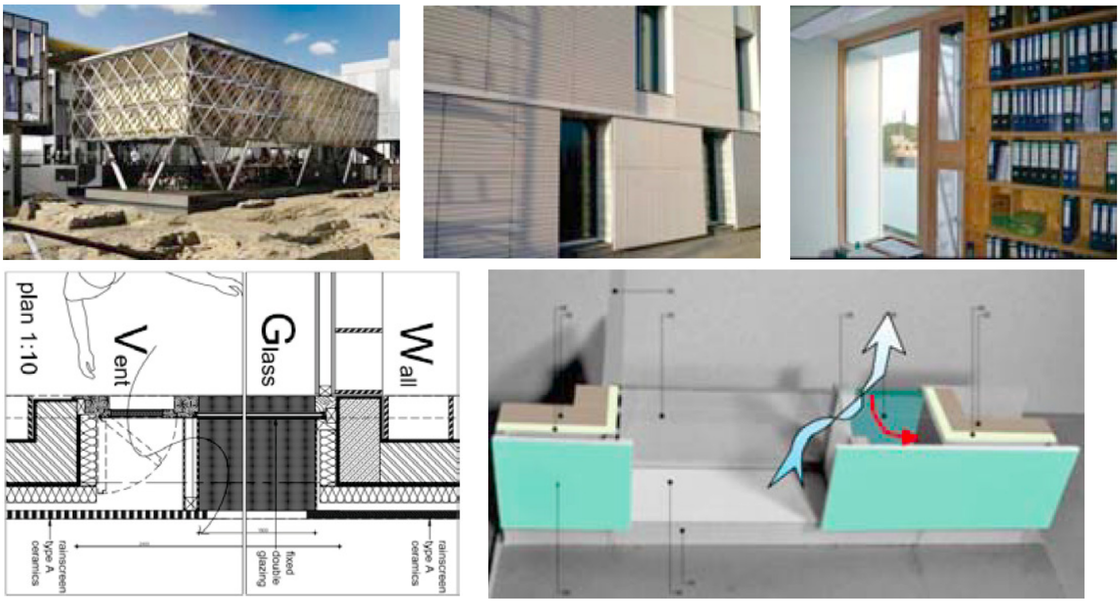


Figure 2: The Nicosia Townhall where special façade vents were developed to realise ventilative cooling in the building [2]

Conclusion

Ventilative cooling can be an attractive and energy efficient solution to avoid overheating of both new and renovated buildings. Ventilation is already present in most buildings through mechanical and/or natural systems and it can both remove excess heat gains as well as increase air velocities and thereby widen the thermal comfort range. As cooling becomes a need also outside the summer period the possibilities of utilizing the cooling potential of low temperature outdoor air increases considerably. The outcome of the Ventilative Cooling track was that especially for residences there is a need to improve both the way we estimate the need and the performance of ventilative cooling as well as for development of new, off-the- shelf and competitive technical solutions.

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Long term monitoring of residential heat recovery ventilation with ground heat exchange

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Abstract

The monitoring of a demand controlled heat recovery ventilation system with ground heat exchange in a zero-energy building in Groenlo, The Netherlands, revealed interesting practical insights.

A healthy indoor climate can be obtained with a high comfort in terms of CO_2 levels and supply air temperatures. The CO_2 level stays well within the comfortable range in the living room and three bed rooms (parents, child, and guests), thanks to the demand controlled ventilation. Supply air temperatures are in the comfortable range thanks to the heat recovery in combination with the ground heat exchange by an earth pipe. This is shown for ambient temperatures between -8°C and +33°C.

The energy efficient behaviour is proven by the avoided heating load of 3465 kWh and free cooling of 1052 kWh during a full year. The observed seasonal performance factor SPF is 17 for the avoided heating and 8 for the free cooling. The thermal efficiency based on the supply air temperature is observed to be as high as 91% for a slightly unbalanced air flow. When this is mathematically corrected, the thermal efficiency would have been 97% for a perfect balance between supply and return air flow.

Keywords

Indoor air quality (IAQ), residential ventilation, heat recovery, ground heat exchange, passive cooling, monitoring, CO_2, recovery efficiency, demand control

Introduction

This article reports the results of a full year monitoring of a zero-energy residential building in Groenlo, the Netherlands. The ventilation system in this building is a demand controlled heat recovery ventilation system in combination with ground heat exchange in the form of an earth pipe. The results of the monitoring show that the ventilation system is highly energy efficient and provides a healthy and comfortable indoor climate.

The building

The monitored building displayed in Figure 1 has been built according to the passive house standards. In general terms, the house has a compact, well insulated envelope and south oriented windows with triple glazing. Photovoltaic panels and solar thermal collectors on the roof provide electricity and hot tap water during sunny weather. A heat pump coupled to a vertical bore hole is providing heating and cooling via a floor distribution system. Details of the house can be found in [1] and [2].

Figure 1: The monitored building in Groenlo, The Netherlands

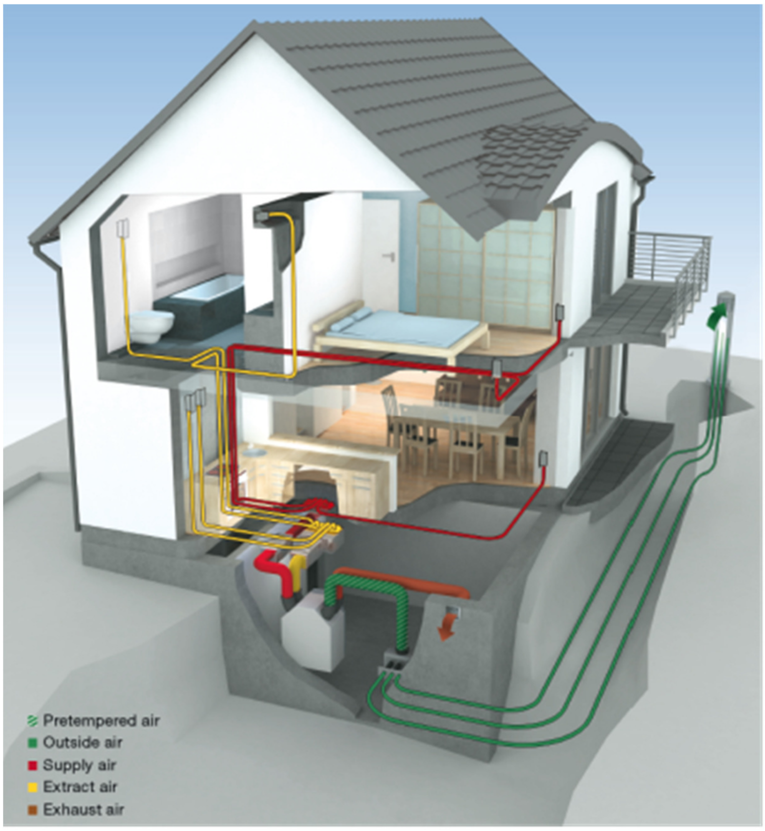

Figure 2: Representation of the ventilation system

The ventilation system

A heat recovery unit is bringing fresh outdoor air into the building and is removing stale air from the building (Figure 2). The heat from the extract air is recovered and returned back into the fresh air for energy efficient ventilation. The standard ventilation volume is 160 m^3/h. In a house with a volume of 840 m^3 this corresponds to a ventilation rate of around 0.2 h^{-1}.

The outdoor air is supplied to the individual rooms by 7 individual flexible circular ducts. Four of them lead to low induction grilles near the floor of the bedrooms (parents, child and guests) and an office room, all situated on the ground floor. The rest leads to the first floor to the living room. Extract air is extracted from the living room, the loft, the bathroom and the toilets via 7 return ducts. Supply air as well as return air are distributed and collected respectively via sound attenuators, one in the extract air stream and two in the supply air stream. The kitchen is ventilated by a separate HRU which is not subject of the monitoring project.

The manual setting of the ventilation volume (standard position 1; 160 m^3/h) is increased automatically by a demand control based on 4 individual CO_2 sensors in the living room and the bedrooms (parents, child and guests). If one of the CO_2 levels is above a pre-set threshold level, a signal is brought to the HRU to increase the air volume.

Ground heat is provided by an earth pipe. The earth pipe is 50 m long with a diameter of 200 mm and a mean depth of 2.5 m. It is buried into the earth at a slope to remove any possible condensation in the pipe. An air damper is installed in the fresh air duct upstream of the HRU. The HRU controls the damper to decide whether outdoor air is brought into the building directly from outside (north façade) or via the earth pipe (inlet see foreground in Figure 1). Note: Figure 2 shows a slightly different version with 3 parallel and shorter earth pipes without a damper installed.

The monitoring

The relevant parameters of the ventilation system have been collected at an interval of 1 minute by a laptop connected to the HRU. The collected data is sent weekly by the resident accompanied by any relevant feedback. The data is transformed into hourly values and analysed in the form of so-called carpet plots, duration graphs, correlation diagrams or bar charts. This report gives the results of a full year starting in February 2011 until February 2012. An intermediate report for a half year period can be found in [3].

Comfortable CO_2 levels

The comfort in the house is assessed by the CO_2 levels in the living room, the master bedroom (2 parents), the child's bedroom and the guest bedroom. The threshold level for the living room was set at 800 ppm and for the bedrooms at 1000 ppm.

As expected, the hourly CO_2 values showed an increased CO_2 level when the rooms were occupied. As an example, Figure 3 shows the CO_2 levels in the child's bedroom for the period February to May 2011. During the day, CO_2 levels are close to the natural background level of 400 ppm while during the night they were in the range 800 – 1000 ppm. When the CO_2 level exceeded the threshold level of 1000 ppm, the ventilation was increased automatically by the HRU to maintain the CO_2 level within a healthy and comfortable range.

From mid-April on, there is a generally lower CO_2 level in all of the rooms resulting from window ventilation used in this period of higher solar irradiation on the south façade.

Another observation is that the CO_2 level during the night was generally higher in the child's bedroom (occupied by one child) than in the master bedroom (occupied by two adults). The reason for this is that the child sleeps with the door closed and the parents sleep with the door open. An open bedroom door results in an exchange of the air in the bedroom (with CO_2 source) with the air in the hallway (without CO_2 source). This pattern was confirmed by CO_2 levels above normal when the master bedroom door was closed occasionally.

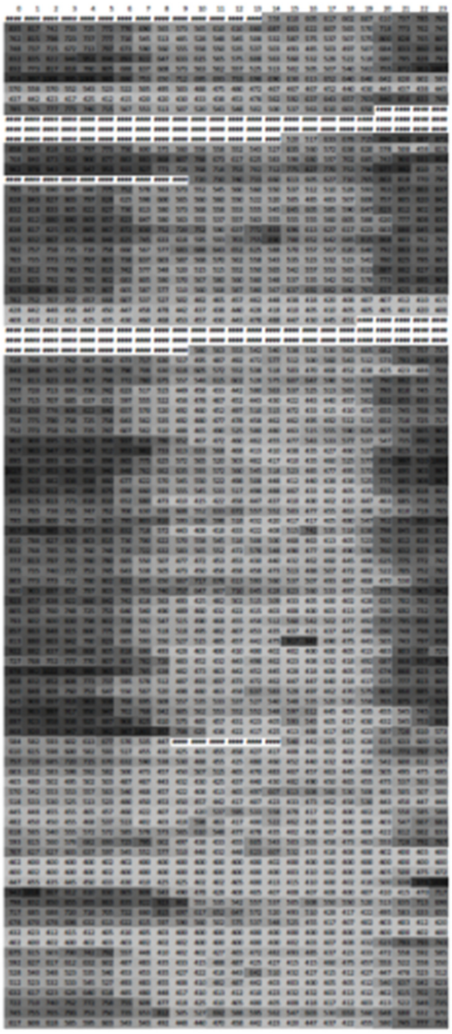

Figure 3: Carpet plot of CO_2 level in the child's bedroom. Rows indicate days from Feb 5th (top row) to May 16th (bottom row). Columns indicate hour of day from 0:00 (left column) to 24:00 (right column). Grey shades indicate CO_2 level ranging from 400 ppm (grey) to 1200 ppm (black), and missing data (white). Note: black represents a CO_2 value that is still below the Dutch guideline values!

The observation of lower CO_2 levels with the door open is confirmed by theoretical calculations. The natural exchange of air by temperature differences between bedroom and hallway can be calculated as 370 m^3/h for a door of 1 m wide and 2 m high with a temperature difference of 1 °C! This is roughly 6 times more than the amount of fresh air of 58 m^3/h provided by the HRU on the maximal level. This means that an open door leads to 6 times faster dilution of CO_2 when compared to a closed door. Note that in case of an open door the CO_2 loaded air is replaced by air from the hallway with unknown air quality, while the HRU ensures the necessary amount of fresh air from outside.

A duration graph of CO_2 levels is given in Figure 4. The uncomfortable level of 1200 ppm is exceeded extremely rare (densely occupied bedroom with closed door). Again, one can see higher CO_2 levels in the child's room than the master bed room during the nights. The CO_2 level of 1200 ppm has never been exceeded in the living room and the child's bedroom. In the master bedroom and the guest bedroom, 1200 ppm has been exceeded only 0.1 % (8 hrs) and 0.2% (16 hrs) of the time, respectively.

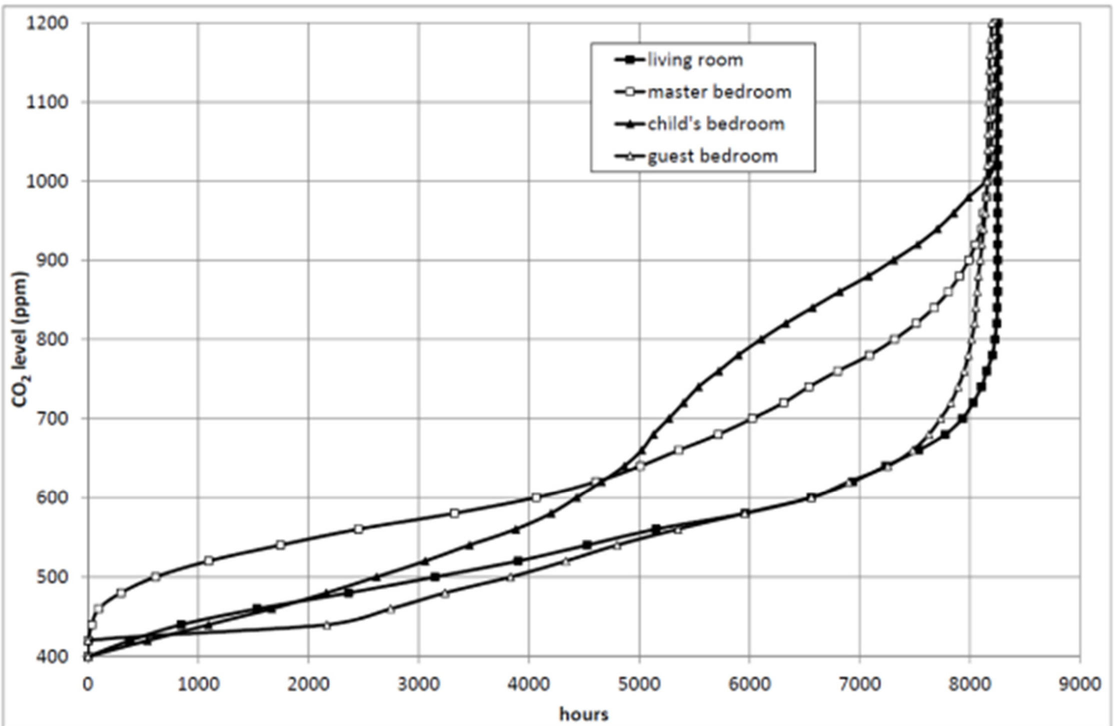

Figure 4: Duration graph of CO_2 levels in living room, master bedroom, child's bedroom and guest bedroom.

Comfortable temperatures

Throughout the year, the temperature of the earth at 2.5 m depth is much less varying than the outside air temperature. Therefore, the earth can be used for preheating incoming outdoor air in winter and precooling it in summer. In winter, the temperature of the earth is generally higher than the outside air temperature. Figure 5 shows that the preheated air (at the exit of the earth pipe) is between 8 and 12°C for outside temperatures between -5 and 10°C. In summer, the temperature of the earth is generally lower than the outside air. Figure 5 shows that the precooled air (at the exit of the earth pipe) is between 12 and 17°C for outside temperatures between 16 and 33°C. For mild outside temperatures between 10 and 16°C the ground heat exchange is switched off by controlling an air valve; outdoor air is taken into the house directly from the north façade (not via earth pipe).

The advantages of the ground heat exchange are the following. In winter, it ensures frost-free operation of the heat exchanger in the HRU, without the need for an electrical anti-freeze heating element. In summer, it decreases the temperature of the outdoor air to a level below the inside temperature, so that free cooling is used for the whole summer period, and not only during cool nights. The extra fan power to draw the air through the earth pipe is negligible (approximately 3 W).

In winter, the (preheated) outdoor air is entering the HRU where it is efficiently heated by the return air in the heat exchanger. Figure 6 shows that ventilation air is supplied to living room and bedrooms with a comfortable temperature of 18°C in winter even at very low outside temperatures[1]. Without heat recovery, ventilation air would enter the rooms with a temperature equal to outside which would result in uncomfortable draughts.

[1]some hours with supply temperature below 18°C are situations with the central heating switched off during absence

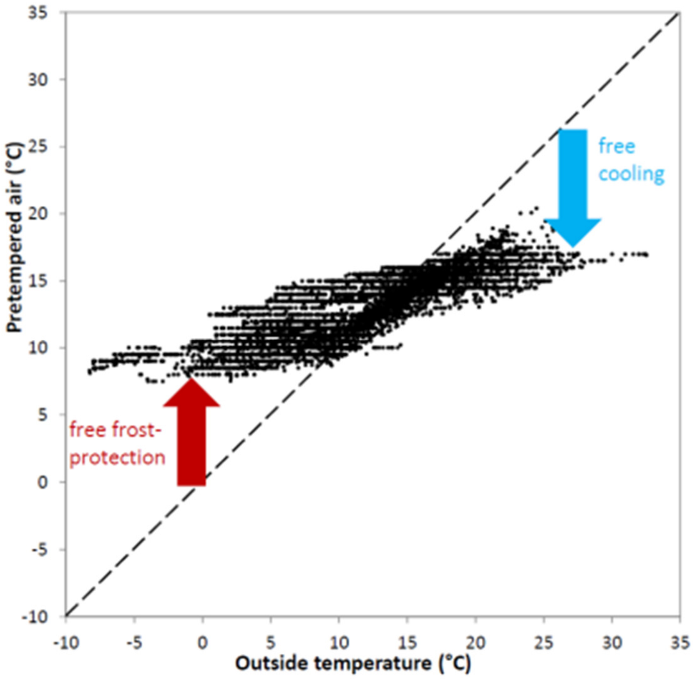

Figure 5: Hourly values of outdoor air entering the house. Ground heat exchange ensures frost protection of the HRU in winter and free cooling for the house in summer.

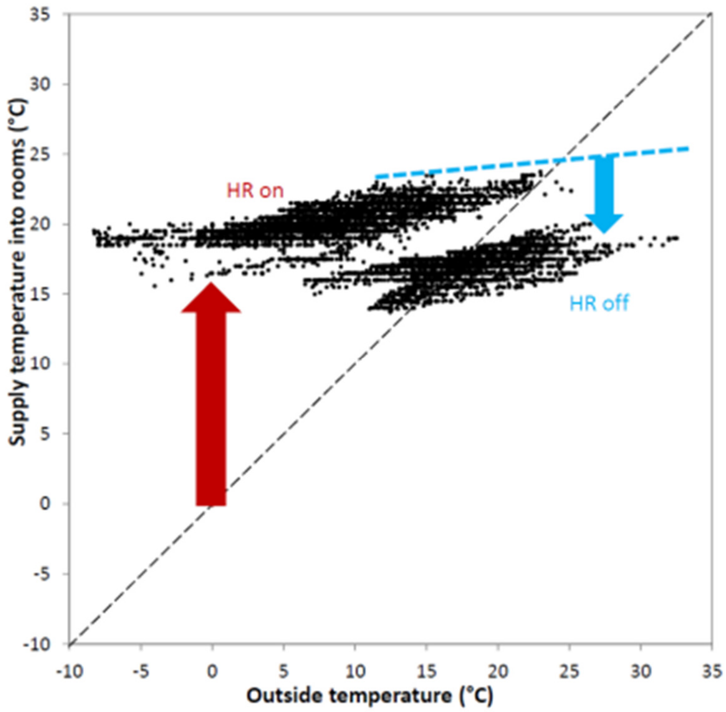

Figure 6: Hourly values of supply air temperature entering the rooms with and without heat recovery (HR on, and HR off). Upward arrow indicates avoided heating and downward arrow indicates free cooling compared to indoor temperature.

For outdoor temperatures above 13°C, the heat recovery is switched off when cooling is both requested and available. This occurs when both of the following conditions are true:

- Actual indoor temperature is above the setting of the comfort temperature (here: 21°C)
- Actual pretempered air temperature is lower than actual indoor temperature.

The heat recovery is switched off by bypassing the heat exchanger in the HRU. Outdoor air is transported directly (without heat recovery) to the rooms. This results in free cooling of the house as the supply temperature is always below the actual indoor temperature. The monitoring shows that the supply temperature is always below 20°C. As the ventilation air flow rate is not large (160 m^3/h), the free cooling cannot be compared with air conditioning equipment, but it raises the comfort and reduces the cooling load of the building.

Energy efficient ventilation

The benefits of heat recovery ventilation with ground heat exchange are expressed in terms of avoided heating and free cooling.

With heat recovery switched on, the avoided heating load (or recovered heat) reflects the fact that, thanks to the HRU, the central heating system does not have to heat cold outdoor air to the desired indoor temperature (see upward arrow in Figure 6). The exact amount can be calculated using the actual ventilation flow rate and the actual difference between supply air temperature and outdoor air temperature.

With heat recovery switched off, the free cooling reflects the fact that the indoor air is cooled by the incoming (lower) supply air temperature (see downward arrow in Figure 6). The exact amount can be calculated using the actual ventilation flow rate and the actual difference between indoor air temperature and supply air temperature.

Figure 7 shows cumulative avoided heating load and free cooling per week during the monitoring period. The energy benefits have been obtained at the expense of the electrical consumption of the fans in the HRU, which consume only 33 W at 160 m^3/h thanks to the low resistance of the flexible air distribution system.

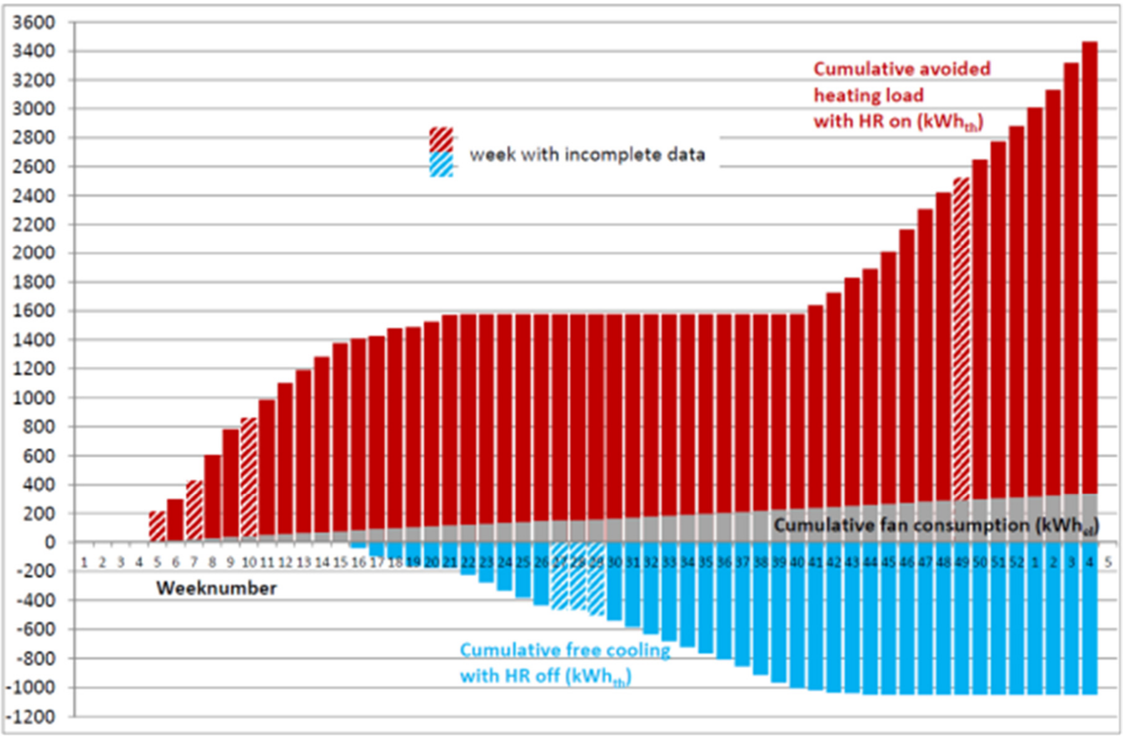

Figure 7: Cumulative sums for avoided heating load (above horizontal axis), free cooling load (below horizontal axis) and fan electricity consumption (above horizontal axis) from week 4 in 2011 to week 4 in 2012.

Table 1 shows a summary of values for the reported period February 2011 until February 2012. The seasonal performance factor SPF for avoided heating (or free cooling) is given by the ratio between avoided heating load (or free cooling load) and the electricity consumption of the fans during hours when the bypass was closed (or open). The observed SPF for avoided heating corresponds reasonably well with an SPF of 22 for the expected gain of a heat recovery system using comparable climate data of Milan, Italy from [4].

	Full year	Electrical consumption of fans during season	Seasonal Performance Factor SPF
Avoided heating load	3465 kWh	199 kWh	17
Free cooling load	1052 kWh	137 kWh	8

Table 1: Annual energy benefit of heat recovery ventilation and seasonal performance factors.

Thermal efficiency of heat recovery in practice

The thermal efficiency is defined as the ratio between outdoor air temperature increase and maximal temperature increase ($T_{\text{supply air}}-T_{\text{fresh air}})/(T_{\text{return air}}-T_{\text{fresh air}}$). When ground heat exchange is used, the outdoor air temperature in this formula is the preheated or precooled outdoor air temperature.

The thermal efficiency of an HRU is dependent on a lot of variables, among which ventilation flow rate and mass balance between supply air and extract air are dominant. Figure 8 shows practical efficiency as a function of fan percentage. The HRU is most frequently in position 1 (fan percentage 35%). Fan positions 2, 3 and absent can also be discerned. Intermediate fan percentages occur when the CO2 demand control increases fan percentage gradually.

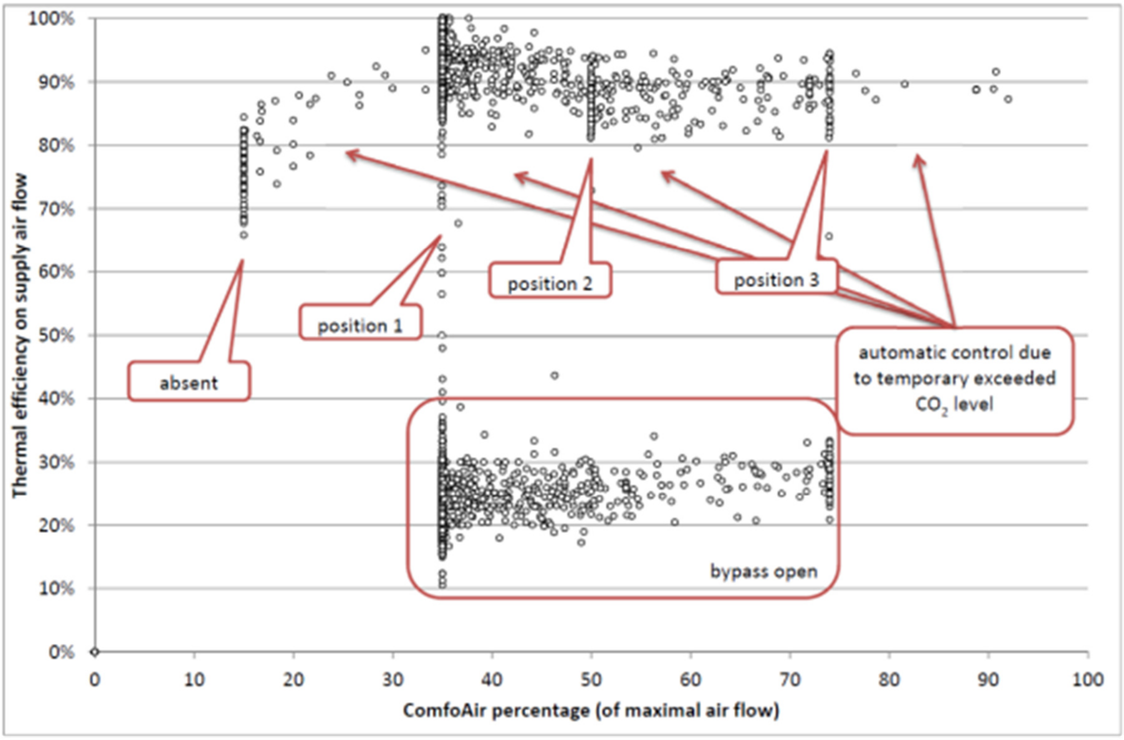

Figure 8: Thermal efficiency of HRU as a function of fan percentage (of maximal rotational speed).

With the bypass open (heat recovery switched off), the average undesired thermal efficiency is still 24%. Optimally, this efficiency would be 0%, but the fans add a small amount of heat (approximately 2°C) to the outdoor air, in spite of the use of efficient EC fans. If AC fans had been used, the thermal efficiency would be even higher.

With the bypass closed (heat recovery switched on), the optimal efficiency is obtained for the most frequently used fan position 1 (160 m^3/h). For the position 'absent' efficiency is decreasing, probably coming from imbalance in mass flows for very low flow regions. For higher fan speeds, the thermal efficiency is slowly decreasing because air is moving fast in the heat exchanger so that the limited exchanger surface becomes noticeable.

The observed average thermal efficiency with bypass closed is as high as 91%. This is a high number considering the fact that the supply air flow and the extract air flow are not perfectly balanced. The resident of the house has commissioned the HRU with a lower extract air flow than supply air flow rate. Detailed flow rate measurements revealed a 6% imbalance in volume flows. Mathematically, one can correct for this imbalance to obtain 91%/(100%-6%) = 97%. This means that, if the HRU system was commissioned in balanced flow, a thermal efficiency of 97% would be obtained, which corresponds perfectly with the thermal efficiency as measured in laboratory.

Conclusion

The monitoring of a demand controlled heat recovery ventilation system with ground heat exchange in a zero-energy building in Groenlo, The Netherlands, revealed interesting practical insights. A healthy indoor climate can be obtained with a high comfort in terms of CO_2 levels and supply air temperatures. The energy efficient behaviour is proven by the avoided heating load of 3465 kWh and free cooling of 1052 kWh during a full year. The observed seasonal performance factor SPF is 17 for the avoided heating and 8 for the free cooling.

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Human preference and acceptance of increased air velocity to offset warm sensation at increased room temperatures

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Abstract

Previous studies have demonstrated that in summertime increased air velocities can compensate for higher room temperatures to achieve comfortable conditions. In order to increase air movement, windows opening, ceiling or desk fans can be used at the expense of relatively low energy consumption.

The present climatic chamber study examined energy performance and achievable thermal comfort of traditional and bladeless desk fans. Different effects of mechanical and simulated-natural airflow patterns were also investigated. 32 Scandinavians, performing office activities and wearing light clothes , were exposed to an increased air movement generated by a personal desk fan. The subjects could continuously regulate the fans under three fixed environmental conditions (operative temperatures equal to 26 °C, 28 °C, or 30 °C, and same absolute humidity 12.2 Kg/m^3).

The experimental study showed that increased air velocity under personal control make the indoor environment acceptable at higher air temperatures. This will during summer season and in warmer countries improve thermal comfort without too high energy costs. There was significant individual difference in the preferred air velocities, which indicate that personal control is important. The accepted air velocities depended on the type and source of the increased velocity. The Scandinavian subjects did not accept so high velocities as found in studies with Chinese subjects.

Keywords

Thermal comfort, air velocity, personal control, desk fan

Introduction

Buildings' construction and operation are considered of central importance on the path of a sustainable development. Passive techniques for heating and cooling have gained more and more audience for their feasibility, their efficacy and the positive effects on human health when compared to traditional air-conditioned systems. Previous studies have broadly demonstrated that in summertime increased air velocities can compensate for higher room temperatures to achieve comfortable conditions [1-4]. In order to increase air movement, windows opening, ceiling or desk fans can be used at the expense of relatively low energy consumption[5-7].

Method

The present climatic chamber study examined energy performance and achievable thermal comfort of traditional and bladeless desk fans. Different effects of mechanical and simulated-natural airflow patterns were also investigated. 32 Scandinavians, performing office activities (1.2 met) and wearing light clothes (I_{cl} equal to 0.5-0.6 clo), were exposed to a direct air movement generated by a personal desk fan in continuous regulation under three fixed environmental conditions (operative temperatures (t_o) equal to 26 °C, 28 °C, or 30 °C, and relative humidity (*RH*) varying in the range of 40%-50% (at constant dew point of 14.8 °C)).

After an adaptation time, the subjects were invited to adjust the air movement for achieving their preferred thermal comfort. Is the preferred equal to the predicted thermal comfort? Will the personal control or different type of fans affect the results?

The individual preferred air velocities were recorded, and the relative energy consumptions were collected in order to estimate the potential energy savings when comparing to AC systems.

Experimental SET UP

The experiment was carried out in an office-like climatic chamber with dimensions 5.9*5.8*3.2 m^3 at the International Centre for Indoor Environment and Energy of Technical University of Denmark (ICIEE-DTU). The chamber reproduces a typical office room, providing occupants with a view on the outdoors garden. Internal and external blinds can be operated in order to let diffuse sunlight enter the room and meanwhile shade the direct sunlight. Eight workplaces, 4 on the right side and 4 on the left, were arranged with desk, office chair, desk lamp, and desk fan. A partition between the right and left side was located in the middle of the room in order to avoid any possible influence of air movement due to others occupants (see Figure 1).

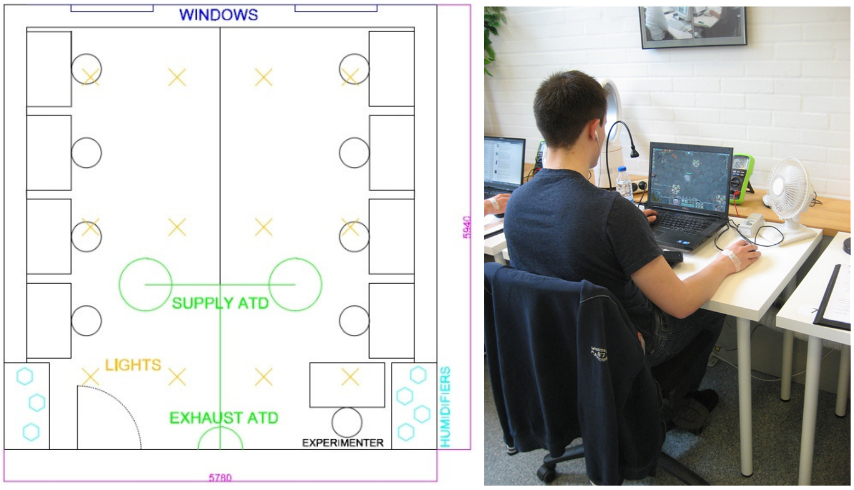

Figure 1: a) Sketch of the experimental chamber, b) view of workplace set up

The air and globe temperature sensors, developed at the ICIEE-DTU [8], have been calibrated for a temperature range of 20-35 °C with an accuracy of ±0.3°C and used during the experimental measurements. Omnidirectional anemometers were used for measuring air velocity. Temperatures and air velocity at 0.1, 0.6, 1.1 m and 1.7 m height above the floor level were recorded in the centre of the room. The skin temperatures of both hands and forehead of the subjects were continuously recorded. For the skin temperatures measures iButtons sensors were used as suggested by van Marken Lichtenbelt et al. [9] and Smith et al. [10]. During the experiment three different types of desk fans were used:

- 8 common two-steps desk fans which were regulated with a dimmer switch in order to obtain a continuous variation of the air speed (**CF** = Continuous regulation Fans)
- 2 bladeless fans (**BL**)
- 2 fans which simulate natural wind (**SN)**

The voltage applied to the CF fan has been correlated with the air speed generated at a distance of 60-70 cm from the fan, distance that intercor between the occupant and the fan, so that the air speed profile for each exposed participant was recovered by the recorded voltage data. The SN fans used on the present experiment were built at Tsinghua University, Beijing, China, where they have been used in a previous experiment with Chinese subjects [11].

Along the experiment the subjects were asked to fill in twelve surveys, 4 *long* and 8 *short* (containing respectively 19 and 10 questions) regarding: 1) thermal environment, namely thermal comfort, thermal acceptability, thermal preference, air movement preference, local thermal sensation and air movement sensation; 2) air quality, namely acceptability of air quality, perception of air humidity, preference on air humidity; 3) satisfaction with light and noise level; 4) experience of symptoms such as headache, dry eyes, irritated throat and nose irritation.

A total of 32 Scandinavian volunteers with good health participated in the experiments, most of them being university students. Their anthropometric data are reported in Table 1.

Sex	**No. of subjects**	**Age (years)**	**Height (cm)**	**Weight (kg)**	**Du Bois area (m^2)**	**Body Mass Index (BMI)**
females	16	23 ± 2	170 ± 6	66 ± 10	1.76 ± 0.14	22.9 ± 3.6
males	16	25 ± 4	180 ± 9	78 ± 22	1.97 ± 0.26	23.8 ± 5.0
females + males	32	24 ± 4	175 ± 9	72 ± 18	1.86 ± 0.23	23.3 ± 4.3

Table 1: Anthropometric data of the subjects

Each subject was exposed at three different conditions of 4-hours experiments in different days. The subjects were asked to wear a typical summer clothing ensemble, consisting in: panties/briefs, bra (if female), T-shirt, jeans or normal trousers, light socks, trainers or normal shoes. No garments that would protect the subjects from the air movement were allowed. The overall clothing insulation, considering the chair insulation of 0.1 clo (EN ISO 7730), resulted of about 0.5-0.6 clo.

Experimental procedure

The conditions investigated in the experiment are reported in Table 2.

Condition	**t_o [°C]**	**RH %**
A	26	50
B	28	45
C	30	40

Table 2: Physical parameters set during the experiment.

The relative humidity was set in order to keep the dew point constant at 14.8 °C. All experiments were carried out in afternoon sessions, in order to exclude confounding factors related to the circadian rhythms of the participants. Eight participants were exposed at the same time and at the same condition. They were allowed to work at their laptop, read a book or perform similar sedentary activities estimated equal to 1.2 met. The exposure time of each condition last in total 4 hours and only consumption of provided mineral water was allowed. Each condition consisted in 4 periods, as reported in table 2, with the adaptation time followed by three rounds of exposure to different air velocities or type of fans.

During the adaptation time (AD) the occupants were exposed for 90 minutes at the room environment having 0.5 m/s of air movement at the upper body part generated by the CF desk fan; it was established during the pre-test analyses. The three round text consisted on 30 minutes of exposure where the occupants were encouraged to freely adjust the air speed level of their desk fan (with fixed orientation) and 15 minutes of exposure to their preferred air velocity. At the end of each round the subjects were assigned to another desk and different type of fan. Each round consisted of same principle and exposure time.

CF fan was connected to a millimetre, which allowed to record the voltages correspondent to the preferred air velocities. The experiment was conceived to that in each session four subjects would be exposed to all three different types of fans, whereas the other four subjects would experience twice the exposure to the CF fan.

	Adaptation	**1st round**		**2nd round**		**3rd round**	
air movement	0.5 m/s	free adjust.	fixed setting	free adjust.	fixed setting	free adjust.	fixed setting
abbreviation	**AD**	**1 R**	**1 R***	**2 R**	**2 R***	**3 R**	**3R***
duration [min]	90´	30´	15´	30´	15´	30´	15´

Table 2: Time schedule of the experimental condition.

Results

Table 4 reports the usage of fans at the different environmental conditions (data from R1*, R2* and R3* are pooled). The three types of fans presented peculiar differences of usage. In condition A the CF fans were used by 71% of subjects, whereas only 26% and 35% used the BL and the SN fans, respectively. In condition C a vast majority of subjects kept the CF on. The usage of BL fans increases by 32% and 33% from condition A to B and from B to C, reaching 91% of usage in condition C. While the SN fans had a large increase of use from condition A to B (65%) and slightly decrease from B to C (4%).

% USAGE of FAN	CF		BL		SN	
condition	**% on (n)**	**% off (n)**	**% on (n)**	**% off (n)**	**% on (n)**	**% off (n)**
A	71 (22)	29 (9)	26 (6)	74 (17)	35 (8)	65 (15)
B	89 (29)	11 (3)	58 (14)	42 (10)	100 (18)	0 (0)
C	97 (30)	3 (1)	91 (21)	9 (2)	96 (22)	4 (1)
(n) represent the number of exposed participants						

Table 4: Usage of fans at the different environmental conditions

Resulted analyses when CF fans were used

The mean preferred air velocity (±standard deviation (SD)) of the three investigated conditions when CF fan was used are reported in Figure 2. It is increasing from condition A to B to C (from 0.56 m/s to 0.69 m/s to 0.85 m/s).

No statistically significant difference was found between male and female subjects.

The number of subjects choosing a certain air velocity is reported in Figure 3. In condition A nine subjects chose to keep the fan off, and 17 subjects chose an air speed between 0.3 and 0.7 m/s. In condition B the highest preferred air velocity was 0.5-0.6 m/s, while 0.9 m/s was chosen in condition C. A tendency towards higher air speeds at increasing air temperatures is clearly visible, large individual can still be observed.

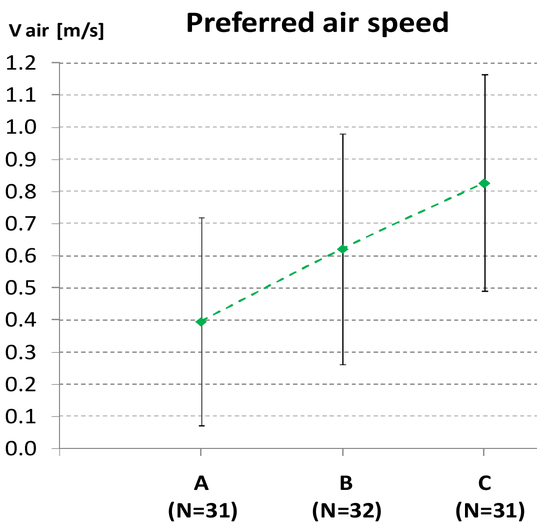

Figure 2: Preferred air velocity at the three investigated conditions.

The mean skin temperature (± (SD)) are reported in table 5 for the occupants exposed to the use of CF fan and for subjects not using their desk fan. The difference was found to be significant in condition A (p <0.0001) and condition B (p <0.002) for the forehead skin temperature, confirming the cooling effect of the CF fans. A slight decrease of forehead skin temperature was noticed with the increasing air velocities. No significant difference was observed between male and female.

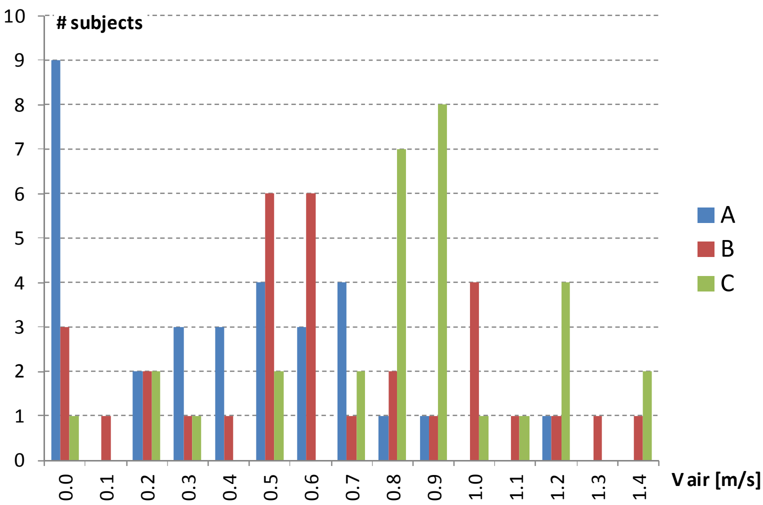

Figure 3: Number of subjects choosing a certain air velocity.

TEMPERATURE ±SD	Forehead (°C)		Right hand (°C)		Left hand (°C)	
Condition	CF-ON	Fan-OFF	CF-ON	Fan-OFF	CF-ON	Fan-OFF
A	33.0±0.4	34.2±0.8	31.6±1.0	32.4±1.1	31.6±1.0	32.4±1.0
B	33.8±0.6	34.5±0.5	33.1±0.8	33.5±0.7	33.3±0.9	33.2±0.7
C	34.2±0.6	34.8±0.7	33.9±0.7	34.0±0.8	34.1±0.6	34.0±0.8

Table 5: Mean skin temperature of forehead, right and left hands of subjects using the CF fans and not using local air movement.

Responses on the thermal environment

Table 6 shows the mean thermal sensation votes (TSV) for females and males at the three investigated conditions. Females felt slightly warmer than males in condition C, but the difference was not statistically significant. The mean TSV increased with increasing room temperatures, passing from neutrality (0) in condition A to “slightly warm” (+1) in condition C. A comparison of mean TSV between subjects using the CF fan and not using any local air movement is shown on the right side in Table 6. In conditions B and C a slight difference can be noticed, as expected the occupants using the fans tend to have a slightly cooler thermal sensation; but that is not statistically significant.

The thermal environment for subjects using CF fans was generally considered acceptable in conditions A and B, while it became critical in condition C, with a percentage of dissatisfied rising up to 55%.

TSV ±SD				
condition	**females**	**males**	**Fan On**	**Fan Off**
A	0.0±0.6	-0.1±0.7	0.0±0.6	0.0±0.6
B	0.5±0.6	0.7±0.9	0.5±0.7	0.8±1.0
C	1.5±0.6	1.1±0.7	1.3±0.7	1.7±1.1

Table 6: TSV organized by sex and by the use or not of the CF fan

Figure 4a reports the actual TSV and the Predicted Mean Votes (PMV) of subjects using the CF fans. As shown in figure 4b, the PMV model confirmed to be a fairly good predictive tool for Danish people. In the warm side the percentage of dissatisfied rapidly increases from “neutral (0) to “slightly warm” (+1) to “warm” (+2) as for the PMV/PPD model; while a large discrepancy was found in the “slightly cold” (-1) condition where 26% of people dissatisfied were expected.

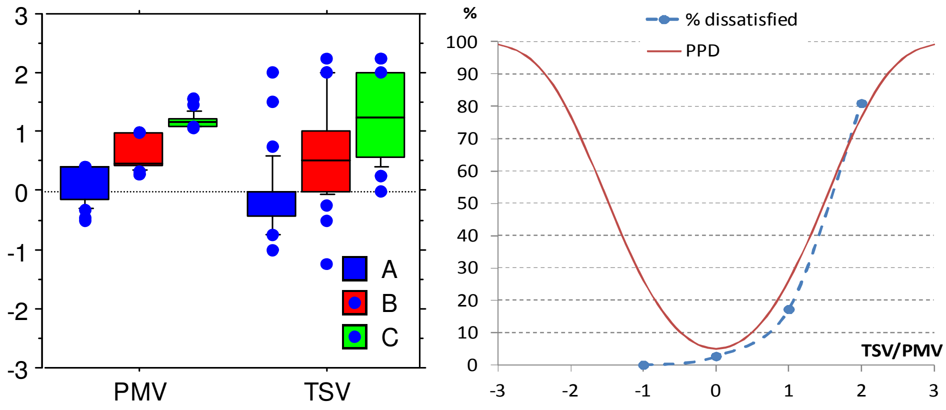

Figure 4: a) Comparison of PMV and TSV; b) Comparison of PMV/PPD model with present results.

The thermal preferences at the three conditions are reported in Table 7. Despite the subjects could increase the air movement, they were not able to reach a satisfying thermal comfort. In fact, in condition C about 85% of the subjects operating CF fans were asking for a cooler environment.

Thermal preference			
condition	**warmer**	**no change**	**cooler**
A	4	13	5
B	0	12	16
C	0	4	25

Table 7: Thermal preference at the three investigated conditions.

No correlation was found between TSV and preferred air velocity or TSV and forehead skin temperature.

Responses on air movement

The vast majority of the subjects using their desk fan could feel air movement. The body parts where the air movement was most commonly felt were the face (more than 95%) and the right arm (60% to 85% depending on the fan type). It was noted that 20 % of subjects keeping their fan off could anyway feel air movement. The mostly recurrent body part where the air movement was felt was the right arm, possibly due to the chosen setting (fan on the right side of the desk, thus blowing partly towards the right arm of the desk neighbour, see Figure 1b.

In Table 8 are reported the acceptancy of the use of different type of fan for the three investigated indoor thermel conditions. Reminding the limits conditions of the draught (DR) model (reported in ISO7730), DR were calculated at different air velocities for an environment having 26°C and 30% of turbolence (Tu) and reported in Table 9.

	CF		BL		SN		OFF	
	acc.	not acc.	acc.	not acc.	acc.	not acc.	acc.	not acc.
A	21	1 (5%)	2	4 (67%)	6	2 (25%)	43	3 (7%)
B	27	2 (7%)	10	4 (29%)	14	4 (22%)	10	6 (38%)
C	22	8 (27%)	11	10 (48%)	15	7 (32%)	2	2 (50%)

Table 8: Air movement acceptability for the different types of exposure.

DR %	5	9	13	17	22	27	32	37	42
V air [m/s]	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5

Table 9: Draught risk at 26 °C and 30% Tu for increasing air velocities

Discussion

Preferred air velocity

When analysing the preferred air velocity (v_{air}), a preliminary question pops up: "Preferred by who?". In order to give a complete view of the human behaviour and response in the present experiment, at least three different groups of subjects can be analysed.

The first group is constituted by all subjects sitting at a desk equipped with a CF fan. This group can be seen as the most representative, however about 30% of the subjects kept their CF fans off in condition A, resulting in low values of mean air velocity of the whole group.

A second group can be identified in the subjects actually using the CF fans. In this way the mean air velocity will not be affected by the null data of the subjects keeping their fan off. In this case, it is important to keep in mind what was the actual usage of the CF fans during the three conditions.

A third group can finally be constituted by those "comfortable subjects" expressing acceptability for both the thermal environment and the air movement, and voting between "slightly cold" (-1) and "slightly warm" (+1) in TSV scale. This group is not reflecting the actual mean level of satisfaction of the experimental sample, but it can be useful to point out which environmental parameters may potentially result in an acceptable thermal balance. It should be noticed that in condition C only 11 subjects (35%) are considered "comfortable".

The TSV and the preferred air velocity of the three mentioned groups is reported in Table 10, where also the effective number of subjects in the three conditions is indicated.

Condition / Group	**A**		**B**		**C**		**Nr. of subjects**		
	TSV±SD	v_{air} ±SD	TSV±SD	v_{air} ±SD	TSV±SD	v_{air} ±SD	A	B	C
	[-]	[m/s]	[-]	[m/s]	[-]	[m/s]	[-]	[-]	[-]
CF on&off	0.0 ±0.6	0.40 ±0.32	0.6 ±0.8	0.62 ±0.36	1.3 ±0.7	0.83 ±0.34	31	32	31
CF on	0.0 ±0.6	0.56 ±0.23	0.5 ±0.7	0.69 ±0.31	1.3 ±0.7	0.85 ±0.30	22	29	30
comfortable	-0.2 ±0.4	0.41 ±0.29	0.4 ±0.5	0.64 ±0.33	0.6 ±0.3	0.91 ±0.25	26	22	11

Table 10: TSV and preferred air velocity in the three investigated conditions.

PMV and TSV

The PMV resulted to be a good indicator of TSV. The PMV index was actually devised on the basis of tests conducted mainly on subjects from temperate climates. Studies conducted in tropical climates [13-15] show that the PMV overestimates the thermal sensation of people, and it is today recognized that it is necessary to take into account the adaptation of people to their local climate.

A study by Humphreys & Hancock [16] questioned "what is the actual preferred thermal sensation of subjects expressing their TSV on an ASHRAE scale?". They found that the most common personal desire was "neutral" followed by "slightly warm" and that it varied with the TSV currently experienced. The data were collected in dwellings and lecture rooms for temperatures between 16°C and 24 °C. In the present work the same question resulted in "neutral" tending to "slightly cool" desire for condition A and B, and "slightly cool" in condition C. The different result suggests that culture and climate may affect people's thermal preferences.

On SN fans use

The SN fans provided an average air velocity of 0.7 ÷ 0.8 m/s, with a dynamic profile of air velocities and gusted up to 1.4 m/s. The comparison between TSV of the present study with the previous one performed by Hua et al. [11] is shown in Figure 5. In the Chinese experiment the subjects felt on average cooler than the Danish ones, with a difference in TSV of 0.6-0.7 units on the ASHRAE TSV scale at 28°C and 30 °C.

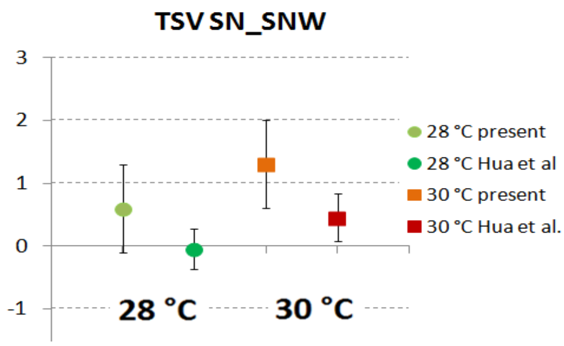

Figure 5: Comparison between TSV of the present study and Hua`s similar experiment.

Sixty per cent of the subjects using the SN fans were asking for less air movement in condition A, while 40% of them requested more air movement in condition C. This may imply that the dynamic airflow profile should be adjusted to the room temperature in order to guarantee a proper cooling effect.

On BL fans use

BL fans could not be operated at low air velocities (lower than 1.3 m/s), it implied that a consistent number of subjects were asking for less air movement and when possible preferred to turn it off. The BL fans had a strong cooling effect in condition A (see figure 6), causing a mean drop in the forehead skin temperature of about 1.4 °C more than the one obtained by the use of CF fans that was 2.6 °C lower than when no local air movement was used. The difference is less evident in the room conditions where temperatures are higher

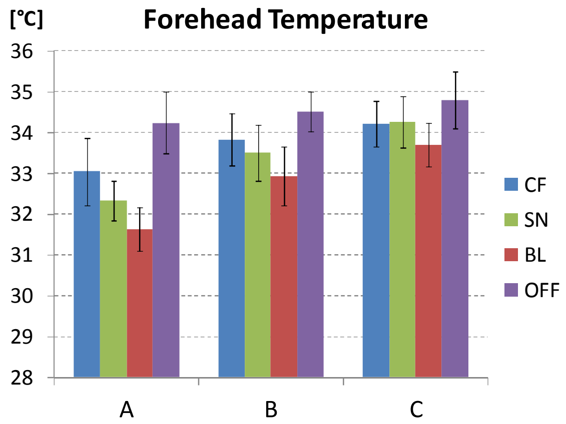

Figure 6: Forehead mean skin temperatures at different room conditions and fans exposure

The bladeless fans present an attractive design style, and the different airflow pattern (a rather constant, non-buffeting airflow) that could result in a high level of comfort. However, new prototypes with also a wider range of air velocities, starting from 0.2 m/scirca, is suggested.

Impact of the operability of CF, BL, and SN fans

The characteristics of the desk fans imposed several limits of use:

- The CF fans had an upper air velocity setting of 1.2-1.4 m/s. It implied that the subject could not further increase the air velocity. However, none of the subjects choosing the highest air velocity asked for more air movement.
- The experimenter was setting the simulated-natural mode of the SN fans. It appeared having an influence on the behaviour of the occupant that repeatedly asked for changes of the fan setting.
- The operability of BL fans operating only at high range of air velocities, reduced the flexibility of use in a sort of on/off use.
- A minor role could have been played by the **aesthetics** of the three types of desk fans, both caused by their “design pleasantness” and by the “perceived familiarity”.
- Noise appeared to be one of the reasons why the occupants preferred not to increase the air velocity of the provided fans having instead a warmer thermal sensation. That was mainly related to the BL fan use that increased from 45 dBA to 54 dBA in the middle of the room when it was switched on and it could reach 72 dBA at the occupant place.

Several subjects indicated that their preferred air velocity was chosen as a trade off between the cooling effect and the drying effect on the eyes.

The present study confirmed the findings of Fang et al. [12], who observed that the perceived air quality (PIAQ) decreases with increasing air temperature and humidity. The use of desk fans has to be carefully considered also as regards the eventuality of cross infection.

Conclusions

The experimental measurements showed that higher air velocity and personal control make the indoor environment acceptable at higher air velocity with a benefit on energy consumption applicable during the summer seasons and in warmer countries. There was significant individual difference in the preferred air velocities, which indicate that personal control is important. The accepted air velocities depended on the type and source of the increased velocity.

The PMV resulted to be a good indicator of TSV, however the PPD curve overestimated the per cent of people dissatisfied. The “slightly cool” sensation was actually chosen by 45% of subjects as preferred TSV in condition C, suggesting that culture and climate may affect people’s thermal preferences.

The responses from subjects exposed to a simulated natural airflow suggest that the dynamic airflow profile should be adjusted to the room temperature in order to guarantee a proper cooling effect.

Although the bladeless fans had a consistent cooling effect, their low flexibility of use resulted in a large number of subjects dissatisfied. New prototypes with a wider range of air speeds should be designed for a deeper investigation on the potentialities of this technology.

The fan usage did not show correlation with perceived indoor air quality, while it was observed that with the increasing of air temperatures the indoor air quality was negatively perceived.

Acknowledgements

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Strategies for controlling thermal comfort in a Danish low energy building: system configuration and results from 2 years of measurements

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Abstract

The thermal comfort of the residential building Home for Life is investigated with a particular focus on the strategies used to achieve good thermal comfort, and the role of solar shading and natural ventilation. Home for Life was completed in 2009 as one of six buildings in the Model Home 2020 project. It has very generous daylight conditions, and is designed to be energy neutral with a good indoor environment.

The kitchen/dining room has a large south-facing window area and is selected for the detailed analyses. The thermal environment is evaluated according to the Active House specification (based on the adaptive method of EN 15251), and it is found that the house reaches category 1 for the summer situation. Some undercooling occurs during winter, causing the room to achieve category 2 if the entire year is considered. The undercooling is due to the occupants' preferred balance between indoor temperatures and heating consumption.

It is found that ventilative cooling through window openings play a particularly important role in maintaining thermal comfort.

Keywords

Thermal comfort, ventilative cooling, residential buildings, natural ventilation, solar shading.

Introduction

Background

Five single-family houses in five European countries were built between 2009 to 2011 as a result of the Model Home 2020 project. The first house (Home for Life, Denmark), was completed in spring 2009 and has been occupied by two different families, of which the last family has bought the house. Measurements were performed for a full year during the occupancy of the two families. The results from year one have been reported already and compared to simulations [1, 2].

The six houses follow the Active House [3] principles which mean that a balanced priority of energy use, indoor environment and connection to the external environment must be made. In practice this means that the houses should have both an excellent indoor environment and a very low use of energy. There is a particular focus on good daylight conditions and fresh air from natural ventilation. The calculated daylight factors are seen at Figure 1.

Figure 2: Calculated daylight factors in “Home for Life”. Ground floor on the left, and upper floor on the right. The dotted rectangle indicates the kitchen and dining room which will be investigated in details.

Measurements of IEQ include light, thermal conditions, indoor air quality, occupant presence and all occupant interactions with the building installations, including all operations of windows and solar shading. Further, an anthropological study of the family’s experiences in the house was performed. Measurements of energy performance include space heating, domestic hot water and electricity for appliances, lighting and technical installations. The occupants also reported their own observations in a diary, and an anthropologist has followed the project and made structured interviews with the family.

The present paper describes the design and setup of the systems that have an influence on the summer, winter and intermediate situations, particularly the natural ventilation system and the solar shading.

The presented results focus on thermal conditions, effectiveness and experience with the applied strategies. Recent examples of demonstration houses in Scandinavia have experienced problems with overheating, often due to insufficient solar shading [4,5].

Technical systems

Home for Life is an experiment and the hypothesis is that a synergy between a low CO_2 emission and a good IEQ can be achieved through optimal window layout (40% of façade area) distributed towards all orientations and the roof for both good daylight conditions (daylight from all orientations) and good natural ventilation performance, through hybrid ventilation and through automatic control of solar shading, heating and ventilation. It is a 1½-storey house with a total floor area of 190 m^2.

The ventilation system is hybrid, i.e. natural ventilation is used during the summertime and mechanical ventilation with heat recovery during the wintertime, while hybrid ventilation is used spring and fall. The switch between mechanical and natural ventilation is controlled based on the outdoor temperature. The set point is 12,5 ^oC with a 0,5 ^oC hysteresis. Below the set point the ventilation is in mechanical mode, above the setpoint the ventilation is in natural mode. In both natural and mechanical mode, the ventilation rate is demand-controlled. CO_2 is used as indicator for the Indoor Air Quality, and a set point of 850 ppm CO_2 is used. Besides that, relative humidity is also used as indicator. When RH is 60% or higher, ventilation is increased step-wise to maximum ventilation, which is used when RH is 80% or higher.

There is external automatic solar shading on all windows towards South, and overhangs are used where appropriate. The solar shading was initially controlled based on the indoor temperature. But based on the responses from the occupants, this control strategy seemed to react too slowly to prevent overheating, and the control was changed so that external solar radiation was used instead. On the con side, using solar radiation can cause unnecessary use of solar shading in the winter period, leading to an increased heating demand.

Each room is an individual zone in the control system, and each room is controlled individually. There are sensors for humidity, temperature, CO_2 and presence in each room.

The solar shading is controlled by external solar radiation for each facade, and not at room level.

The building occupants can override the automatic controls, including ventilation and solar shading at any time. Override buttons are installed in each room, and no restrictions have been given to the occupants. As house owners they have reported a motivation to minimise energy use on an overall level, and to maximise IEQ on a day-to-day basis.

Figure 2: "Home for Life". South and east facades (left). Concept for daylight, ventilation and energy (right)

Data recording

The data from the sensors that are used for the controls of the house is recorded. The IEQ data is recorded for each individual zone as an event log, where a new event is recorded when the value of a parameter has changed beyond a specified increment from the previously recorded value. The event log files are automatically converted to data files with fixed 15-minute time steps, which are used for the data analysis.

Data analyses

The recorded temperature data is evaluated according to the Active House specification [3], which is based on the adaptive approach of EN 15251 [6].

Results

The results presented here are based on the measurements and analyses for the second year of occupancy. The main part of commissioning and adjustment of all systems took place during the first year occupancy, and year two is for that reason considered the most representative. Figure 3 shows the outdoor conditions during the reported period.

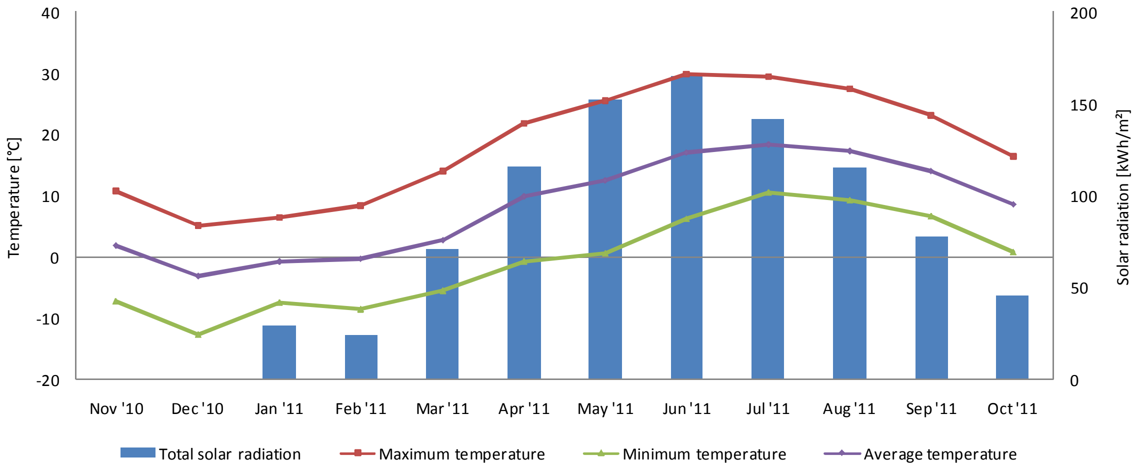

Figure 3: Outdoor conditions during the measurement period. The pyranometer was installed in January 2011, and therefore data is not available for the previous months.

Figure 4 shows that five rooms achieve category 2, while six rooms achieve category 4. It is clear from the figure that the majority of the hours in category 2, 3 and 4 are caused by low temperatures, i.e. undercooling rather than overheating. When undercooling is disregarded, all rooms except bedroom and scullery achieve category 1.

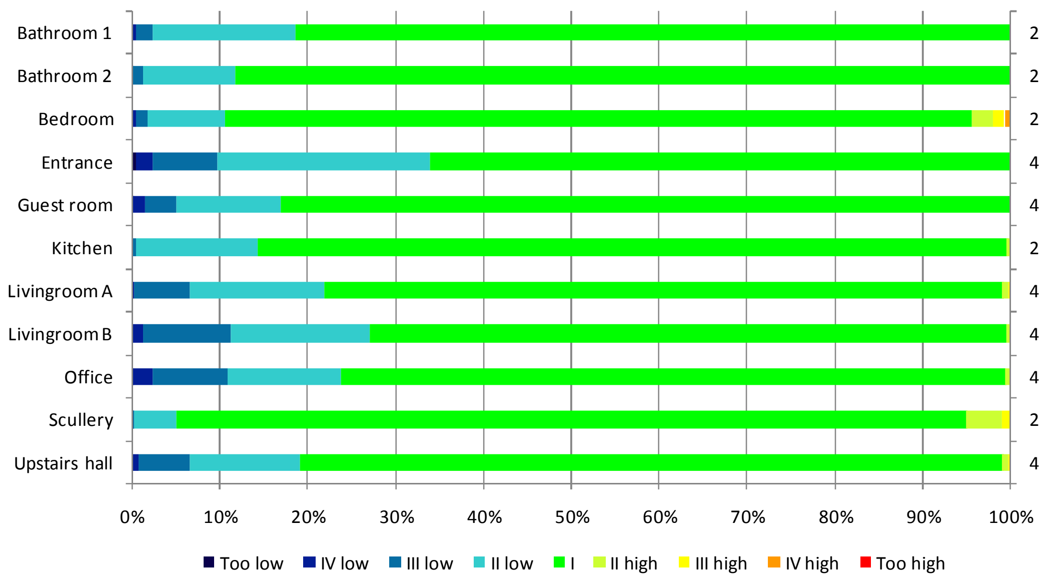

Figure 4: Thermal comfort for each of the rooms evaluated according to Active House specification (based on adaptive method of EN 15251). Criteria are differentiated between high and low temperatures. The number at the right side of the diagram indicates the score for each room (1 to 4).

The focus of the present paper is on the performance related to ventilative cooling and potential overheating. The further analyses will focus on the performance of the kitchen, which is a combined kitchen and dining room with a large south-facing window section, Figure 2.

The distribution of categories between months are seen in Figure 5. As expected from Figure 4, the undercooling is an issue in 4 winter months from November to February. From April to October, category 1 is achieved.

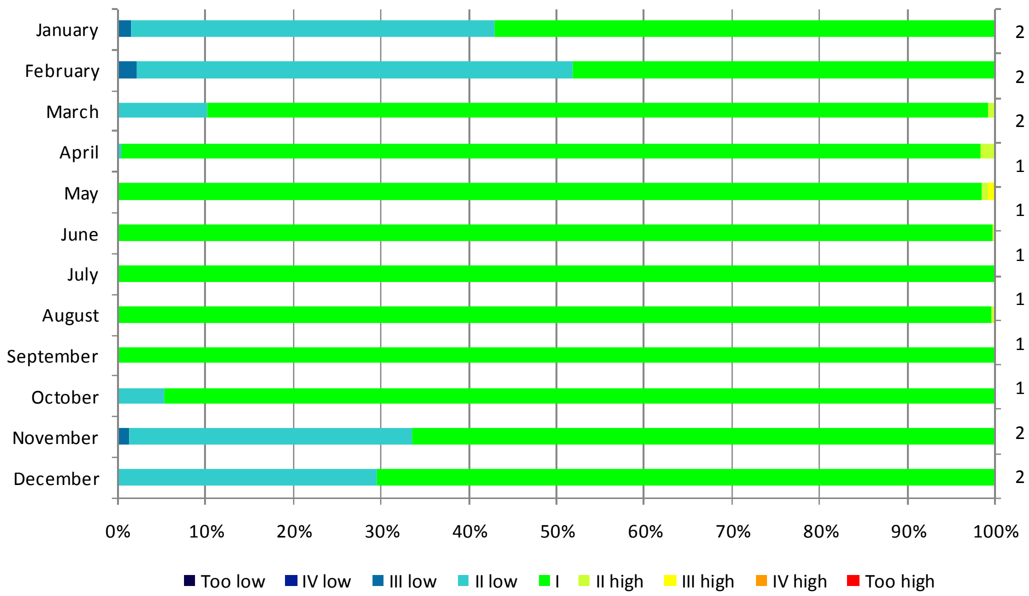

Figure 5: Thermal comfort categories for each month of the year for the kitchen and dining room. The number at the right side of the diagram indicates the score for each month (1 to 4).

Figure 6 shows the indoor temperature at each hour of the year plotted against the running mean outdoor temperature as defined in EN 15251. It is seen that temperatures below the category 1 limit (21 °C) occur both in winter and in the transition periods, but only with a few hours below the category 2 limit (20 °C). It is suspected that the episodes with temperatures below 21 °C are caused by either manual or automatic airings, or more simply by user preference. The occupants have not reported discomfort due to undercooling in their diaries.

Some episodes with temperatures above 26 °C are also seen during winter and in the transition periods, suggesting large variations in temperature during short periods of time. This is suspected to be due to solar gains. The automatic control of window openings and solar shading is setup to prevent overheating, but especially during winter the system will accept high solar gains to reduce the heating demand.

During summer the system prioritizes to maintain thermal comfort, and Figure 6 shows very limited overheating during summer, with only a few episodes with temperatures above category 1. Relatively low temperatures are observed during summer, with episodes with temperature drops below 21 °C. This is suspected to be caused by night cooling, where the temperature decreases during the night to reduce overheating the following day, which in some situations lead to temperatures in the morning between 20 °C and 21 °C.

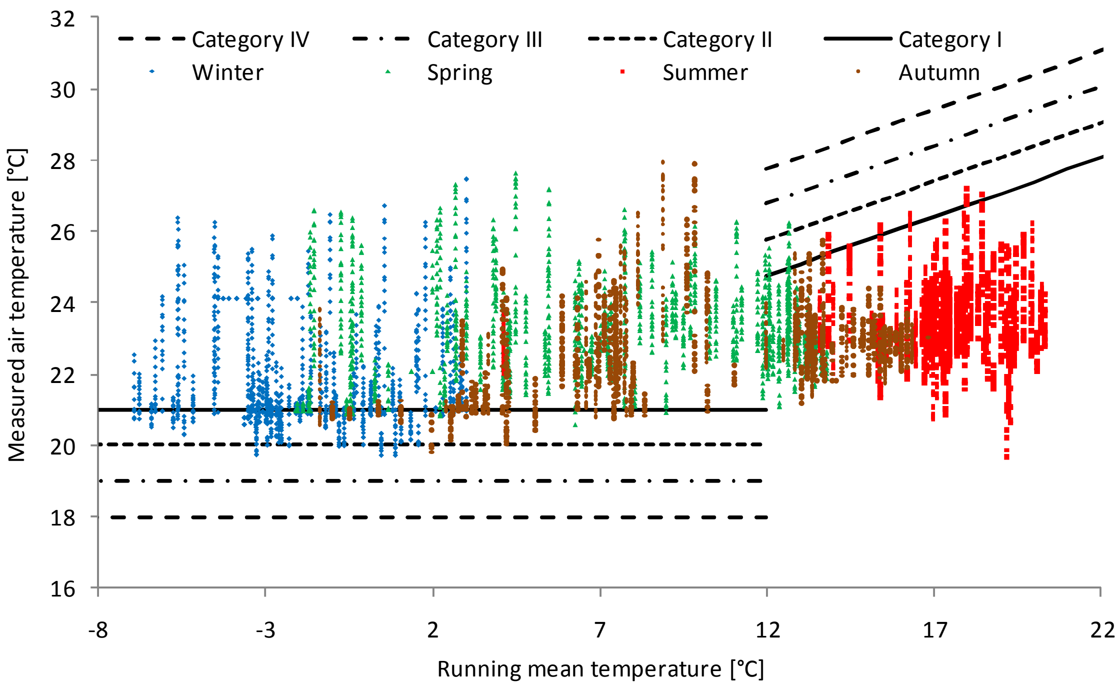

Figure 6: Indoor temperatures in the kitchen plotted against running mean temperature for each hour of the year including the Active House category limits. The dots are coloured to represent a season.

The variation over time-of-day and time-of-year is further investigated in Figure 7. It is seen that the episodes during winter with temperatures below category 1 can last for several days during the winter, but that in many of the episodes, the temperature reaches category 1 between 12:00 and 20:00, possibly due to solar gains.

During summer, only few episodes with temperatures beyond category 1 are observed.

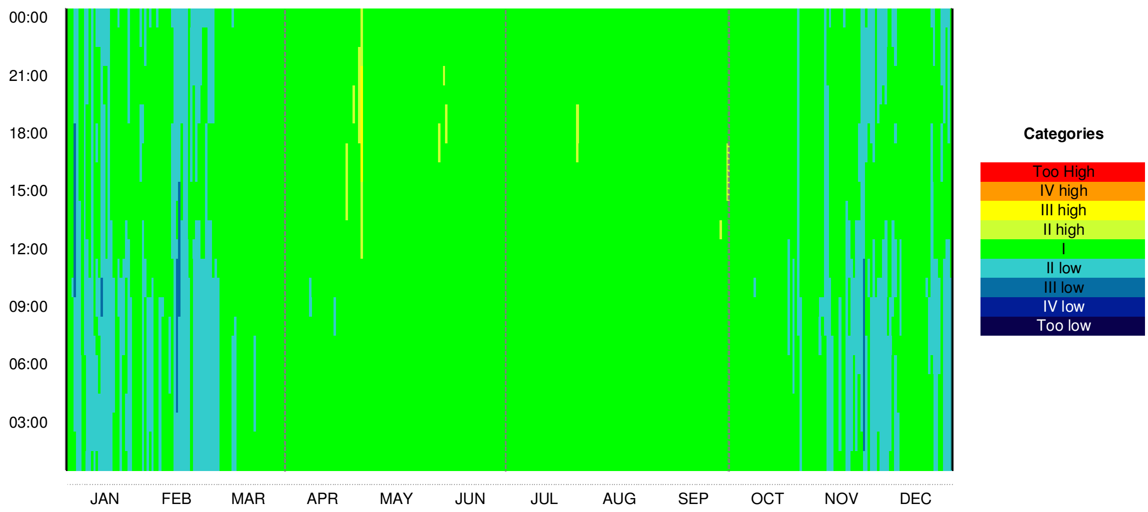

Figure 7: The comfort category of each hour of the year is plotted as a temporal map (kitchen and dining room)

To investigate the role of window openings in maintaining comfort, Figure 8 is used. A rather strict comfort definition is imposed for the sake of the analysis (category 2 was the design target), where only category 1 is considered comfort. The figure also shows if windows where active during each hour.

Figure 8 shows that windows were not open during the winter episodes with temperatures below category 1 (orange), indicating that these episodes were not caused by airings. The heating system during winter is controlled in such a way that the supply temperature for the floor heating system is set at the heat pump control. The lower the supply temperature, the better the system efficiency. The occupants have reported that they set the supply temperature so that the room temperature would reach 20-21 °C to reduce heating consumption. The episodes with winter temperatures below category 1 can thus be attributed to user preferences.

A few episodes with red colour are seen during summer in the late afternoon, indicating that overheating occurred and that windows were opened, but that this was not sufficient to maintain category 1.

Figure 8 further shows that during the summer, windows are almost permanently open between 9:00 and 22:00 and that category 1 is maintained during these hours (green). The figure shows many episodes with open windows between 22:00 and 9:00 (green), which can be assumed to be caused by automatic window opening for night cooling. Also in the transition periods (March to May and September to October) windows are used to a large extent, with openings between 12:00 and 18:00 as a typical episode (green).

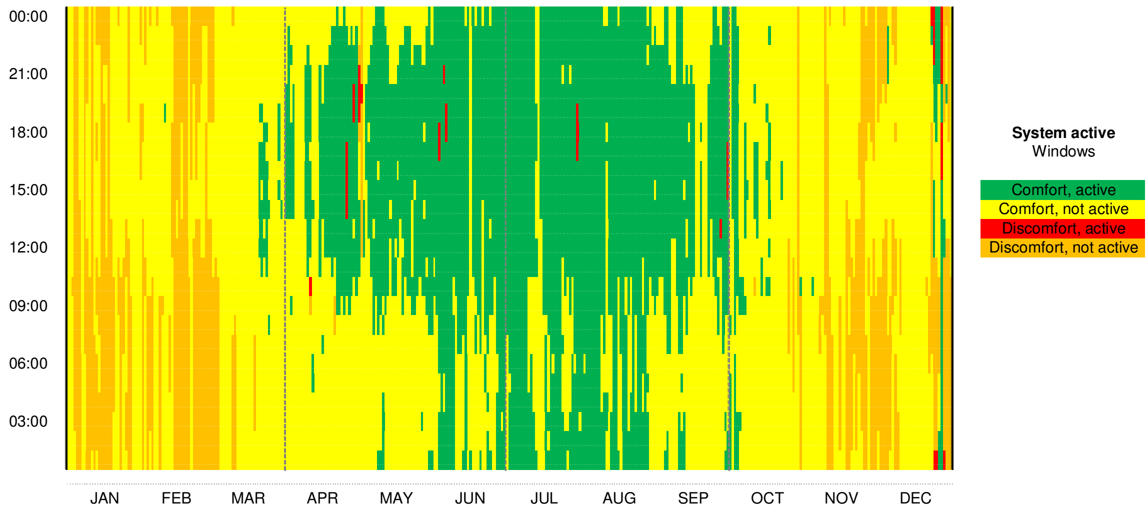

Figure 8: Temporal map showing comfort or discomfort (discomfort is here temperatures in category 2, 3 or 4) and if windows were open or closed (active or not active). Figure shows kitchen and dining room.

The occurrence of windows in relation to outdoor temperature is further investigated at Figure 9. It is seen that windows are generally closed (red dots) when the running mean temperature is below 10 ºC. When the running mean temperature is above 12 ºC, windows are generally opened when the indoor temperature exceeds 22-23 ºC, which is in accordance with the control strategy.

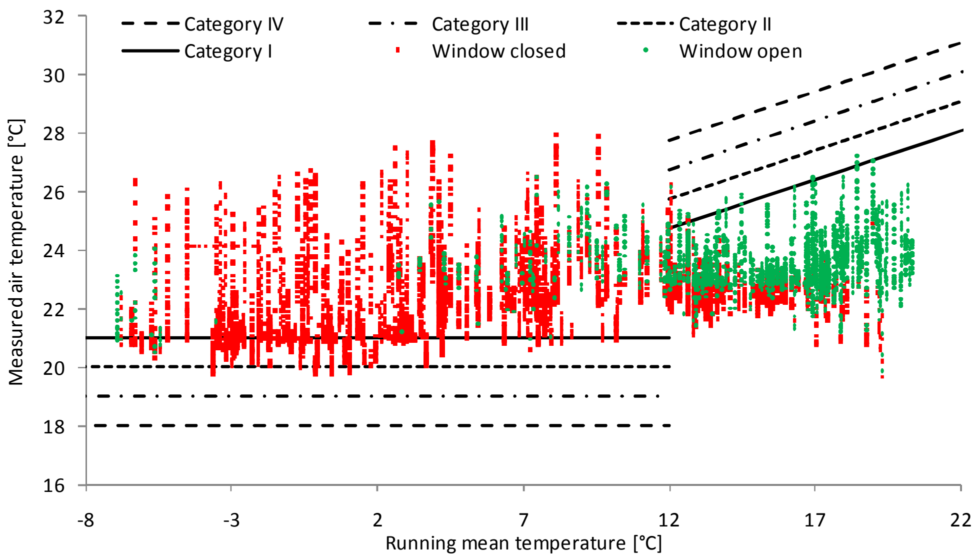

Figure 9: Indoor temperature vs. running mean outdoor temperature. The colour indicates if windows were open (green) or closed (red) for each hour (kitchen and dining room)

The role of the external solar shading is investigated at Figure 10. The figure shows no clear correlation between use of shading and indoor temperature nor running mean temperature. The external shading is activated when the external solar radiation exceeds a threshold and so is not controlled by temperatures, but some relation between temperature and use of shading could have been expected, as was the case for window openings. A further explanation is that solar shading can also be activated if glare is experienced, and also for privacy.

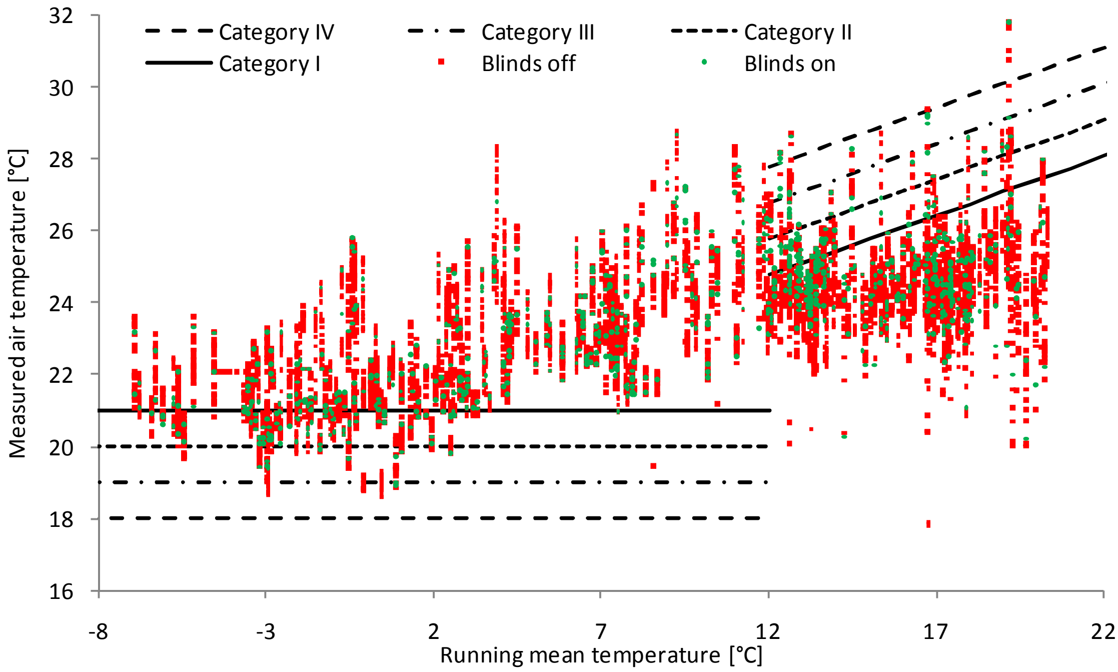

Figure 10: Indoor temperature vs. running mean outdoor temperature. The colour indicates if external solar shading was active (green) or inactive (red) for each hour (kitchen and dining room).

To further investigate the temporal use of shading, the use of shading is plotted at Figure 11, where colour codes indicates the use of shading and the comfort categories. The use of shading is increased during summer (green dots). The use appears to be distributed evenly over the day, due to the control being based on external radiance. The consistent use of shading during an hour in the morning and the late afternoon/evening has been investigated, but no clear explanation is found. It is not suspected to be caused by user controls, but rather as a result of the automatic control algorithms.

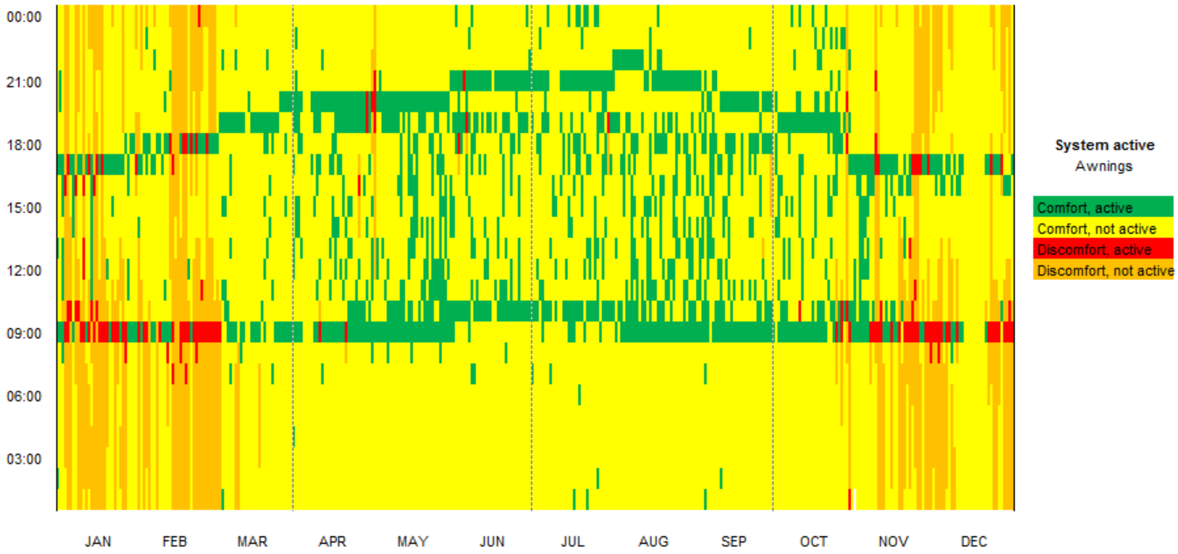

Figure 11: Temporal map showing "comfort" or "discomfort" (discomfort is here temperatures in category 2, 3 or 4) and if shading was active or not active (kitchen and dining room).

Discussion

For the rooms in Home for Life, half fall in category 2 and the other half in category 4 with regards to thermal conditions, when evaluated according to the Active House specification, which uses the same methodology and criteria as EN 15251 with regards to thermal comfort. The hours not in category 1 are mainly hours with undercooling, while overheating is rare. If undercooling is disregarded, the primary rooms of the house achieve category 1. For low energy houses, overheating should be prevented by the building design, as overheating may require substantial measures if handled after completion. Home for Life thus meets the category 1 with regards to overheating, which is very satisfactory, given the generous daylight conditions.

The episodes with undercooling could be caused by insufficient heating capacity, window airings, poor building airtightness or occupant preferences. It was found that there was no correlation between window openings and undercooling. The airtightness has been verified by a blowerdoor test. The heating system is known to have a sufficient capacity, but the supply temperature was actively reduced by the occupants to reduce the heating consumption. Undercooling in Home for Life is therefore explained by occupant preferences.

In the kitchen/dining room, a correlation between window openings and the combination of high indoor and outdoor temperatures was found. Further, a clear correlation between window openings and acceptable thermal comfort was found.

This indicates that window openings have contributed to achieving and maintaining good thermal conditions.

No clear correlation between use of external solar shading and temperature. Users may often have used the override function to deactivate the automatic control of solar shading, which could explain the missing correlation between use of shading and the combination of high indoor and outdoor temperatures.

In conclusion, Home for Life achieves a good thermal performance in real use, which should be seen in connection to the high daylight levels of the building. The good performance is achieved with automatic control of window openings and solar shading, where especially the ventilative cooling from open windows was important.

Acknowledgements

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Natural ventilation Strategy Potential Analysis in an existing school building

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Abstract

Natural ventilation is increasingly considered a promising solution to improve thermal comfort in buildings, including schools. However in order to support its planning and implementation, quantitative analysis on airflow paths and heat-airflow building interactions are needed. This requires an adequate accounting of both internal effects, from building layout and structure, and external forcing from atmospheric factors.

The paper analyses the performances of natural ventilation strategies as retrofit solutions to improve thermal comfort in an existing school building in Lavis (Trento, Italy).

A climatic analysis is performed to define the potential of wind driven natural ventilation. Meteorological data collected on site are analysed to identify typical wind conditions during the cooling season. The resulting daily cycle of wind speed and direction in sunny days reflects the typical dynamics of a regular valley wind, but also displays the peculiar characteristic of being strongly affected by the outbreak of a lake breeze flowing from a nearby valley and originated from Lake Garda.

Based on these findings, three natural ventilation strategies are proposed (night cooling, wind driven cross ventilation and stack and wind driven cross ventilation), and their effectiveness on thermal comfort are compared by means of dynamic simulation tools.

The thermal comfort in classrooms is evaluated according to the standard UNI EN 15251. For a standard occupant behaviour, discomfort situations from overheating occur in 34% of occupational period hours in the spring-summer season. The proposed ventilation strategies allow to reduce this value by up to 4%. Natural ventilation turns out to be an interesting low cost solution to control indoor temperatures without mechanical cooling systems.

Keywords

Natural ventilation, school building, passive cooling, thermal comfort, wind.

Introduction

Existing school buildings in Italygenerally have no cooling system. However nowadays an increasing number of overheating discomfort situations occurs, as school building facilities are used also during the summer season for extra-scholastic activities. These situations are likely to increase, as a consequence of observed modifications of local atmospheric regimes, connected with either urbanisation [1] or global warming effects [2].

Natural ventilation seems a promising technique to improve thermal comfort in classrooms [3]. Besides requiring higher energy performances, the Italian legislation (D. Leg. 311/2006) recommends the exploitation of natural ventilation to reduce cooling demand. A more recent regulation (D. Leg. 5/2012) promotes the modernization of school buildings, improving energy efficiency and reducing management costs.

Natural ventilation as passive cooling strategy is particularly suitable in school buildings, as they have a defined use pattern and a flexible indoor layout [4].The present paper analyses the performance of natural ventilation strategies as retrofit solutions to improve thermal comfort in an existing school building located in Lavis, a suburban zone near the city of Trento (Italy), in the Alpine Adige Valley.

Building description

The Lavis secondary school was built about seventy years ago, and renovated in the last years (Figure 1). There is no cooling systems or mechanical ventilation plants, except in the canteen underground. The four story building has a very complex layout due to the recent extensions. The present analysis focuses on the southern building part, where the classrooms are located. Every floor has seven classrooms connected by a corridor to the rest of the building and to two stairwells. One of the stairwells is enclosed by fire resistive elements. Two classrooms are oriented eastward, two westward, while three classrooms face south. Each classroom has two windows with triple-casement and tilt and turn opening. No vents are installed. Teachers and students report discomfort situations during the middle season. The rest of the building is allotted to services, offices, library and laboratories for artistic and musical activities, and has not been modeled in detail for the purpose of the present analysis.

Figure 1: Lavis School location. Source: © Google 2012

Wind potential analysis

In the design of natural ventilation it is mandatory to understand how and when a wind induced internal flow is exploitable. To accomplish this goal, the building orientation and the internal space layout with respect to the main surrounding wind directions must be considered.

The Adige Valley in the Alps, where Lavis lies, is north-south orientated. Furthermore two tributary valleys join the Adige Valley near Lavis: the AvisioValley, east of the town, and the Lakes Valley, south-west of Lavis. Therefore wind speed and direction are expected to be strongly influenced by airflows occurring in this complex topography. During sunny days in the warm season valley winds generally blow in the above-mentioned valley. Valley winds, which typically blow up-valley during the day and down-valley at night, develop as a consequence of the horizontal pressure gradients due to the temperature differences between different valley cross-sections or between the valley and the plain [5]. Moreover the local circulation blowing in the Lakes Valley is not a typical valley wind, but a combined “valley-lake” circulation, which starts blowing on the shores of Lake Garda, located in the southern part of this valley. This “valley-lake” breeze is generally rather strong and arrives into the Adige Valley from the Lakes Valley in the early afternoon.

In the present case typical daily cycles of wind speed and direction representative of the conditions occurring around the school building have been calculated from wind measurements acquired every 10 min by an amateur weather station located close to the school. A statistical analysis performed on wind speed and direction data for the whole 2011 shows that during the spring-summer season a typical daily cycle of wind speed and direction occurs (Figure 2). At night and in the early morning wind blows from east, following the down-valley wind flowing from the Avisio Valley, as the building is located close to the end of this tributary valley. Later in the morning and in the early afternoon wind blows from south-west, following the development of the up-valley wind and the outbreak of the “valley-lake” breeze from the Lakes Valley. Finally in the evening a transition can be seen from an up-valley wind, to a wind from east flowing from the Avisio Valley.

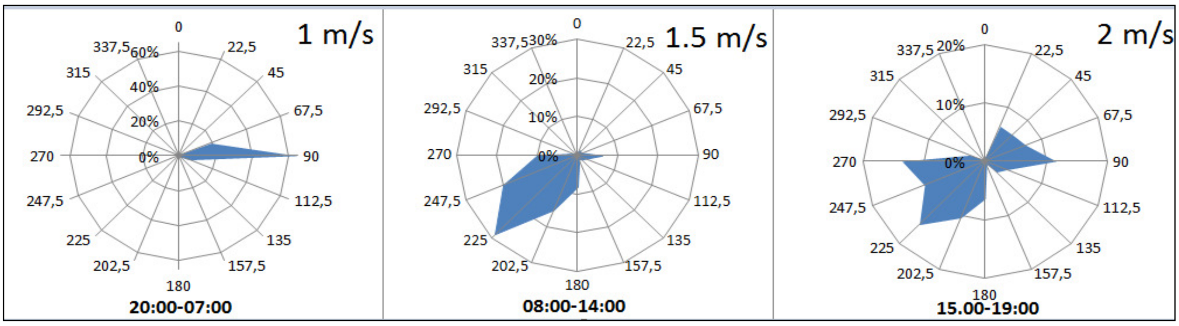

Figure 2: Average wind speed and directions in the typical day of spring-summer period.

Natural ventilation strategies

The three main strategies proposed here are based on fundamental principles of natural ventilation and implemented to the case study, namely (i) night cooling, (ii) wind driven cross ventilation and (iii) stack driven cross ventilation.

Night ventilation (Figure 3) rejects excess heat cooling the building structure, taking advantage of the lower night external temperatures [6,7]. Opening actuators can be applied to the existing windows, controlled by external temperatures and humidity sensors. Night ventilation is activated if:

- outdoor temperature is lower than indoor temperatures;
- indoor temperature is higher than 24°C;

- outdoor temperature is higher than 14°C;
- it is not raining.

During the day a venting schedule based on the current school occupation pattern and use has been set.

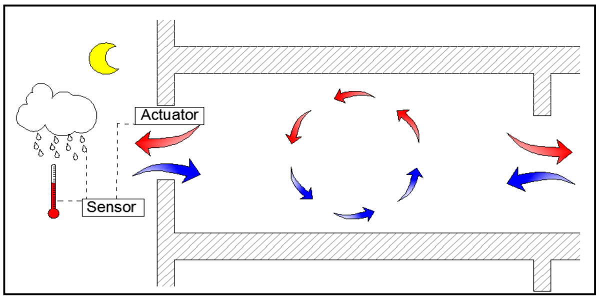

Figure 3: Night ventilation strategy.

During school time classroom doors stay closed, and single-sided ventilation occurs when windows are opened manually by the occupants. A wind driven cross ventilation (Figure 4) can be easily implemented by installing vents above doors, allowing airflow from one side to the other of the building even if doors stay closed. Specific vents that combine acoustic attenuation, to fulfil acoustic standard requirements in classrooms, with very low airflow resistance (discharge coefficient c_d = 0.71) are available on the market.

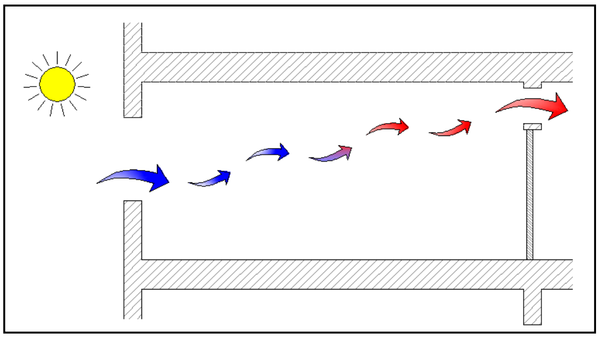

Figure 4: Wind driven cross ventilation strategy.

The third strategy proposed exploits the staircase in the South-East part of the building to increase the stack effect (Figure 5). A stack driven cross ventilation can be implemented by connecting corridors to the staircase through fire-resistant vents and adding a chimney on the top of the staircase. Classroom openings act as inlets allowing fresh air flowing through the building driven by the passive stack force of the exhaust air outlet at the top of the staircase. Some companies are specializing in the field of natural ventilation and provide specific products.

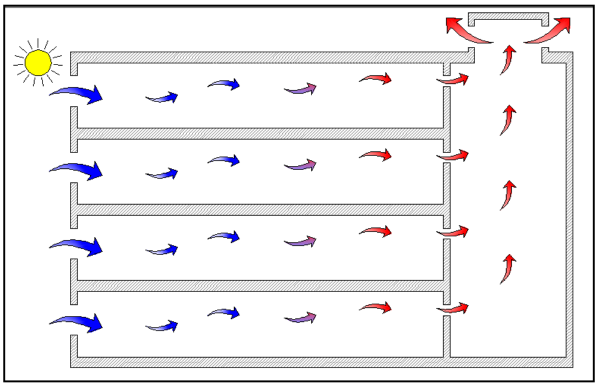

Figure 5: Stack driven cross ventilation strategy.

The THERMAL – AIRFLOWNETWORK SIMULATION model

The building model has been set up in Design Builder v.3, a graphical interface of the EnergyPlus building energy simulation engine [8, 9]. The southern building part (grey volume in Figure 6) has been divided into thermal zones, according to occupation patterns, occupant activities, comfort needs, zone orientation and elevation [10]. The rest of the school building is modeled as an adiabatic block (red volume in Figure 6).

Standard year weather file data for the city of Trento are used [11]. Wind speed profile is modified by terrain roughness parameters for suburbs.

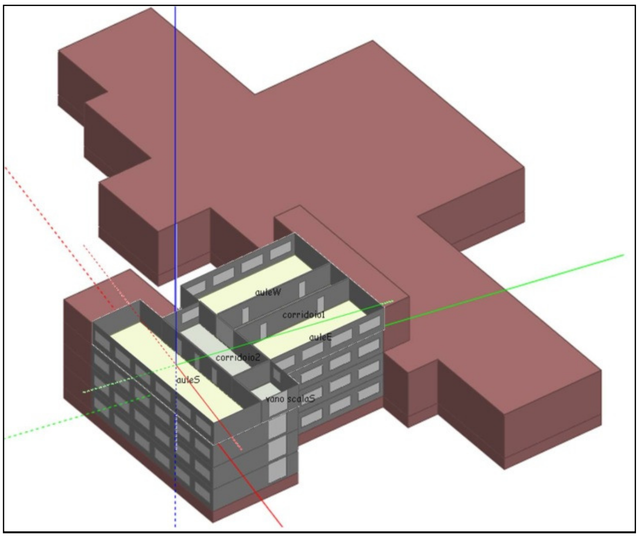

Figure 6: Lavis School model in Design Builder.

The proposed natural ventilation strategies were modeled by means of the AirflowNetwork object implemented in EnergyPlus [11] and compared. The airflow network object represents each thermal zone as a node of a network, characterized by a uniform temperature and a pressure varying hydrostatically (Figure 7). Windows, vents and cracks are represented as leakages, which link pressure differences due to wind or air density variations to power law equations. Each airflow path is calculated by Bernoulli equations and numerical solution of the problem gives the thermo-hygrometric conditions in the nodes. When temperatures and pressure are known, the program calculates airflow rates and latent and sensible heat exchanges. Thermal and airflow network model are coupled: they run in sequence and each uses the results of the other model (zone temperatures and pressures, airflow) in the previous time step [13].

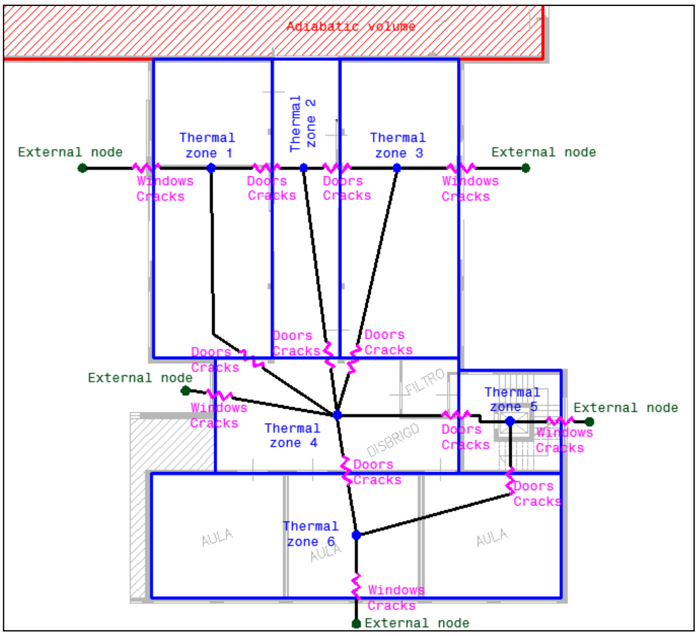

Figure 7: Schematic representation of Airflow Network in a floor type.

The simulations are performed in free-running mode to analyze the passive behavior of the building during the cooling season (15 April - 15 October). External openings are controlled by time schedules and by external temperature and humidity. Comfort temperatures are calculated from the CEN 15251 adaptive comfort model. Windows open if venting availability schedule allows venting and zone operative temperature is higher than the comfort one. Doors and vents are controlled by venting availability schedules according to a school standard occupation pattern. In general, doors must be closed during lesson periods and are a barrier to the natural airflows. Vents stay open every time. AIVC wind pressure coefficient data sets [14] are used to calculate wind-induced pressure on each external node. They are defined for wind incidence angles in 45°increments for each surface in the model.

Results

The three natural ventilation strategies proposed have been compared in terms of thermal comfort conditions. The European standard UNI EN 15251 defines three comfort categories limited by three temperature ranges. Thermal comfort is evaluated depending on the difference between the optimal operative temperature, according to the climate condition in the last week, and the simulated operative temperatures. The operative temperature outputs are compared with the comfort categories defined in the standard.

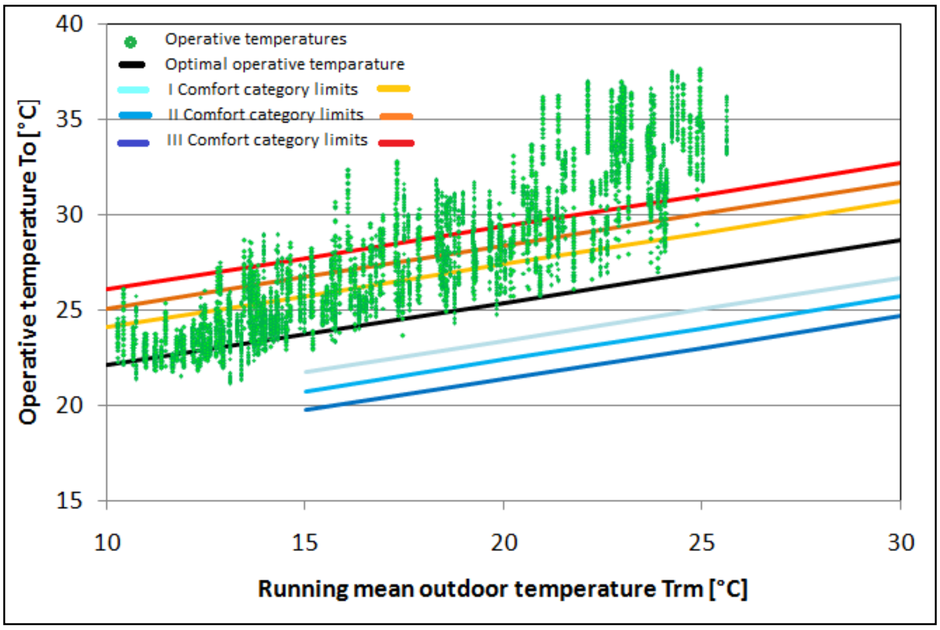

Figure 8: Base case simulation results. Operative temperatures of the most disadvantaged thermal zone (south oriented – third floor) are plotted against the outdoor running mean temperature. Green points above the superior limit of the third comfort category represent discomfort conditions due to overheating.

Simulations results of the most disadvantaged thermal zone, i. e. the one oriented to south at the third floor, were compared in the base case and in the natural ventilation strategies cases. In fact, the worst discomfort conditions are complained for that building part (Figure 8). Figure 9 shows the percentage of time during the occupation period when operative temperatures are within the comfort categories bounds.

Assuming a standard occupant behavior, discomfort situations occur in the 34% of occupational period hours during the spring-summer season due to overheating.

Discomfort situations can be reduced to 5% by the natural night ventilation strategy, to 4% by the cross driven natural ventilation strategy and to 8% by the stack driven natural ventilation strategy. Furthermore, the percentage of hours when operative temperatures are within the first category boundaries increases significantly: from 43% in the base case, to 76% by natural night ventilation, to 69% by cross driven natural ventilation and to 59% by wind and stack driven cross ventilation. There is a 2% of time in which UNI EN 15251 standard comfort can not be applied as the Running Mean Outdoor Temperature is out of the required limits: upper limit temperature has to be between 10°C and 30°C and lower limit temperature has to be between 15°C and 30°C.

Simulation results show the global behaviour of the building in the assumed standard conditions. Mean values of solar and internal loads during the whole cooling season have been compared with the estimated heat rejected by airflow and the results are shown in Figure 10. It could be noted that the best results are reached by night ventilation, where the 90% of internal gains can be rejected.

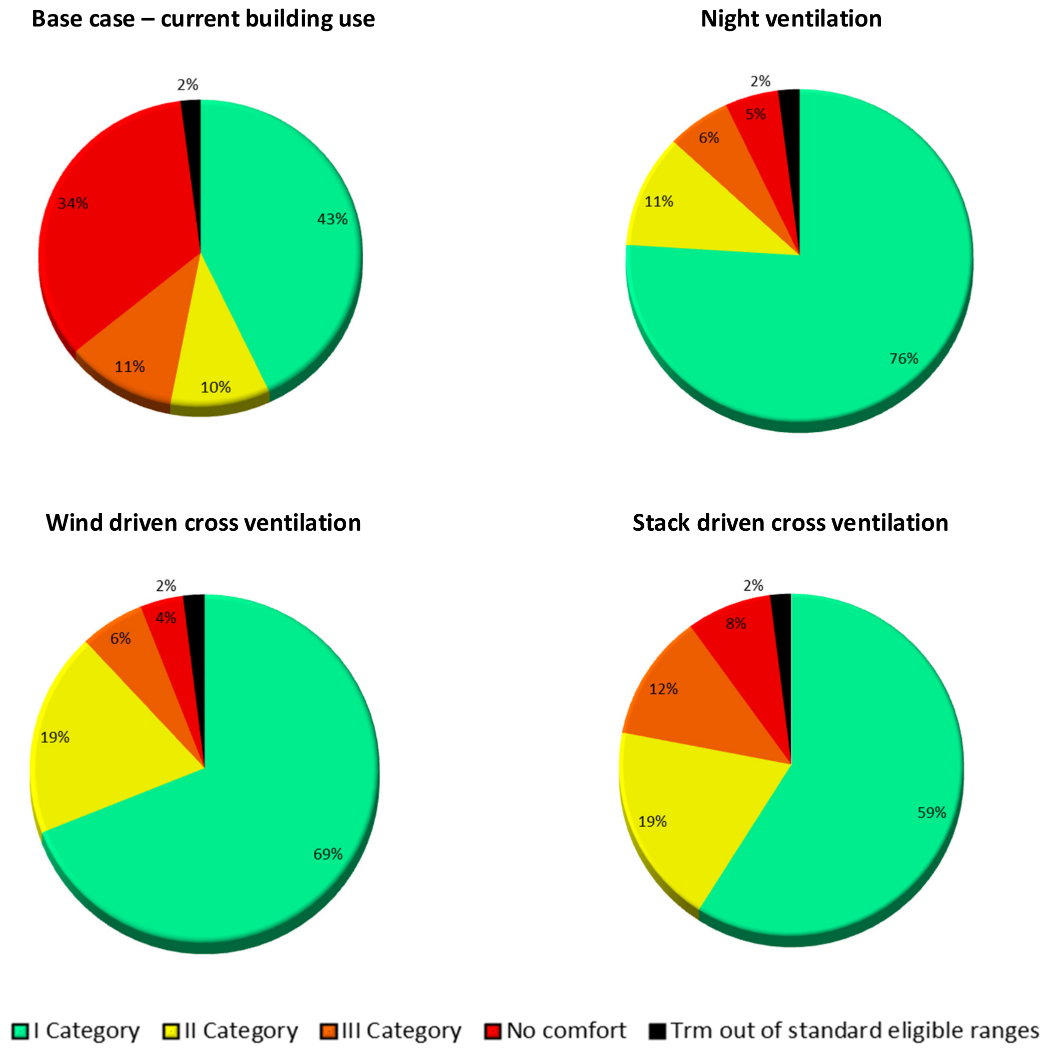

Figure 9: Comfort conditions in thermal zone considered according to UNI EN 15251 comfort standard.

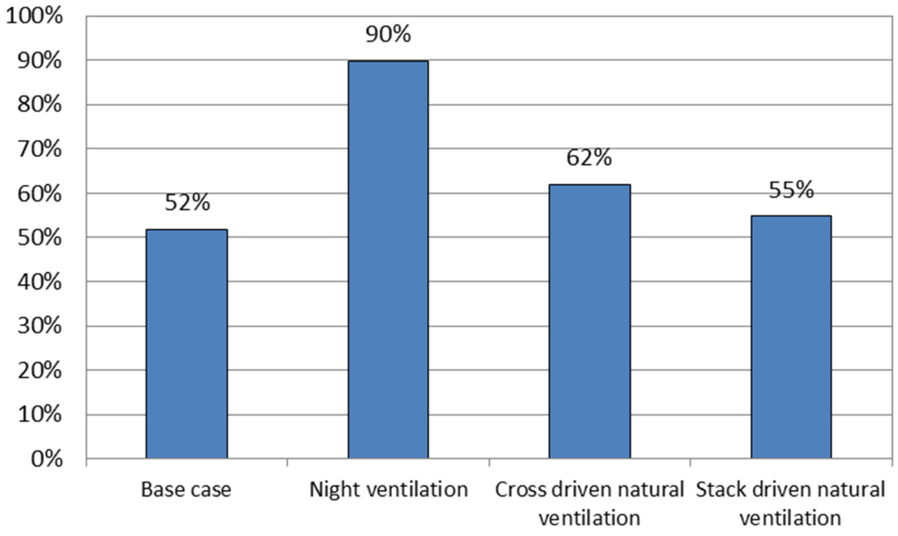

Figure 10: Mean percentage of heat (due to solar and internal loads) rejected by ventilation during the cooling season.

Conclusions

Natural ventilation seems to be a promising technique to improve thermal comfort in school buildings during the cooling season. In this work the potential of natural ventilation strategies in an existing school building has been studied.

The climate analysis showed how typical wind scenarios can be found by analysing data from a local weather station. The wind conditions at the building site are proven to rely on the dynamics of valley winds.

Three feasible natural ventilation strategies have been proposed on the basis of the climate analysis and the existing natural ventilation design guidelines: night ventilation, wind driven cross ventilation and stack driven cross ventilation.

The strategies effects on thermal comfort have been compared by means of coupled heat and airflow dynamic simulation models. The thermal comfort in classrooms has been evaluated according to the standard UNI EN 15251. Assuming a standard occupant behavior, discomfort situations occur in the 34% of occupational period hours during the spring-summer season due to overheating. The night ventilation strategy proposed allows reducing this value to 5% and also allows increasing the percentage of hours when operative temperatures are within the first comfort category boundaries. The study was completed by a survey on existing technologies, available on the market, to evaluate the practical feasibility of the proposed solutions.

It has been demonstrated that natural ventilation is an interesting low cost solution to control indoor temperatures and avoid mechanical cooling systems installation.

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Ventilated courtyard as a passive cooling strategy in the hot desert climate

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Abstract

Traditional architecture gives ideas to enrich modern architecture. In traditional architecture, local materials and renewable energy resources have been used. The courtyard was one of the traditional architecture solutions as a climate modifier. The inclusion of an internal courtyard in buildings design is attributed to the optimization of natural ventilation in order to minimize indoor overheating conditions.

The paper investigates the potential of a ventilated courtyard for passive cooling in a small building in a hot desert climate. The analyzed model is one of the low-income housing models in New Aswan City - Egypt. Which was built and provided with main services by the government, these housing models were characterized by their improper design in many cases, especially, concerning with climatic design.

To evaluate the performance of a ventilated courtyard, building simulation software TRNSYS 16 (The coupling between TRNSYS and COMIS) was used. The courtyard parameters considered were the courtyard orientation and the courtyard geometry. To evaluate the performance of a ventilated courtyard, the average monthly indoor air temperature for the purposed building determined in the overheating summer season depended on the weather data for the building site. The results of the investigations of the courtyard parameters indicate that there are some important parameters and other are of less significance which affect the thermal performance of the courtyard building model.

Keywords

Courtyard, hot desert climate, natural ventilation, low-income housing.

Introduction

Natural ventilation is one of the natural passive cooling strategies recommended for hot desert regions. In desert areas, this strategy is utilised to conserve energy while maintaining appropriate thermal comfort for inhabitants inside the building.

The aim of this study is to further the understanding and optimisation of natural ventilation cooling in buildings that have a ventilated courtyard. The performance of the courtyard was evaluated by using of TRNSYS 16 simulation tool, in a multi-story house located in New Aswan City, which is about 850 km South of Cairo, Egypt.

Brief literature review

Many researches such as Dunham, Fathy, Givony, Hinrichs, Konya, Lippsmeier, Olgyay and Saini reached the conclusion that a building with a courtyard is a desirable concept in the hot dry climate [1].

In a recent study, Al-Hemiddi describes an experiment to investigate the effect of a ventilated interior courtyard on the thermal performance of a single-family house in a hot–arid region. Statistical analysis of data recorded during the summer of 1997 was carried out. The results indicate that the courtyard gives high efficiency in providing cool indoor air through cross-ventilation [2]

In another study, a comparison between different geometries of courtyards in terms of wind flow characteristics and indoor air speed is performed based on the validation of Computational Fluid Dynamics (CFD) simulations with 2D published wind-tunnel experiments. In addition, assessment of thermal comfort is made inside a number of selected dwelling rooms facing different courtyard geometries. It is confirmed that rooms with cross ventilation have higher indoor air speed values and therefore a better thermal comfort than with single-side ventilation. The courtyard dimensions, the position of the room and the orientation are important aspects influencing the indoor air speed and thermal comfort [3]

In their research, Rajapakshaa, Nagaib and Okumiya investigate the potential of a courtyard for passive cooling in a single storey high mass building in a warm humid climate. From the results of thermal measurements, a significant correlation between wall surface temperatures and indoor air temperatures is evident. A reduction of indoor air temperature below the levels of ambient is seen as a function of heat exchange between the indoor air and high thermal mass of the building fabric. However, this behaviour is affected by indoor airflow patterns, which are controlled through the composition between envelope openings and the courtyard of the building. From a computational analysis, several airflow patterns are identified. A relatively better indoor thermal modification is seen when the courtyard acts as an air funnel discharging indoor air into the sky, than the courtyard acts as a suction zone inducing air from its sky opening. The earlier pattern is promoted when the courtyard is ventilated through openings found in the building envelope [4]

The passive cooling effects of a courtyard of a small building were determined numerically by Safarzadeh and Bahadori in their paper, employing energy-analysis software developed for that purpose. The passive cooling features considered were the shading effects of courtyard walls and two large trees (of various shapes) planted immediately next to the south wall of the building, the presence of a pool, a lawn and flowers in the yard, and the wind shading effects of the walls and trees. It was found that these features alone cannot maintain thermal comfort during the hot summer hours in Tehran, but reduce the cooling energy requirements of the building to some extent. They have an adverse effect of increasing the heating energy requirements of the building slightly. The same savings in cooling energy needs of the building can be obtained through many features such as wall and roof insulation, double-glazed windows, and special sealing tapes to reduce infiltration. They all save on heating energy requirements as well [5].

It is clear from the foregoing review that the courtyard building form, although being described as a suitable solution in hot dry climate, has not investigated with regard to the multi-storey residential building and evaluation concerning the interaction between the courtyard geometry and the orientation. Consequently, the importance and significance of the present study are apparent.

Analysis of case study

Climatic features of New Aswan City

New Aswan City is located on the west bank of the Nile, 10 Km northern of the present Aswan City and 850 km South of Cairo, so it is located in the south of Upper Egypt region which characteristic of a hot, dry climate, with a very wide difference between day and night temperatures. During the summer season, the day-by-day mean of maximum outdoor air temperature reaches 35° C in the north of Upper Egypt, and 41° C in the south. In winter season, the day-by-day mean of minimum outdoor air temperature reaches 6° C in the north of Upper Egypt, and 10° C in the south. In general, the Upper Egypt has a typical desert climate with large variations between seasons and between day and night temperatures [1].

The level of humidity in Upper Egypt is relatively low, especially during the summer months. The yearly mean of humidity in the Upper Egypt differs from 55% to 20%, north to south respectively. In New Aswan City relative humidity records the lowest value in May and June (12%), whereas reach to the highest value in December and January (36%, 34% respectively). On the other hand, The annual wind rose for New Aswan City indicates that most winds blow from north, northwest, and northeast (49.2%, 21%, and 12.9% respectively) [1].

Building description

The use of courtyards in residential buildings in Egypt, and other countries in the Middle East, is many centuries old. The courtyards provided security and privacy for the residents, and daylight for the rooms which were built around them. By building a pool, fountain and planting trees in the yard, the architects created a very pleasant space for the residents to spend a portion of their time during summer months in the yard.

In Egypt at the present time, the courtyard usually employed in buildings to collect sanitary pipes of the service's rooms (w.c, kitchen), which were built around it. Also, provide security, privacy and daylight for the residents in these rooms.

The original residential building was chosen for the study consists of three floors, and has not a courtyard as shown in Figure 1. It was constructed in 2005 within one of national housing projects was built and provided with main services by successive governments. Such a policy was adopted and applied since the fifties and is up until now with the main care devoted to produce as many residential units as possible, with less care of units' quality. Consequently, these projects were characterized by their improper design in many cases, especially, concerning with climatic design.

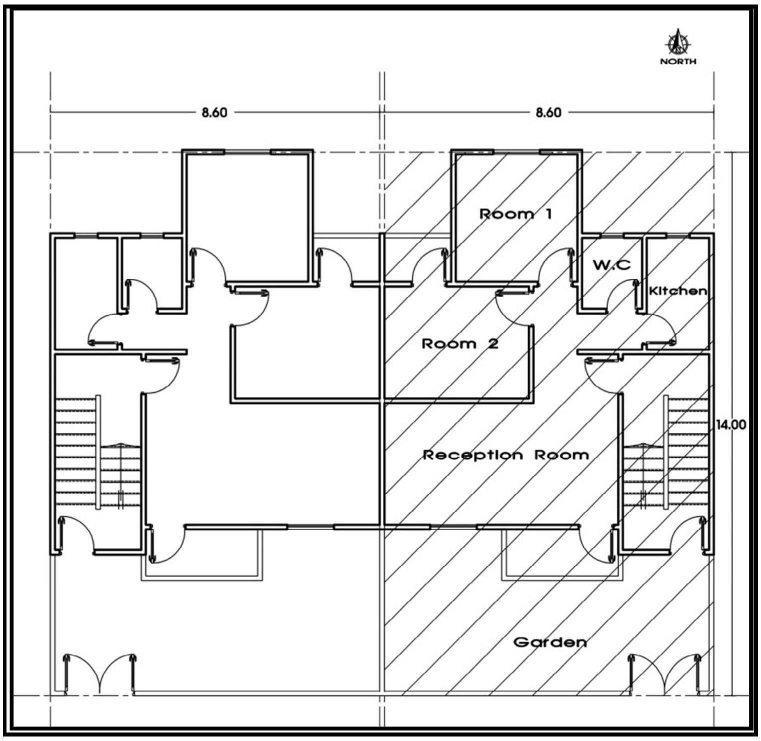


Figure 1: The original residential building

The building does not have insulation on the walls and roof. It is constructed using standard local building materials. The external walls of the rooms are constructed with single bricks, 16 cm in thickness. The roof is flat and made of a concrete slab, 12 cm in thickness. External walls are painted light brown.

In a recent study TRNSYS 16 simulation tool was used to predict the hourly indoor air temperature which shows that the results indicates that the maximum temperature obtained is 43.6° C in the ground floor, while, in the first floor the maximum temperature obtained is 43.9° C, and in the last floor the maximum temperature obtained is 44.9° C.

In addition, the predicted average monthly indoor air temperature in the summer season (as shown in Table 1) indicates the overheating conditions which exceed the comfort limits to some extent [6].

Month		Jun.	Jul.	Aug.
Temperature (deg. C)	Ground floor	36.6	37.1	37.5
	First floor	36.5	37.0	37.4
	Last floor	37.1	37.7	38.0

Table 1: The predicted average monthly indoor air temperature in the summer season

The previous results declare the need for cooling system to exceed climatic difficulties, so, the results guide to the importance of employing solutions such as courtyard in buildings to act as a climate modifier.

The modified building has a courtyard in the central area of the house. The same area and dimensions were preserved; the courtyard opened to sky, and has ventilation open, and connected with the rooms around it with windows.

In order to determine the efficiency of the courtyard, the simulation process will be carried out for the last floor which considers the worst case. On the other hand, the courtyard parameters investigated will be in different four scenarios as follow.

Case 1: rectangular courtyard with ventilation open facing the north orientation (Figure 2).

Case 2: rectangular courtyard with ventilation open facing the south orientation (Figure 3).

Case 3: square courtyard with ventilation open facing the north orientation (Figure 4).

Case 4: square courtyard with ventilation open facing the south orientation (Figure 5).

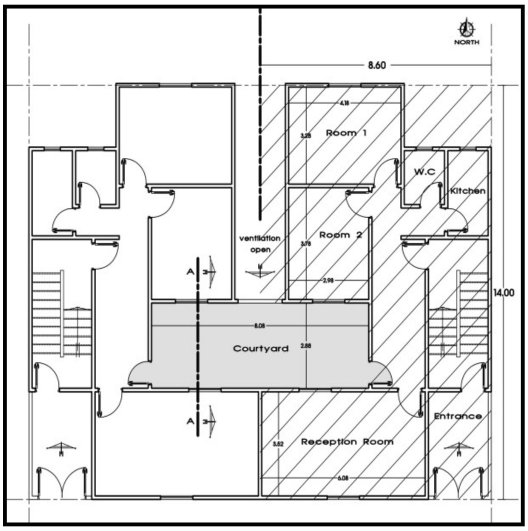


Figure 2: Rectangular courtyard with north orientation

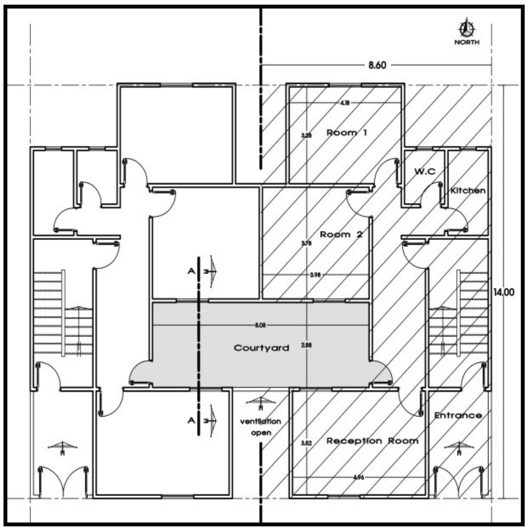


Figure 3: Rectangular courtyard with south orientation

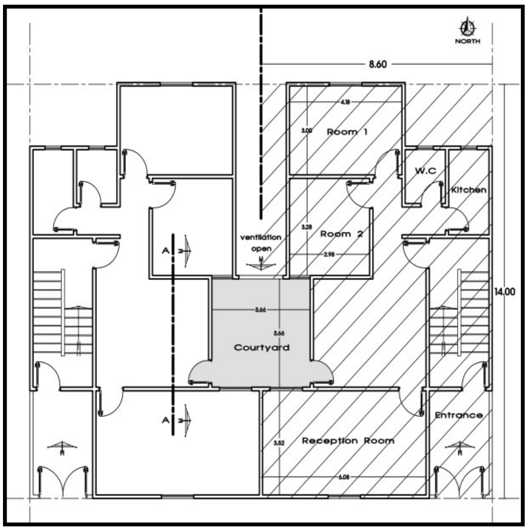


Figure 4: Square courtyard with north orientation

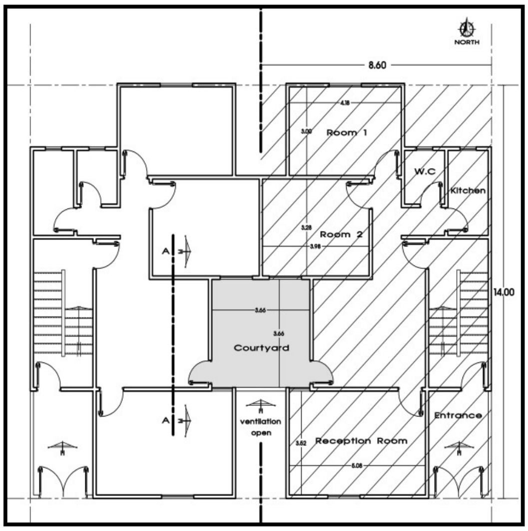


Figure 5: Square courtyard with south orientation

Figure 6 shows the cross section of the purposed building.

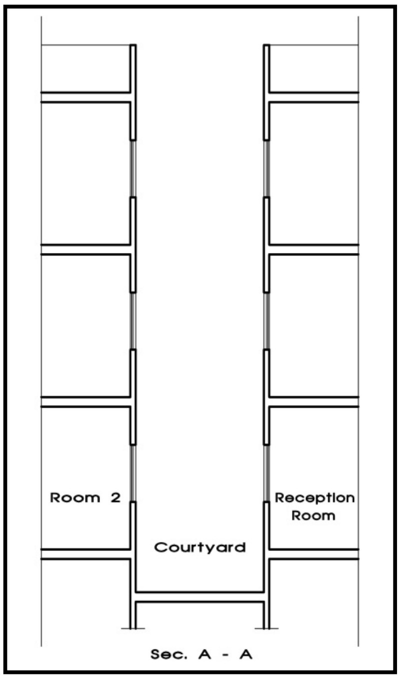


Figure 6: Cross section through the building

TRNSYS program

The present research uses the computer simulation program TRNSYS 16 (The coupling between TRNSYS and COMIS).

TRNSYS (the TRaNsient SYstem Simulation program) is a flexible tool designed to simulate the transient performance of thermal energy systems. TRNSYS can be connected to COMIS (Conjunction Of Multizone Infiltration Specialists) through use of an add-on link component called Type157. This type recasts COMIS as a TRNSYS component. In this case, the COMIS input file is generated not using a separate graphical interface but using the TRNSYS simulation studio itself [7].

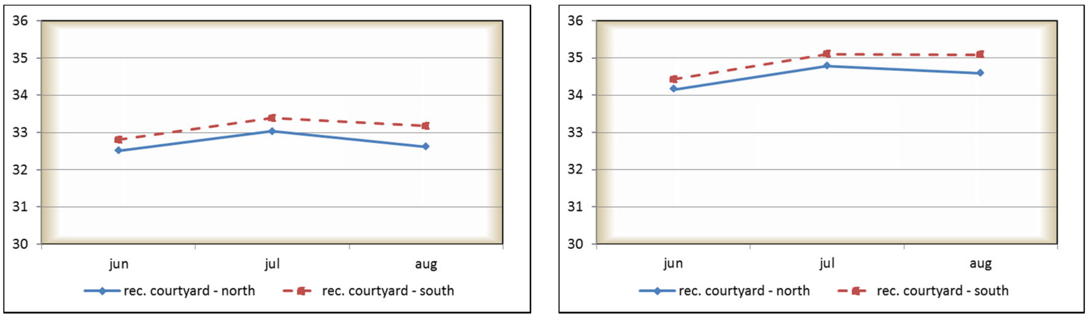
Results

Effect of the orientation

The comparing between the north and the south orientations clear that:

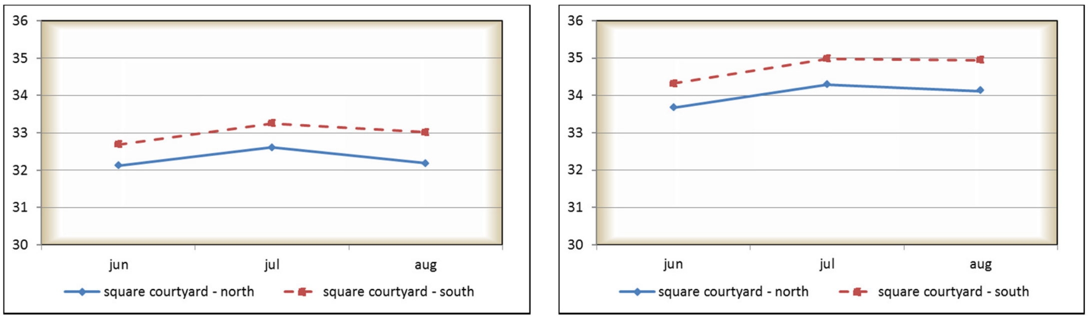
In the case of the courtyard which facing the north, the indoor air temperature recorded either in room 2 or in reception room values less than the indoor air temperature which recorded in the case of the courtyard facing the south (Figure 7). Also, the same results were recorded in the case of the square courtyard (Figure 8).

The square courtyard is more efficient than the rectangular courtyard.



Room 2 | Reception room

Figure 7: Results recorded in the case of rectangular courtyard



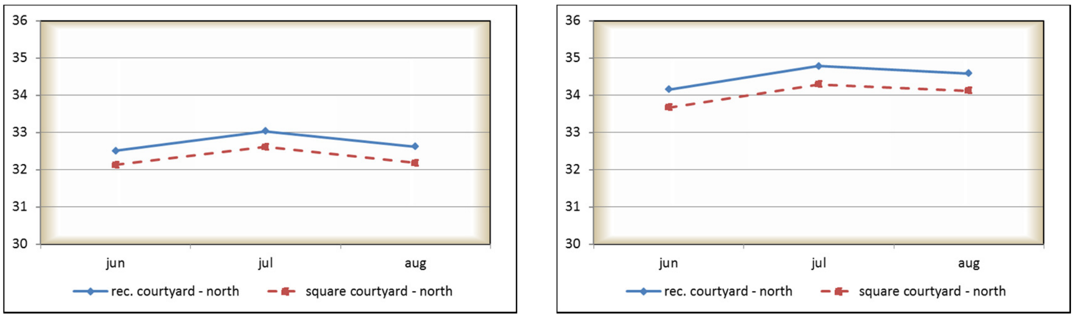
Room 2 | Reception room

Figure 8: Results recorded in the case of square courtyard

Effect of the geometry

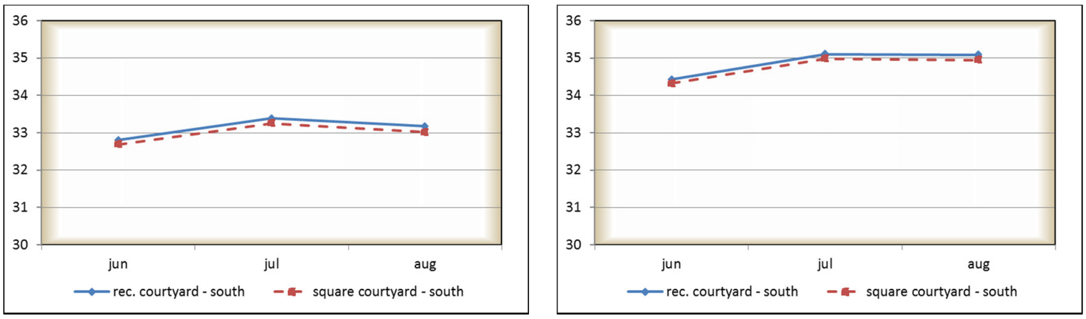
On the other hand, the comparing between the rectangular and the square courtyards indicates that the square courtyard is the best in all cases (Figure 9, Figure 10).

In the same time, the north orientation is more efficient than the south orientation.



Room 2 Reception room

Figure 9. Results recorded in the case of courtyard facing the north



Room 2 Reception room

Figure 10. Results recorded in the case of courtyard facing the south

In general, it is clear to note that employing the courtyard in the building reduce the indoor air temperature in all cases by 3°C : 5°C (as shown in Table 2).

	Indoor air temperature in room 2			Indoor air temperature in reception room		
	jun	**jul**	**aug**	**jun**	**jul**	**aug**
Building without courtyard	37.1	37.7	38.0	37.1	37.7	38.0
Rectangular courtyard facing the north	32.5	33.0	32.6	34.2	34.8	34.6

Rectangular courtyard facing the south	32.8	33.4	33.2	34.4	35.1	35.1
square courtyard facing the north	32.1	32.6	32.2	33.7	34.3	34.1
square courtyard facing the south	32.7	33.3	33.0	34.3	35.0	34.9

Table 2: The comparison between the four scenarios

The reception room records more values of indoor air temperature than the room 2.

The square courtyard which facing the north orientation is the best case correspond to the rooms around the courtyard.

In all cases, it is noted that the reduction of the indoor air temperature is not enough to achieve the thermal comfort during the summer season.

Conclusion

The results showed that it is possible to get considerable reduction in the indoor air temperature with natural ventilation by using the ventilated courtyard in hot desert climate. However to ensure a good performance designers and other professionals should pay attention to some parameters like the orientation and the geometry of the courtyard.

The courtyard is an applicable design strategy for a building from the perspective of climatic and cost–benefit analyses. These strategies can be applied to single storey or multi-storey building, also, the use of the courtyard with a pond, fountain and trees could be an applicable strategy during the hottest periods.

Finally, the development of the above formulae helps designers and architects to predict the thermal performance of courtyard buildings cooled by natural ventilation.

However, this approach only helps to predict the indoor air temperature of the purposed building, under given climatic conditions. Therefore this approach is not universally applicable, and values and judgements based upon it could change when we explore the effects of other designs and/or climatic conditions on courtyard performance.

Acknowledgements

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Towards the aeraulic characterization of roof windows ?

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Abstract

Low energy buildings, being highly insulated, are subject to important overheating risks. Thermal simulation as well as experimental studies have shown the large potential of ventilative cooling. One barrier against this approach is the difficulty of evaluating air flows. Appropriate calculation methods and characterization of openings are needed, so that these systems can be dealt with in design, regulation and certification tools.

The present study is based upon the monitoring of a 135 m2 zero energy house situated near Paris. Temperature profiles have been measured when varying the ventilation pattern, i.e. opening or closing vertical windows, roof windows and internal doors. A dynamic thermal simulation tool is used to evaluate temperature profiles in the house under the climatic conditions corresponding to on site measurements (external temperature and solar radiation). The model accounts for the conductive, radiative and convective heat transfer, as well as energy storage in the building envelope related to solar and internal gains. In this highly insulated new construction, the most uncertain parameter is the natural ventilation flow rate. This parameter, and the related aeraulic characteristics of the openings, can be calibrated by minimizing the discrepancy between calculated and measured temperature profiles. Given the small window height, and the large height between ground floor and roof windows, a one way flow model is considered.

The roof window characteristics will also be evaluated in a laboratory benchmark. A cell (3m x 3m x 2m) is divided into two compartments by a slanting wall including the window. A ventilator blows air into one space and the pressure difference is measured between both sides of the window. Varying the air flow rate allows a relationship between the flow rate and the pressure difference to be identified. This relationship may depend on the pressure difference between both sides of the openings, therefore calibration using on site

measurement is helpful. The air exchange rate estimated by this method will be compared to measurements using tracer gas, performed in the house as well as in the laboratory benchmark. The possibility to use anemometers will also be tested.

The method proposed here, combining a benchmark in a laboratory with numerical simulation and on site monitoring may bring a supplementary input, complementing the existing knowledge in the field of passive cooling of buildings. The feasibility of using this method in order to prepare appropriate input data for numerical models implemented in regulation, design and certification tools will be studied.

Keywords

Ventilative cooling, energy performance, roof windows, characterization

Introduction

Low energy buildings, being highly insulated, are subject to important overheating risks. Thermal simulation as well as experimental studies have shown the large potential of ventilative cooling [1]. One barrier against this approach is the difficulty of evaluating air flows. Appropriate calculation methods and characterization of openings are needed, so that these systems can be dealt with in design, regulation and certification tools.

The present study is based upon the monitoring of a zero energy house situated near Paris. Temperature profiles have been measured when varying the ventilation pattern, i.e. opening or closing vertical windows, roof windows and internal doors. Measurements will be compared with dynamic thermoaeraulic simulation results in order to identify an air change rate and calibrate aeraulic characteristics of the openings. These characteristics will also be evaluated in a laboratory benchmark, but may depend on the pressure difference between both sides of the openings, therefore calibration using on site measurement may be helpful. The air exchange rate estimated by this method will be compared to measurements using tracer gas, performed in the house as well as in the laboratory benchmark. The possibility to use anemometers will also be tested.

Description of the zero energy house

Architectural concept

The 130 m^2 floor area extends over one and a half storeys, with the spaces under the roof put to full use. Maison Air et Lumière, using a design principle that integrates architectural quality and energy efficiency, manages to place the emphasis on interior comfort whilst respecting the energy and environmental objectives for new detached houses for 2020. A model of the building is shown in Figure 6

Daylight

Particular attention has been paid to daylight to ensure the physical and psychological health and well-being of the residents, and to enlarge the visual perception of the indoor spaces whilst saving energy by reducing the need for artificial lighting. The amount of daylight and the quality of its distribution have been carefully studied using VELUX Daylight Visualizer 2.

Ventilation

According to the season and weather conditions, ventilation is provided by a hybrid system that combines the advantages of mechanical ventilation with heat recovery in winter and, in summer, natural ventilation by window opening (supplemented by mechanical extraction in bathroom and kitchen).

Energy design

The energy concept of Maison Air et Lumière is based on the maximum use of renewable resources (solar energy, natural light, passive cooling) in order to minimise the need for air conditioning in summer, to reduce heating in winter and to reduce artificial lighting and energy use for ventilation. The combination means a neutral environmental impact and maximum comfort for the residents. The house, which is built on a concrete slab on an earth platform insulated on the underside, is constructed with a well-insulated wooden frame and with a window-floor ratio of nearly 1:3. All windows are equipped with dynamic solar protection and the operation of all systems in the building (heating, ventilation, shading window-opening, lighting etc) is fully automated.

With its interplay of roof structures, the building is compact and very well insulated and, in order to create a stable and comfortable room temperature, the interior walls are lined with terracotta tiles, appreciably improving the thermal mass of the building. Heating and hot water are provided by a heat pump connected to thermal solar panels and a low-temperature underfloor heating system. The artificial lighting, domestic appliances and multimedia equipment were selected on the basis of their low consumption. Moreover, to reduce electricity consumption further, the washing machine and dishwasher can be directly connected to a cold and hot water inlet. All electric power consumption will be offset by the contribution from 35 m^2 of photovoltaic panels integrated in the roof. In normal use of the building, the overall annual energy balance is positive.

Experimental protocol and description of the monitoring

Measurements in a laboratory facility

A benchmark is installed at the CEP laboratory in order to identify the characteristics of a roof window. A ventilator is used to create a pressure difference in a test cell divided in two compartments. The roof window is installed on a 45° sloped wall between these compartments. The pressure difference (P_2-P_1) is measured, as well as the air flow rate Q: a diaphragm and Pitot tubes are used in the inlet air pipe in order to get a reference value of the flow rate (Figure 1).

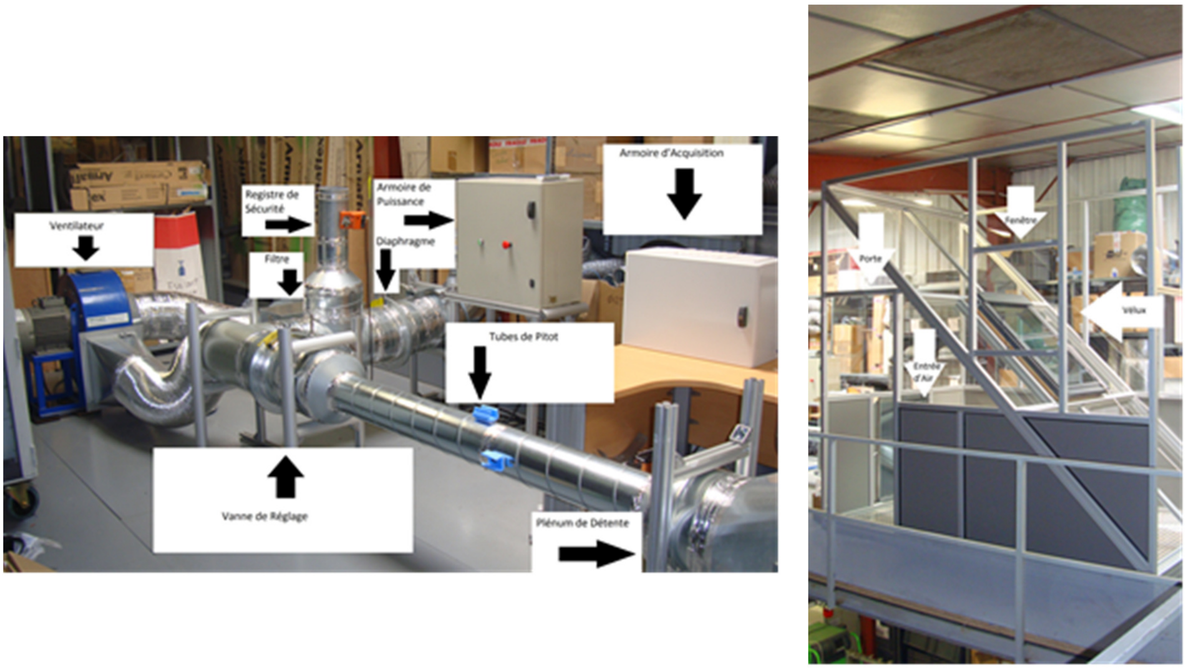


Figure 3: Air inlet of the laboratory benchmark, including flow rate measurement

This reference value will be compared to the air flow rate obtained from analysing CO_2 concentration profile (tracer gas method). First a profile will be measured with the window closed in order to identify the air infiltration flow rate. Then the window will be open and the additional flow rate will be derived by difference.

A one way flow is assumed in the conditions of this experiment. The ventilation flow rate and therefore the pressure difference will be varied in order to draw a curve relating the flow rate Q to the section S and pressure difference, and to derive characteristics of the roof window Cd and n [2]:

$Q = S.Cd\,(P_2 - P_1)^n \qquad (1)$

The section S considered is the geometrical opening section. The difference $P_2 - P_1$ will be in the range from 0.05 to 1 Pa. In the real house, it may be lower, depending on wind conditions, but it is hoped that the values of Cd and n will not vary too much. The laboratory test will therefore provide two parameters that can be used in the analysis of on site measurements, and that can be refined using a calibration step.

Anemometers will be used in order to study the possibility to measure the air speed at different locations of the opening, and to derive the flow rate. A CFD model of the benchmark has been developed [3] in order to know if some position of the anemometers leads to a more precise evaluation of the flow rate, see Figure 2. The velocity is assumed uniform on the inlet section (0.1 m/s), which corresponds in the experiment to the use of a honeycomb structure.

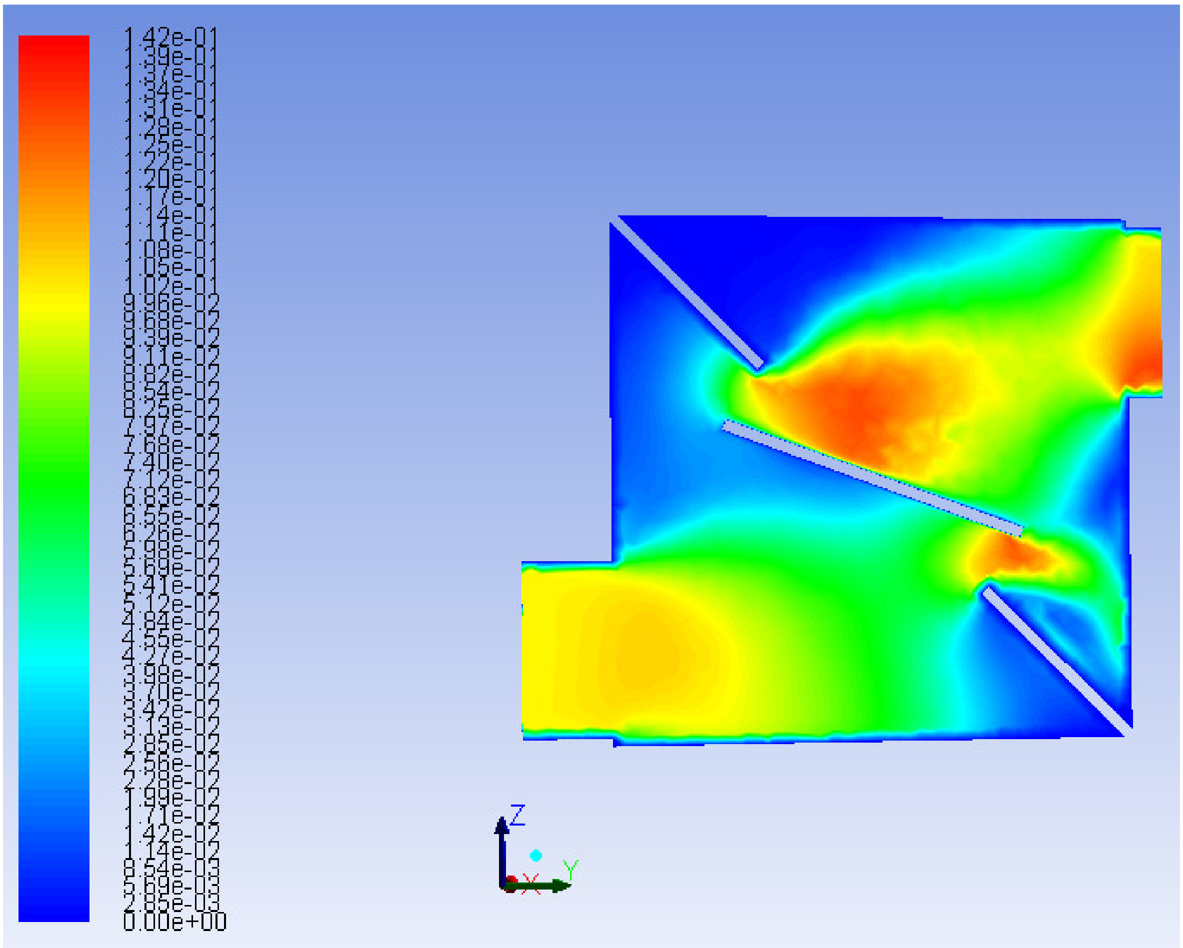


Figure 2: Example CFD model of the bench mark and air velocity results

CFD can also be helpful to choose the position of pressure gauges, see Figure 3.

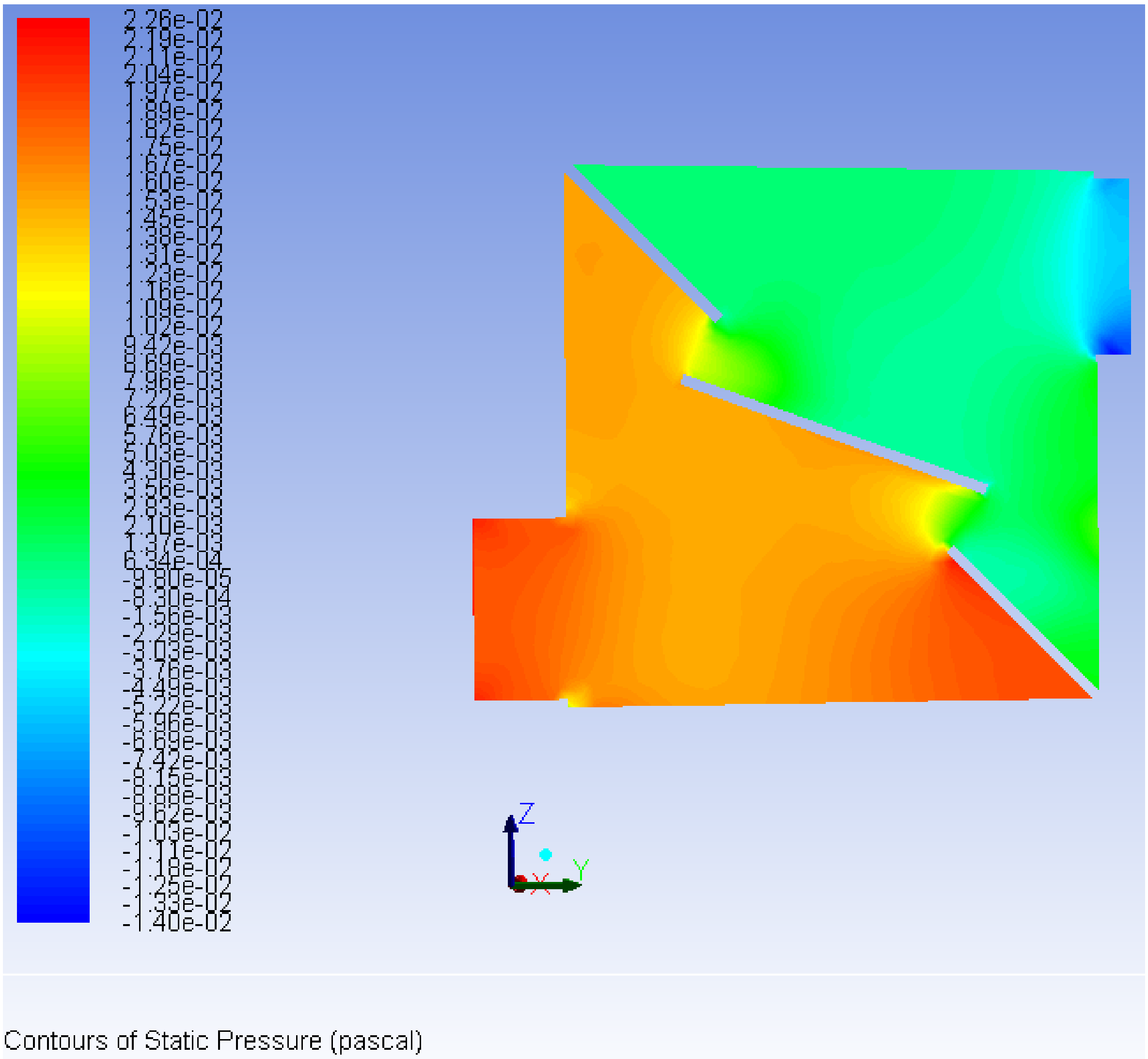


Figure 3: Example CFD results, Static pressure in the benchmark

Monitoring of the house

The measurements have been performed from 20 July to 20 August 2012, according to four scenarios successively:

- Without natural ventilation (all windows are closed), in order to obtain a reference,

- With natural ventilation (all roof windows and top vertical windows are open), without movable shading and with internal doors open (to get the maximal effect of natural ventilation),
- With natural ventilation, with movable shading and internal doors closed (to get the minimal effect of natural ventilation),
- With natural ventilation, with controlled movable shading and internal doors closed (to get the more realistic effect of natural ventilation).

Figure 4 shows temperature profiles in different rooms without ventilation (first period), with uncontrolled ventilation (second period, doors open), and with controlled ventilation (fourth period).

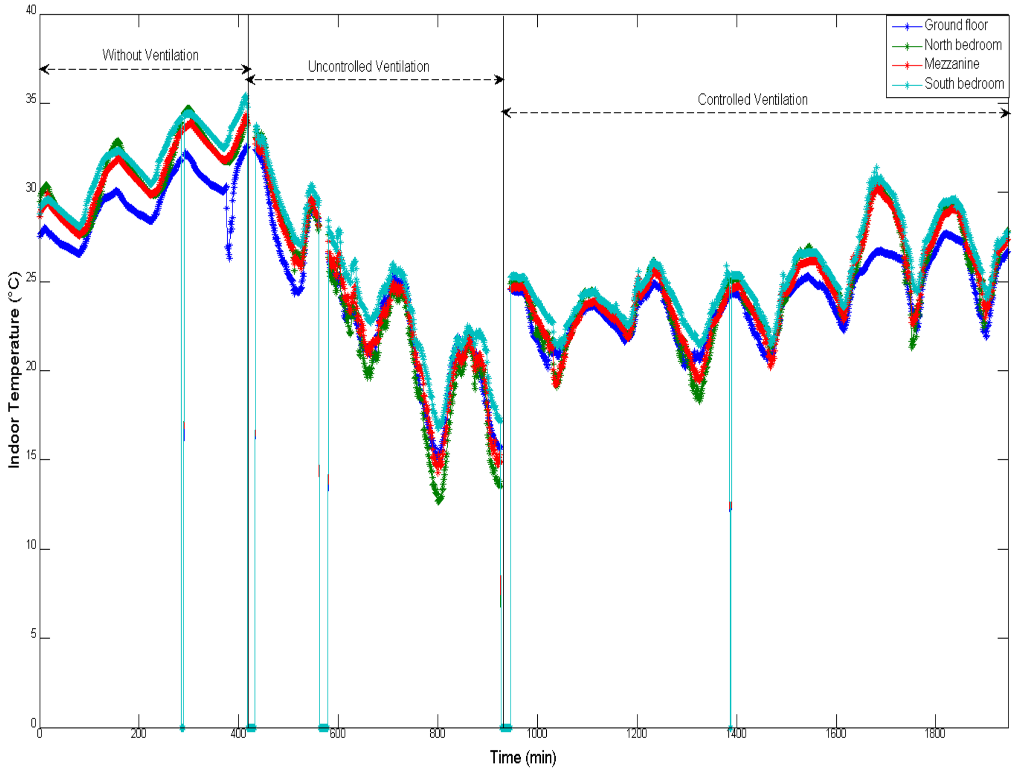


Figure 4: Example measured temperature profiles in Maison Air et Lumière

Pressure differences between outside and inside will be measured at certain times (not continuously) through windows situated on the different facades. Wind velocity and direction are also measured, so that pressure coefficients can be derived, allowing pressure on the different facades to be estimated over the whole period.

Given the small window height, and the large height between ground floor and roof windows, a one way flow model is considered. Discharge coefficients Cdi and exponents n, identified for roof windows using the laboratory benchmark presented above, or collected in the literature for other windows, allow the air flow rates through the different windows to be evaluated in terms of the internal pressure. The sum of inlet and outlet flow rates being zero, this internal pressure can be derived. An air exchange rate can then be evaluated.

The global air exchange rate can also be evaluated using a tracer gas method, which provides a second estimation of this parameter, but the precision is also questionable as it will be addressed in the discussion §. In a homogeneous zone (volume V), the internal concentration Cint depends on the emission from the source So, the fresh air flow rate Q, internal and external air densities ρint and ρext, and external concentration Cext [4]:

$$\rho int\ V\ dCint/dt = So - \rho ext\ Q\ (Cint - Cext) \qquad (2)$$

In a steady state, the concentration is constant (dCint/dt = 0) so that:

$$So = \rho ext\ Q\ (Cint - Cext) \qquad (3)$$

Due to the high flow rate in the house when opening all roof windows, using the steady state option would require a large quantity of gas. It seems therefore preferable to inject a certain quantity of gas and to measure the decrease of gas concentration. Equation (2) allows the air flow rate to be identified.

Example measurement results are shown in Figure 5 for two configurations: internal doors being closed or open. The logarithm of the concentration has been derived, so that the air flow rate Q can be identified using a least square method. In the example below, the result is around 4.5 air change per hour (ach) when the doors are closed and 5.5 ach doors open.

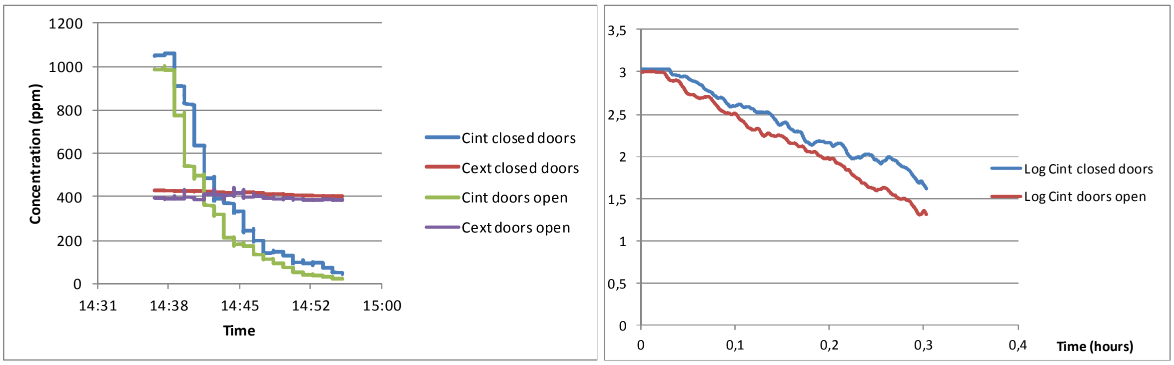
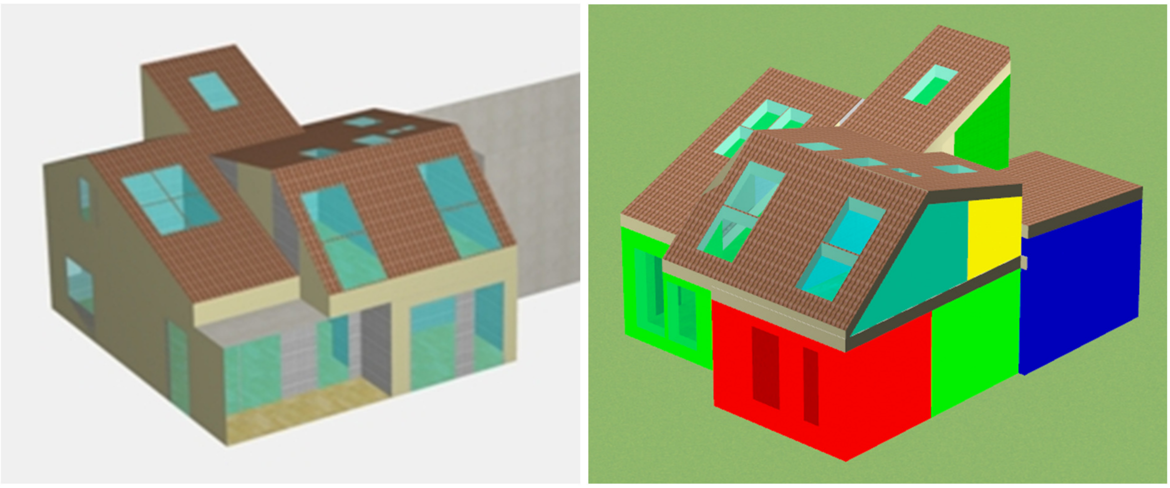


Figure 5. Example tracer gas concentration measurements in Maison Air et Lumière

Dynamic thermoaeraulic simulation

Complementing monitoring results, numerical simulation constitutes another way to better understand the behaviour of a building. A dynamic thermal simulation tool is used to evaluate temperature profiles in the house [5], which has been modelled using 7 thermal zones : the living space (on two levels), three bedrooms (with different orientations), a garage, and other rooms (ground floor and first floor), see Figure 6.



3D model Thermal zones

Figure 6 : Thermal model of Maison Air et Lumière, graphic modeler ALCYONE

The model accounts for the conductive, radiative and convective heat transfer, as well as energy storage in the building envelope related to solar and internal gains. In this highly insulated new construction, the most uncertain parameter is the natural ventilation flow rate. Figure 7 shows simulation results for a typical summer week in Greater Paris Area, considering three levels of ventilation flow rate (no ventilation, 5 ach and 20 ach).

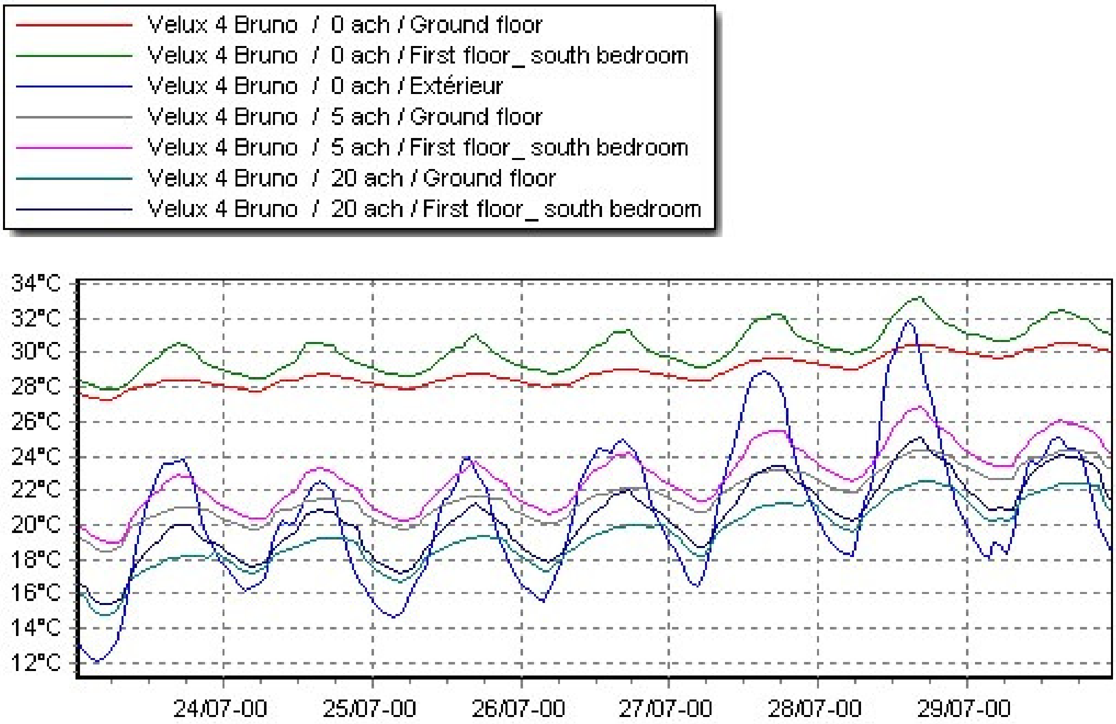


Figure 7. : Example simulation results of Maison Air et Lumière, dynamic thermal simulation tool COMFIE

According to these results, this parameter has a large influence on the temperature profiles. It can therefore be calibrated by minimizing the discrepancy between measured temperature profiles and simulation results [6] using the climatic data corresponding to on site measurements (external temperature and solar radiation). This constitutes a third way to evaluate the global air exchange rate, and may be helpful to refine the characteristics evaluated in the laboratory benchmark, by taking into account the actual conditions in the real house.

Another added value of numerical simulation is, once the model has been calibrated, to compare, under the same climatic conditions, temperature profiles with and without ventilative cooling in order to evaluate the benefit of this approach. Such evaluation would otherwise require the construction of two identical houses, which is technically difficult and of course expensive.

Discussion, study of a characterization method

Evaluating the interest of ventilative cooling requires simulation methods with at least hourly time steps because monthly or annual methods are not able to evaluate temperature profiles. Such methods need input data regarding the aeraulic properties of openings. The most common models for one way flows consider two characteristics: a discharge coefficient and an exponent, relating the air flow rate to the pressure difference on both sides of the opening (see equation 1).

As seen in § 2, using on site monitoring in a real house to determine these parameters is very difficult due to very low pressure differences on both sides of windows, therefore measurable with a high uncertainty. Air flow rates are also difficult to measure: velocities can be measured, but a flow rate is an integration of these velocities over the opening section. Anemometers can be used to measure velocities, but they have to be placed in specific points in order to obtain an average value through the whole section. The precision of such measurement is then very low. If both the pressure difference and the flow rate are not precisely known, it is very difficult to derive the two parameters Cd and n of equation (1).

A benchmark in a laboratory has two advantages:

- the flow rate can be higher than in the real building, so that the pressure difference is higher and therefore easier to measure;
- the flow rate can be measured using e.g. a Pitot tube, constituting a reference value.

The exponent n of equation (1) may be somewhat different in real conditions, because it depends on the pressure difference. On site measurement is therefore needed to calibrate this parameter. At the moment the tracer gas method is used in practice, but there is a large uncertainty due to possible uncomplete mixing following the density difference between the air and the most common gases (CO_2, SF_6, N_2O). These gases being heavier, they accumulate in the lower part of the building so that their concentration is not varying as modelled in equation (2). The measured air change rate may therefore be underestimated. Other gases like some VOCs might be used in the future, provided that they are not emitted by building elements (e.g. painting, glue etc.).

Calibrated window characteristics allow the ventilative cooling potential in a building to be evaluated, using thermoaeraulic simulation. This procedure may be validated thanks to a comparison between calculated and measured temperature profiles. Calibration of the simulation model using the measured temperature profiles may also constitute an alternative to using tracer gases: measuring temperatures is simpler and cheaper than measuring gas concentration.

Conclusions

The method proposed here, combining a benchmark in a laboratory with numerical simulation and on site monitoring may bring a supplementary input, complementing the existing knowledge in the field of passive cooling of buildings. The feasibility of using this method in order to prepare appropriate input data for numerical models implemented in regulation, design and certification tools will be studied.

Acknowledgements

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Reducing energy consumption in an existing shopping centre using natural ventilation

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Abstract

The energy consumption needed for establishing a good indoor climate in shopping centres is often very high due to high internal heat loads from lighting and equipment and from a high people density at certain time intervals. This heat surplus result in a need for cooling during most of the year, typically also during the winter, and often the needed cooling is provided by a mechanical ventilation system with integrated mechanical cooling.

However, for certain areas, especially the hallways connecting the individual shops, natural ventilation might be an energy efficient alternative or supplement to the traditional mechanical system.

This paper presents a case study on an existing shopping centre in Copenhagen, Denmark (Fields shopping centre). The building owner's key reason for considering natural ventilation was a desire to improve the thermal indoor climate in the hallways, and, at the same time, reduce the energy consumption for ventilation.

On this background, WindowMaster conducted a number of simulations in the dynamic simulation program BSim2002. These calculations suggested a significant energy saving potential (60% reduction) and a significant improved thermal indoor climate (70% reduction of annual hours above 28 °C) by adding natural ventilation to the ventilation strategy.

Thus, in the beginning of 2011 the building owner decided to install automatically controlled natural ventilation in the hallways in the shopping centre in addition to the existing mechanical ventilation system. The basic control idea was to use the natural ventilation system in the summer and transient seasons and the mechanical ventilation system in the winter (hybrid ventilation).

Measurements of the thermal indoor climate in the first year (September 2011 - august 2012) show that the indoor climate has improved significantly. In this year, the actual results

outperform the expected results from the simulations, and the building owner has expressed his satisfaction with the improvements in the thermal indoor climate.

Keywords

Thermal building simulation, energy savings, natural ventilation, hybrid ventilation, natural cooling, shopping centre, case study, log-data analysis.

Introduction

As of today, only few shopping centres have adopted natural ventilation in the hallways. This might be caused by a limited number of reference cases thus creating uncertainty for the building owner about the potential benefits in terms of reduced CO_2 emissions, running costs and improved indoor climate.

It has however been demonstrated in a number of practical cases e.g. Ernst-August-Galerie in Germany [1] and Green Light House in Denmark [1] that the use of natural ventilation in combination with mechanical ventilation might give significant improvements in the thermal indoor climate and the energy consumption in comparison with pure mechanical ventilation. Both cases have a DGNB certificate showing high performance on ecological quality and indoor climate.

In general, natural ventilation is preferable to mechanical ventilation in multi-storey rooms as the heat and stale air generated in the air volume is easily expelled through automatically opened roof windows. Such rooms exist not only in the hallways of shopping centres, but also in many other buildings, like atriums in office buildings, hallways in airports, exhibition halls, production halls and in sport facilities. Thus, the results and principles given in this paper are useful in a number of building types.

The Fields shopping centre, see Figure 1was completed in 2004 and it is as of today the biggest shopping centre in Denmark and one of the largest in Scandinavia. It is located in Ørestad, Copenhagen and the total size of the centre is 115,000 m^2 with a total shopping area of 65,000 m^2. The centre contains more than 140 retailers and it is the workplace for around 2,500 employees. After the completion, the building owners have been committed to implementing new technologies for reducing the building energy consumption and increasing the shopping experience of the customers.

Originally the centre was equipped with a full mechanical ventilation system. However, as explained in the Abstract, in the beginning of 2011 it was decided to install automatically controlled natural ventilation in the hallways in the shopping centre to support the mechanical ventilation (hybrid ventilation). The purpose was twofold – in order to reduce energy consumption and to improve the thermal indoor climate for the benefit of customers and employees.

Automatically controlled natural ventilation is controlled depending on variations in the outdoor conditions measured from a weather station and the indoor climate measured from for example temperature/CO_2/humidity sensors and information about the usage of the building. Automatically controlled natural ventilation is highly advanced ventilation and should not be confused with manually controlled natural ventilation.

This paper presents the dynamic building simulations which were used as basis for the decision to implement natural ventilation in the building, together an analysis of the log-data of the indoor climate obtained during the first year of service. Results for the thermal indoor climate and energy consumption for ventilation are presented.

Figure 1: Picture of the shopping centre

Building and ventilation principles

The roof above the hallways is made of glazed sections as illustrated in Figure 2 and Figure 3. In the glazed areas automatically controlled window openings are established. Through these openings air enters and leaves the hallways. Based on WindowMaster's experience, the hallways have been partitioned into eight zones in the simulations – divisions which are later followed in the design of the system implemented in the building.

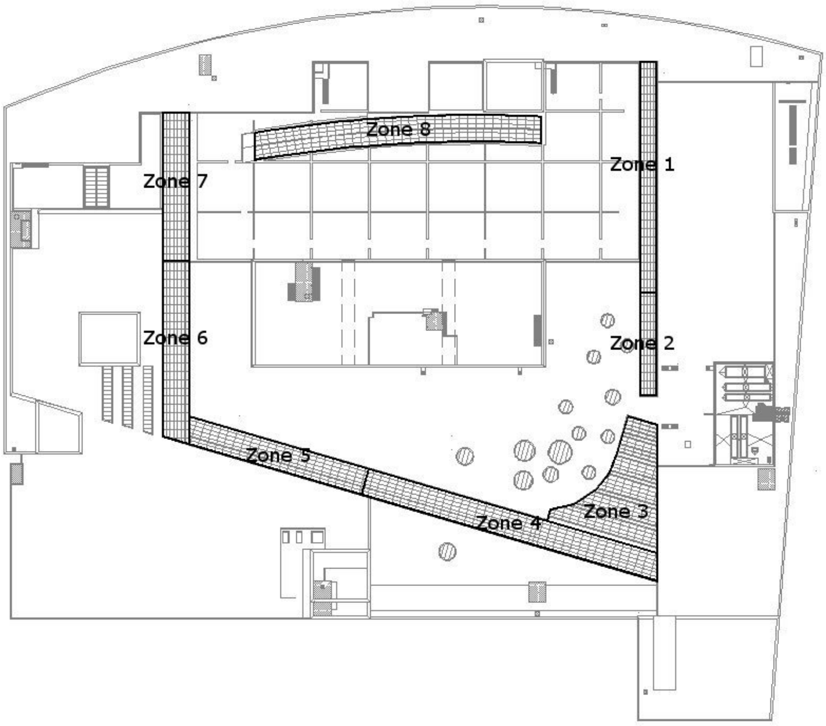

Figure 2:.Drawing of the shopping centre roof including marking of the eight zones.

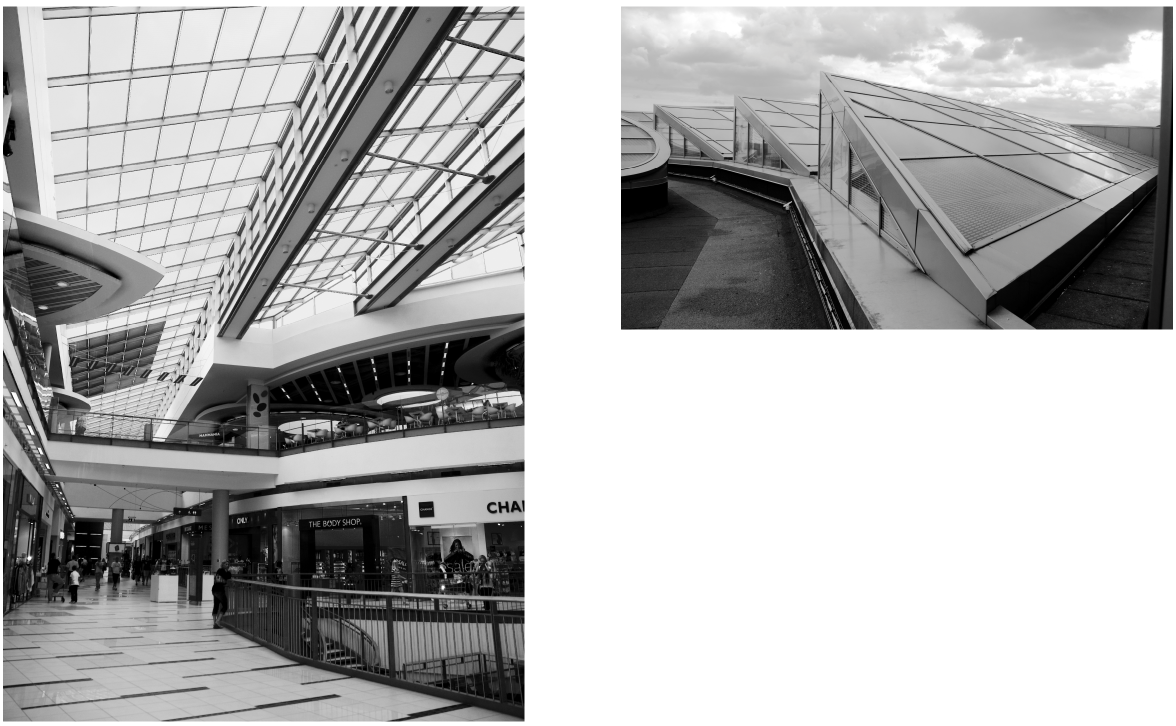

Figure 3: Picture of hallways from the inside and from the roof.

Natural ventilation uses the natural driving forces (wind and thermal buoyancy) for moving air and is therefore able to ventilate without any energy consumption for air movement (SFP=0). Therefore, in situations where the room or building in consideration needs cooling, natural ventilation is able to supply fresh air without energy consumption. This is an advantage for the shopping centre since warm air can be removed, while at the same time the energy consumption for air movement is reduced.

In Figure 4 the natural ventilation principle for the hallways is illustrated.

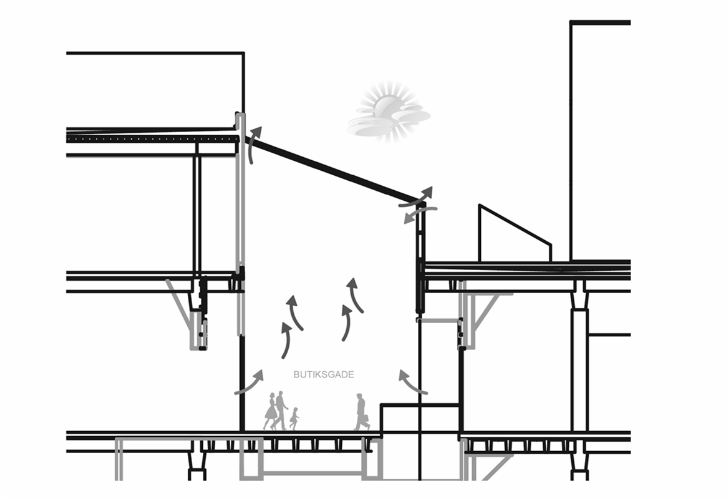

Figure 4: Ventilation principle for the natural ventilation system in the shopping centre

The primarily ventilation principle is stack-ventilation. This type of ventilation is primarily driven by warm air rising to the top whereby it creates a pressure difference which drives the ventilation.

Most of the opening area for natural ventilation is established in the glass roof above the hallways - only about 2% of the opening area is established in the facade. This distribution is not optimal, but caused by the fact that the natural ventilation system is retrofitted to an existing building. For a new shopping centre, where natural ventilation is included from the early design phase, it is recommended that the opening area is more evenly distributed between facades and roofs in order to achieve an optimal placement of the neutral pressure level. Nevertheless, the natural ventilation system capacity in the Fields shopping centre is still high as openings are placed in different levels in the roof and since it was possible to place some openings in the facade.

Simulation techniques and assumptions

The method used is thermal building simulation of the hallways made in the dynamic simulation program BSim2002 [3]. The software simulates the thermal indoor environment and energy performance during a year based on the specific constructions, usage and the outdoor weather conditions. Weather data from Denmark is used (DRY [4]). Based on steady state calculations, made prior to the dynamic simulation, the capacity of the natural ventilation system is determined. The methods are described in [5] and [6].

The natural ventilation system modelled is demand controlled, as the heat loads from especially people and sun changes a lot during the opening hours.

Figure 5 shows an illustration of the 3D section of the hallways which was modelled in the simulation software. This section is representative for all the hallways since these were sufficiently similar with regard to their design, internal heat loads and solar gain.

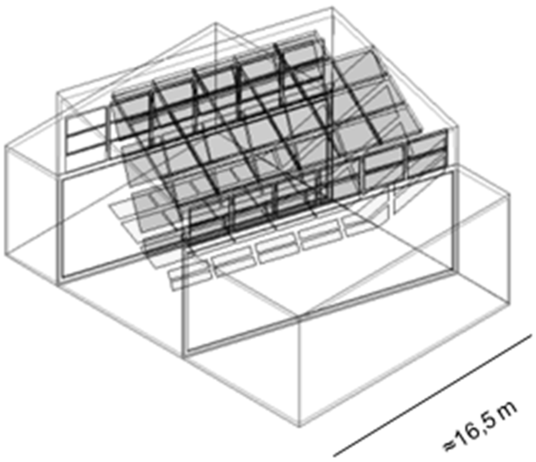

Figure 5: Calculation model for the BSim2002 simulations

Three different ventilation strategies were modelled to investigate the thermal indoor environment and the energy saving potential:

Model 1: Mechanical Ventilation (MV)

- In this model, the already existing system with mechanical ventilation was modelled in detail
- No natural ventilation system was added

- In the calculations it is assumed that the mechanical ventilation system is a CAV system with a ventilation rate of 4 h^{-1} and a heat recovery coefficient of 0.7
- The inlet air is at least 17°C
- The mechanical ventilation system is turned off during the night

Model 2: MV in winter, NV in summer (hybrid ventilation)

- In the second model, the mechanical natural ventilation was used in the winter only (week 1-15 and 46-52, in total 22 weeks of the year)
- A new natural ventilation system was implemented in the transient season and during the summer time (week 16-45, in total 30 weeks of the year)
- The natural ventilation system is set to have a maximum capacity of 4 h^{-1} and it is assumed that the natural ventilation is automatically controlled and utilized for both day ventilation and night cooling.

Model 3: MV all year, NV in summer (hybrid ventilation)

- The third model allowed the natural ventilation and mechanical ventilation to run at the same time in the summer (same weeks as given above)
- The purpose is to further reduce the number of warm hours in the hallways
- During summer mode the mechanical ventilation system is activated when the operative indoor temperature reach 26°C

For all three calculations the opening hours is Monday to Friday 10AM - 8PM and Saturday - Sunday from 10AM to 5PM. The people load is 5 m^2/person and the usage level is 100 % Monday - Friday from 2PM - 6PM and 50% in the rest of the time on weekdays. Saturday and Sunday the usage level is set to 90 % from 10AM - 5PM. The lightning level is set to 30W/m^2 in the opening hours. The g-value of the glass in the roof is 0.3.

Simulation results

Figure 6 shows the results of the thermal building simulations for the three models. The figure shows the total number of hours in the hallways for which the operative roomtemperature is above 28°C, 30°C and 32°C respectively. Note that only hours within the opening hours of the shopping centre have been counted. It is clearly demonstrated that usage of a natural ventilation system in the summer and transient seasons will reduce the number of "warm hours" significantly – from 677 hours to 182 hours above 28°C. It is noted that the results after installing natural ventilation would have been even better if the system had been implemented in the building from the early design phase.

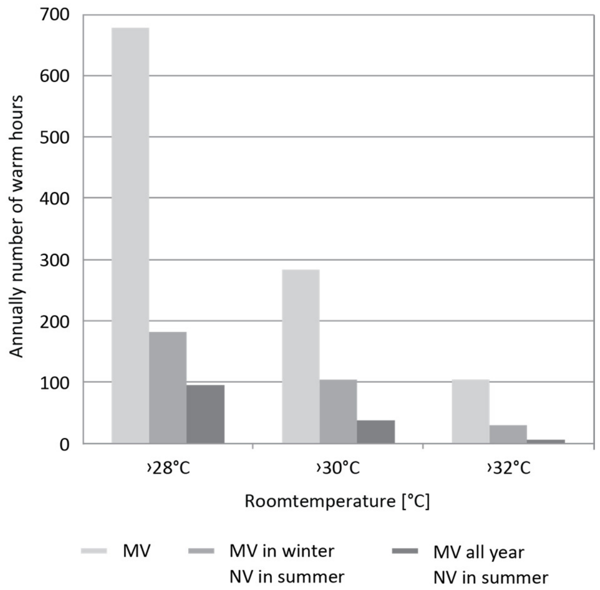

Figure 6: Annual number of hours with room temperature above 28, 30 and 32 °C.

Based on the simulations, calculation of the electrical energy consumption for air transport has also been made, see Figure 7. Energy consumption for heating is not included as the natural ventilation system is only used when there is a need for cooling (week 16-45). It is seen that the introduction of natural ventilation has a potential for reducing the energy consumption for ventilation by almost 60% (~ 40 kWh/m^2 year). To achieve these savings in practice, this requires that the mechanical ventilation system can be switched off – this is therefore an important design criterion for hybrid ventilated buildings.

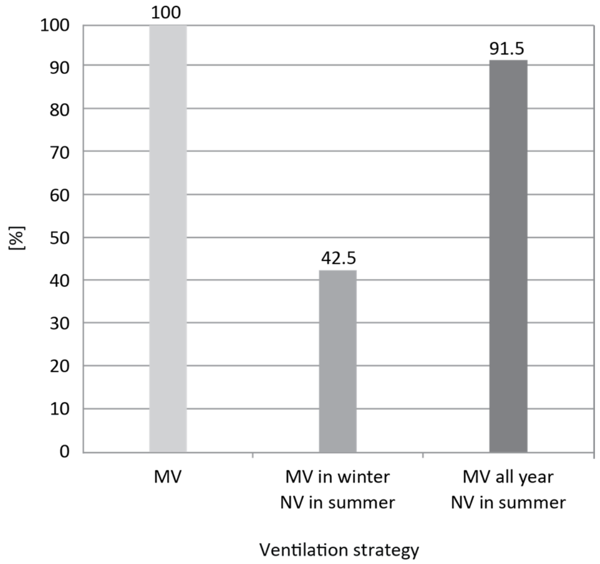

Figure 7: Energy consumption for air transport (indexed).

The results also show that in Model 3, where both natural and mechanical ventilation is used during the summer an additional reduction of 87 hours above 28°C could be achieved. This strategy does of course increase the energy consumption for the mechanical ventilation system significantly, but an 8.5% reduction compared with the pure MV system is still achieved. This is due to the fact that it is possible to turn off the mechanical system in selected periods.

The results suggest that there is a significant potential for reducing energy consumption in hallways of shopping centres by the use of natural ventilation. It is noted that the savings would have been even higher if the natural ventilation system had been designed into the building from the early design phase.

Measurements

In reality the hallways are split into eight indoor climate zones. In each zone four or five combined temperature and CO_2 sensors are placed. These sensors form the basis for calculation of an average value for the temperature and the CO_2 level in the actual zone. The values are used for regulating the natural ventilation system and deciding whether the mechanical system should be turned on or off.

Figure 8 shows an example of a sensor placement.

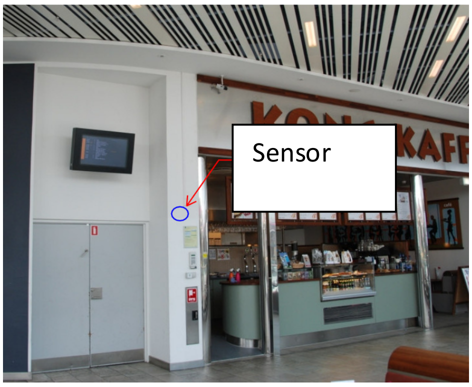

Figure 8: Example of placement of temperature/CO2 sensor and detail showing the sensor.

Figure 9 shows actual measured temperatures in four representative hallways for one of the warmest summer weeks in Denmark, 2012 (week number 33). In this week, the hallways were ventilated using both the natural and the mechanical ventilation system (hybrid ventilation).

Figure 9 clearly shows that the natural ventilation system is capable of conditioning the thermal indoor climate in the hallways during hot periods in the year. The local outside temperature (measured on the roof) in the end of the week is close to 35 °C while the indoor temperature, despite the high heat loads, is kept below 28 °C in general.

Note also that the room temperature experienced by the occupants will feel lower than the calculated temperature due to the air speed which it is possible to establish when natural ventilation is used. The air movement may reduce the experienced temperature with up to 2-3°C. This additional cooling effect is beneficial during hot periods and will be experienced as a cool breeze by the occupants.

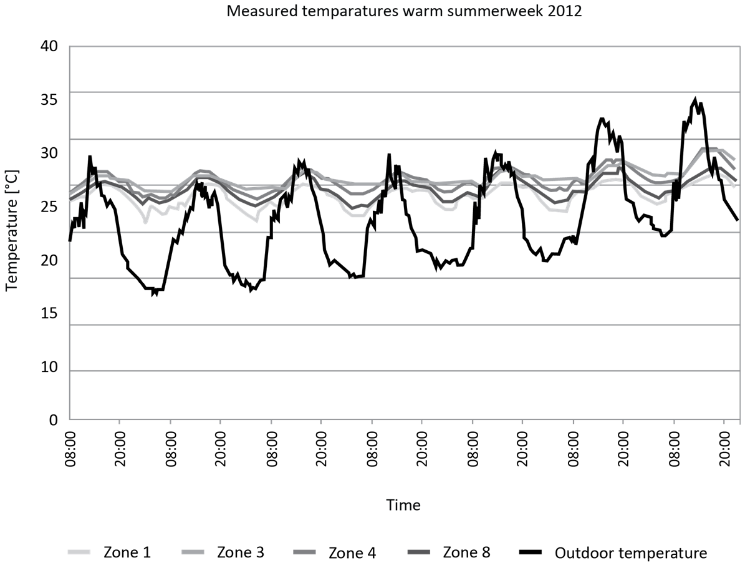

Figure 9: Measured temparatures in four representative hallways in a warm summerweek in august 2012.

Figure 10 shows the number of warm hours for one of the hallways - zone 3. The temperature has been logged for the period 01.09.2011 - 31.08.2012 – i.e. a whole year. It is noted that zone 3 is one of the warmest areas - which is also indicated in Figure 9. During this period, the system is ventilated using both the natural and mechanical ventilation system. Note, that Figure 10 compared with Figure 6 use a different temperature interval for the bars. This has been chosen since there are no temperatures above 30°C in the building – in fact the temperatures are almost entirely below 28°C.

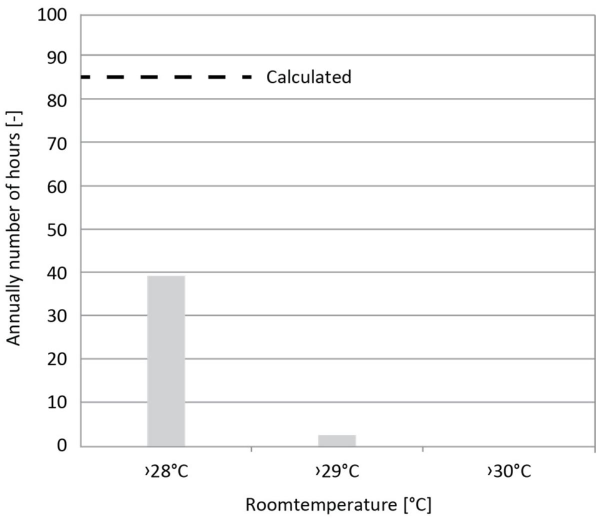

Figure 10: Annually number of hours above 28, 29 and 30 °C for the period 01.09.2011 - 31.08.2012 in hallway 2.

The number of hours with room temperatures above 28°C is approximately 40 for the year 2011 - 2012. In the dynamic simulation the annual number of hours was approximately 85 for simulation model 3 where mechanical ventilation was active all year and natural ventilation during summer and transient seasons. It is noted, that no recordings of the temperature levels in the hallways before the installation of the natural ventilation systems were made. Thus, the measured data from the building may only be compared with the simulation results.

However, based on the results after the installation of the natural ventilation system, it is clearly indicated that the theoretical calculations are accompanied by a real improvement in the indoor thermal climate. It is also noted that employees working in the centre has expressed that the indoor thermal climate has improved.

It is unfortunately not possible to present data on the electricity consumption as this was not recorded before the implementation of the natural ventilation system, and due to the fact that after the implementation a number of other energy optimization projects have been carried out. However the measured temperatures indicates a dramatic improvement in the thermal indoor climate and since natural ventilation is able to supply fresh air without energy consumption in situations where the room or building in consideration needs cooling the indoor climate has improved without increasing the energy consumption.

Discussion

This particular case study is a renovation project where natural ventilation is established in hallways with existing mechanical ventilation. Natural ventilation was therefore not included from the early design phase. For a new shopping centre, where natural ventilation is included from the early design phase, the effect of the natural ventilation system can be even higher. For instance more openings can be established in the facades to increase the capacity of the ventilation and the relation between inlet air and the building constructions can be optimized for optimal night cooling.

For a new shopping centre it might as well be possible to include the shops in the natural ventilation strategy maybe for night cooling. E.g. small openings in the facades can let the outdoor air into the shops, where it cools down the construction and inventory during the night so the energy demand for cooling is reduced the following day.

For many low-energy buildings we see an almost year-round cooling demand. For these buildings we expect natural ventilation to have an even greater potential since the energy demand for ventilation almost can be eliminated as the electrical energy consumption for natural ventilation is zero.

Looking at a building in a life cycle and life cost perspective (LCA and LCC) natural and hybrid ventilation systems have a highly performance as well partly because the material consumption for a natural and hybrid ventilation system often is much less than for a mechanical ventilation system. This has been demonstrated in recent analysis of office buildings [7,8].

Conclusion

This case study indicates a large potential for natural ventilation and hybrid ventilation in hallways in shopping centres as well as in similar rooms and buildings.

The dynamic building simulation shows that adding natural ventilation to the existing mechanical ventilation system can reduce the number of hours with operative room temperatures above 28°C with more than 80%. Measurements of the thermal indoor climate in the first year (September 2011 - august 2012) support the building simulation. In this year, the actual results outperform the expected results from the simulations, and the building owner has expressed his satisfaction with the improvements in the thermal indoor climate.

The dynamic simulation further shows that improving the thermal indoor climate can result in energy savings since the electrical energy consumption for air transport can be reduced. The size of the energy saving potential depends on the quality of the thermal indoor climate desired by the building owner – and they will be by far greatest if the mechanical system is switched off during the summer months. It has been demonstrated that doing so might result in an annual reduction of hours with operative room temperatures above 28°C of approximately 70% and energy savings of almost 60%.

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Numerical prediction of the air exchange in the museum premises equipped with natural ventilation systems

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Abstract

Ensuring a proper indoor environment in the museum exhibition rooms requires, among others, the achievement and maintenance of the proper air change rate. It is important because of the minimum rate necessary to remove the excessive heat gains and moisture and energy demand for the ventilation purposes.

Two existing museum buildings were selected for the research purposes. First is located in a historical small castle built in 14th century and the second one is a modern building constructed as a museum in the 20th century. Both buildings are equipped with the natural ventilation system.

The numerical models of both buildings were created and the simulations of the air changes were performed using the CONTAM software. The simulations were carried out for the winter period on the basis of the measured and recorded at this time weather data. As the result of simulations the air change rate in the selected museum rooms are presented against a background of the external temperature.

Keywords

Natural ventilation; museum building; simulation methods; CO_2 concentration.

Introduction

Museum buildings belong to the special group of buildings where the indoor air quality (IAQ) is important. Particularly, the certain parameters, such as indoor air temperature, relative humidity and solar radiation decide on the IAQ. Ensuring the proper conditions in connection to the safety of the exhibits sometimes stands in contradiction to the primary function of museum – making the exhibits available for visitors. The large number of visitors can make the IAQ unsuitable because of the sudden growth of the internal heat gains and the air humidity that can be even dangerous for the exhibition [1]. Fluctuations of temperature and humidity in time affect the exhibits even more than their - although unfavourable - but constant level. Recommended parameters of the internal climate in museums are a relative humidity of 50 % and the temperature of 15 ^oC ÷ 25 ^oC. Deviations of $\pm$ 10 % for humidity are admitted, as well as deviations of temperature by $\pm$ 2 K (these ranges concern short fluctuations of the parameters due to instantaneous redundant gains and the non-homogenous environment – gradients of these parameters over the space of the museum) [2].

Additional difficulties in ensuring the proper IAQ in the museum buildings results from their historical value. Very often those buildings are lack of proper heating and ventilating systems, and because of their heritage importance there is no allowance for rebuilding and reconstruction of the HVAC system.

The paper presents the preliminary results of the investigations carried out within the frame of the research project which targets the identification and assessment of the indoor air quality in the museum buildings and points out the possible activities leading to the improvement of it.

Buildings description

The investigation concerns two museum buildings located in different cities in southern part of Poland in the Upper Silesia region. The buildings differ significantly in relation to the construction, materials used as well as the size and volume (Figure 1).

The first museum is located in a historical small castle built in 14th century. The exhibition rooms are located on three levels of the building. Besides the exhibition rooms there are also storage rooms, offices and laboratories located on the floors. Internal structure of the building is rather complicated because the castle was rebuilt many times giving the existing state. The building is equipped with the ventilating duct system connecting some rooms with the outlet chimneys on the roof. The fresh air infiltrates through the window cracks. Originally probably all castle rooms were ventilated but now because of some alterations of the building a few exhibition rooms are lacking the ventilation. The total exhibition area amounts to 68 m^2, 104 m^2 and 114 m^2 on the 1st, 2nd and the 3rd floor respectively.

The second museum was erected in 1929 - 1930 and was specially designed to serve exhibition purposes. It is a five-storey, double-winged building with the exhibition rooms on the 2nd , 3rd and 4th storey. The total exhibition area is much greater than in the previous building: 400 m^2, 900 m^2 and 650 m^2 on particular floors. The very unfavourable feature of this building is total lack of any ventilating system. The whole building ventilation is maintained only by infiltration mode.

Figure 1: Selected museum buildings.

Method

The aim of the work – assessment of the ventilation in the investigated museums – was realized by the numerical simulation.

The calculations of the ventilation air flows within the considered building was made by CONTAM program. This program is designed for multizone analysis of the ventilation and indoor air quality in buildings [3]. CONTAM can be applied to the global assessment of the ventilation effectiveness in the whole building, search for the time variation of the ventilation air flows in the particular zones or for checking the influence of building air-tightness on the air infiltration. The research program comprises continuous measurement of the main indoor environmental parameters: air temperature and humidity and CO_2 concentration.

The measurement campaign was carried out for whole heating season. In both museum buildings all main exhibition rooms were equipped with the measurement sensors located in selected points of every room. The local weather station provided the set of the meteorological data necessary for simulations.

Models

Two numerical models of both museums were built reproducing internal structure of the whole buildings. Figure 2 presents, as an example, the CONTAM models of the third floors of both buildings. All identified flow paths were modelled, mainly through the windows cracks. The stairwells in both building were also included in the model as the important flow path in case when the ventilation is realized by stack effect. For the model of the museum located in the old castle the system of the ventilating ducts was identified and modelled.

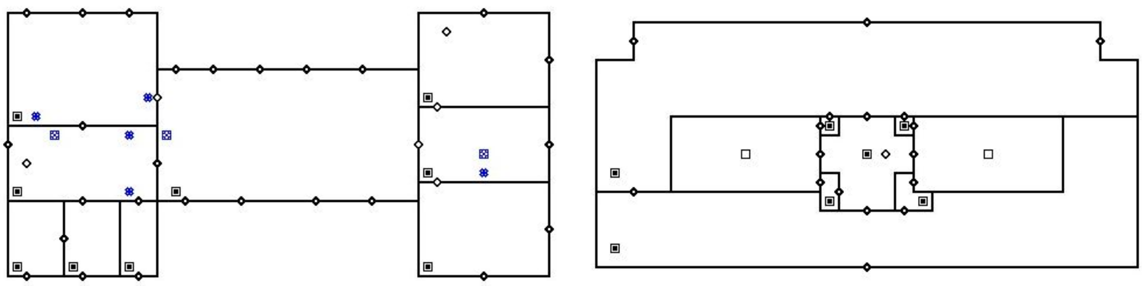

Figure 2: CONTAM models of the museum buildings

One of the biggest uncertainties when the model is created is the value of the air infiltration coefficient which describe the air tightness of the windows. Because of the lack of the windows characteristics the assessment of these parameters were based on the authors experience. In the contemporary building three types of windows were identified: metal, wooden and PVC – all are weather-stripped and double glazed. For the preliminary simulations the air infiltration coefficient was assumed to be equal to 0.2 $m^3/m{\cdot}h{\cdot}Pa^{0.67}$ which means that the windows were tight. In the old castle this issue was more complicated: almost all window openings were different and for this reason 16 types of windows were declared. After inspection in the building the air infiltration coefficient was taken on the level of 1 $m^3/m{\cdot}h{\cdot}Pa^{0.67}$ because the windows were old and not airtight.

The model was adjusted and tuned using the results of the CO_2 concentration measurements. It was assumed that the CO_2 concentration changes because of the presence of the people in the exhibition rooms. The levels of concentration differs for both buildings and for different storey. Thanks to the recording of the CO_2 concentration variation it was possible to apply the concentration decay method [4] to determine the air change rate in particular exhibition rooms in the buildings. After that the results were compared with the simulation results giving the possibility to correct the air infiltration coefficients for windows.

Results

Measurements

The measurements were performed during the last year's heating season, between October and April. In the paper some results for February and March are presented. From the effectiveness of ventilation point of view the CO_2 concentration was interesting. Figure 3 presents the variation of the CO_2 concentration in the chosen exhibition rooms in both buildings. The repeatable changeability of the CO_2 level can be observed: increase of the concentration during the day results from the presence of the personnel and visitors in the museum. In the museum located in the small castle the actual concentration reached sometimes even more than 2400 ppm. In the largest exhibition rooms of the newer building these concentrations are much lower, the maximum was about 1100 ppm. It can be explain by the greater cubature of these rooms when the source of contamination (number of people) is usually comparable in both museums.

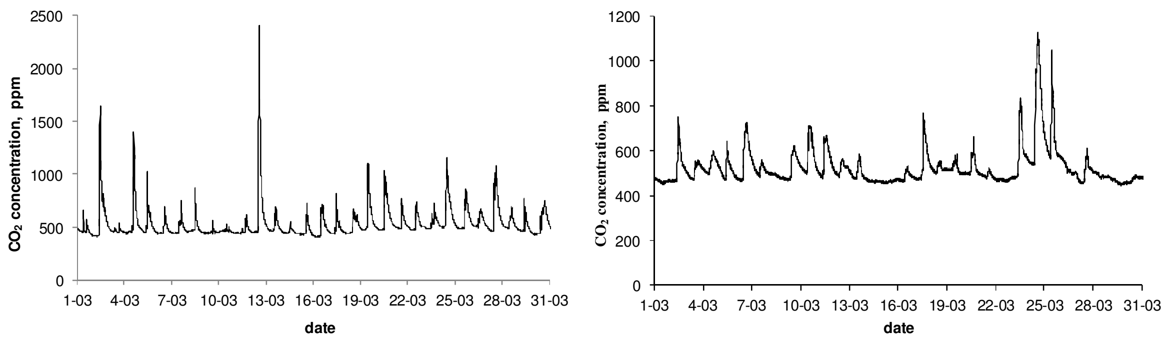

Figure 3: Variation of the CO_2 concentration on the selected storey's of the museum buildings (left – castle, right – modern building)

For several cases the CO_2 concentration decay was calculated giving the air change rate in the particular exhibition rooms. The results could serve for models adjusting and validation.

Simulations

The series of ventilating air flows simulations were performed using the meteorological data recorded during the heating season by the local weather station. The results of the air change rate simulation were compared with the data resulting from the CO_2 concentration decay measurement. After that the tuning of the models was made – that was done mainly changing of the air infiltration coefficients of particular type of windows. Thanks to the number of simulations performed the new value of the air tightness of windows were assumed. In case of the greater building the air infiltration coefficient of the new PVC windows was equal 0.2 $m^3/m{\cdot}h{\cdot}Pa^{0.67}$ and 0.5 $m^3/m{\cdot}h{\cdot}Pa^{0.67}$ for other type of windows (wooden and metal). Regarding the windows in the small castle where the windows were old and not so airtight the air infiltration coefficient was established 0.5 and 1.0 $m^3/m{\cdot}h{\cdot}Pa^{0.67}$.

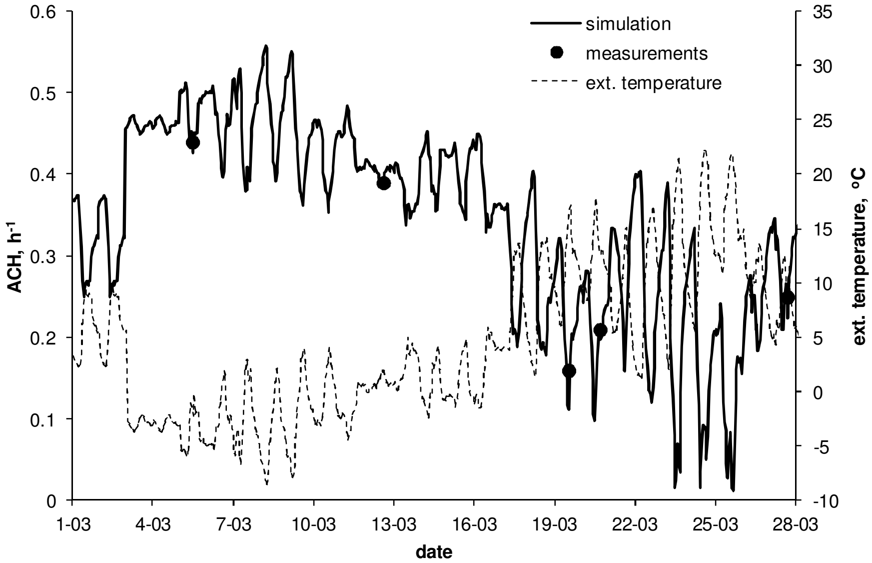

Figure 4: Air change rate in the selected exhibition room in the castle

Figures 4 and 5 show the run of air change rate variation. The measured values of the air change rate were marked with points. In case of the museum located in the old castle quite good compliance was achieved. The results for the second museum are not so good. In fact in the most cases simulation results differ from the measurements. The reason for that is not obviously the fault of the numerical model. The analysis of the measurement data shows not a very big difference between the high level of the CO_2 concentration and the basic level. Hence the calculation of the CO_2 concentration decay can be imprecisely. The second reason of the incompatibility is that in the large cubature of the exhibition room the CO_2 concentration was measured only by one detector. On the other hand the proper use of the concentration decay method consist on the excellent mixing of the contaminant in the whole volume. The ventilation of the exhibition rooms in the considered museum was very poor so the perfect mixing is also uncertain.

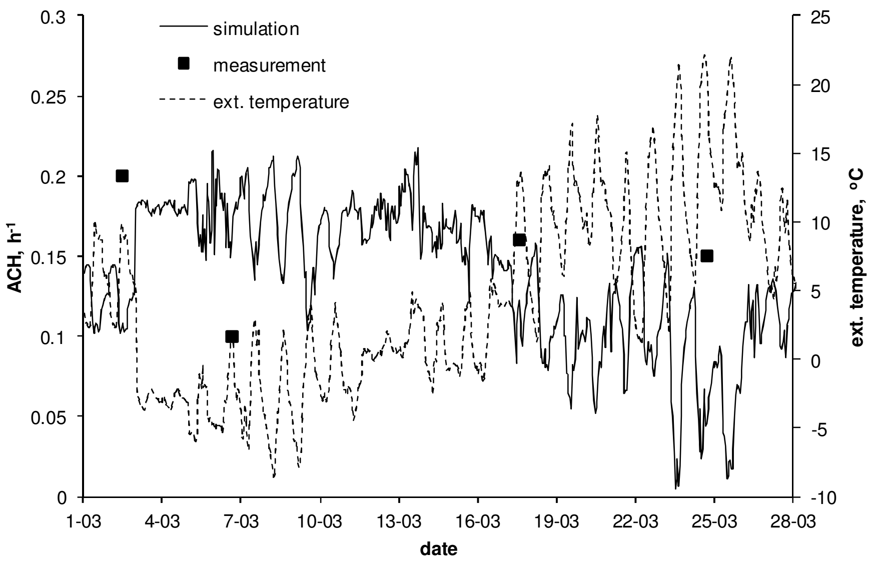

Figure 5: Air change rate in the selected exhibition room in newer building

Conclusions

Ventilation of museums in most cases is limited to natural ventilation. Sometimes the air is exchanged only by opening windows or doors what may cause rapid change in temperature and humidity indoors and may be dangerous for artifacts. Insufficient ventilation and lack of natural ventilation systems may also create problems by excessive increase in humidity and temperature in the exhibition rooms.

The research performed shoud be treated as the introduction to the complex analysis of the ventilation effectiveness in the selected museum buildings.

The work performed gives only preliminary results concern a short period of time. Based on these results some potential problems with ventilation was indicated.

The numerical models presented here were adjusted and partially validated by measurement of CO_2 concentration. The next analysis for the longer period of time (e.g. for the whole heating season) will give more precise assessment of the reliability of simulation models.

Acknowledgements

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Natural ventilation and passive cooling simulation is not any more a privilege of experts

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Abstract

Natural ventilation and dynamic temperature simulation of buildings was until now a privilege of highly skilled building physicists. Combined simulation of both is even rarer.

A new software approach guiding the user to construct rapidly, easily and intuitively step by step a valid thermal model, makes passive cooling design affordable to any building designer, even to a non building physics experts. Dial+ software offers the possibility to simulate for the same room natural ventilation air flow, dynamic thermal behaviour and natural lighting performance. Behind its intuitive interface, Radiance for natural lighting, Cocroft algorithm for natural ventilation and EN ISO 13791 indoor air dynamic simulation give answers to the designers questions in order to optimise design and meet passive building specifications.

The article presents how the software was employed to optimise two similar office buildings situated one in Geneva and one in Nicosia. Both buildings consume less than 40 kWh/m2 of primary energy. The results show how essential is night cooling in both Mediterranean and central Europe climates and quantify the contribution of passive techniques (insulation, glazing performance, solar shading, thermal mass etc.) to the high energy performance. Both buildings were constructed and the performances verified in practice.

Keywords

Potential for ventilative cooling strategies, design approaches for ventilative cooling and case studies, ventilative cooling in energy performance regulations, summer comfort and ventilation.

Introduction

Estia SA in collaboration with the Laboratory of Urban Architecture and Energy Reflexion (LAURE/EPFL) has developed a suite of software tools aiming to simultaneously assess:

- Daylighting: **Dial+Lighting,**
- Natural window ventilation: **DIAL+*Ventilation***,
- Thermal dynamic behaviour for winter or summer conditions: **DIAL+*Cooling***.

The possibility to combine natural and artificial lighting analysis with natural ventilation and dynamic indoor temperature analysis existed in the past, but it was a privilege for researchers, or building physics experts. The corresponding tools did require a high level of expertise, many weeks of training and many working hours to build credible models able to calculate correctly a given reality. Thanks to its intuitive interface, DIAL+ innovates and gives to non-expert users the opportunity to quickly model a complex room and to calculate the following results:

- Dynamic indoor temperature (ISO 13791)
- Number of overheating hours (SIA 382/1, EN 15251)
- Solar gains (taking into account fixe or movable shading and horizon obstacles)
- Stack effect natural ventilation airflow rate (dynamically hour by hour)
- Annual energy demand for heating and cooling (EN 15255, EN 15265)
- Daylight factor values (BREEAM, CERTIVEA)
- Daylighting autonomy (MINERGIE ECO, SMEO, LEED)
- Annual energy consumption due to artificial lighting (SIA 380/4, MINERGIE)

The three software modules are designed with a particular focus on user-friendliness. User data input is simple and straightforward and thus allows planners to correctly model the room and quickly analyse lighting and thermal indoor comfort, even without profound knowledge of building physics.

The software suite DIAL+ therefore represents a simple and efficient professional tool, ideal not only for proving the conformity to various norms and building labels, but also for optimising building design, especially in early planning phases.

Software interface description, methods and more references may be found in [1]. The software possibilities are presented through optimisation in the design process of two similar office buildings, one situated in Geneva and one in Cyprus.

Presentation of the two buildings and some interface windows of the software Dial+

Figure 1: Picture of the south facade and plan of the second floor of the two buildings. Building Ge is on the left and the building Ni on the right

Building Ge is situated in Geneva suburbs - Switzerland in an industrial area, while building Ni in Nicosia city centre – Cyprus. Both of them are low energy office buildings, privileging passive strategies and technical sobriety rather than high standard complex technical installations. Both buildings are naturally ventilated and heated / cooled with a standard solution common in the local market. Both buildings rely on a high energy-performance envelope, passive solar gains and passive cooling. Inauguration of building A took place in 2009, and of building B in 2012.

	Building Ge	Building Ni
Thermal insulation position	external	external
Wall thermal insulation thickness	16 cm	10 cm
Roof thermal insulation thickness	20 cm	10 cm
South glazing g value	0.6	0.4
North glazing g value	0.4	0.4
Window U value	1.3 W/m^2K	1.3 W/m^2K
Glazing light transmittance	0,7	0.7
Ventilation opening dimensions	55X170	40/60X300
South glazing dimensions	340 X 170	140X300
North glazing dimensions	340 X 200	2 X 140X300
Static solar shading south	100 cm top	60 cm top, 60 cm sides
Static solar shading north	-	Vertical fin 60 cm
South movable solar shading	G shading 0.2	-

Table1: technical characteristics of the two buildings

To compare the behaviour of the buildings we isolate a section of the building of 3.6 m large

Some windows from the software interface to enter the space characteristics

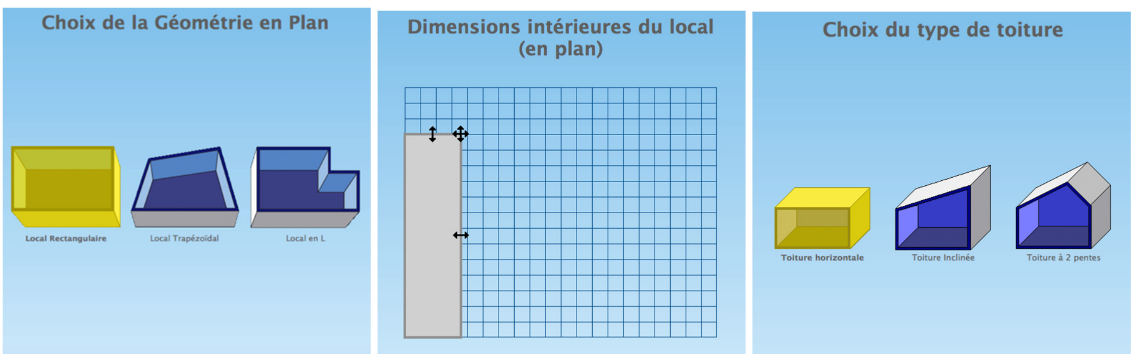


Figure 2: Intuitive interface makes it easy to enter the building dimensional characteristics

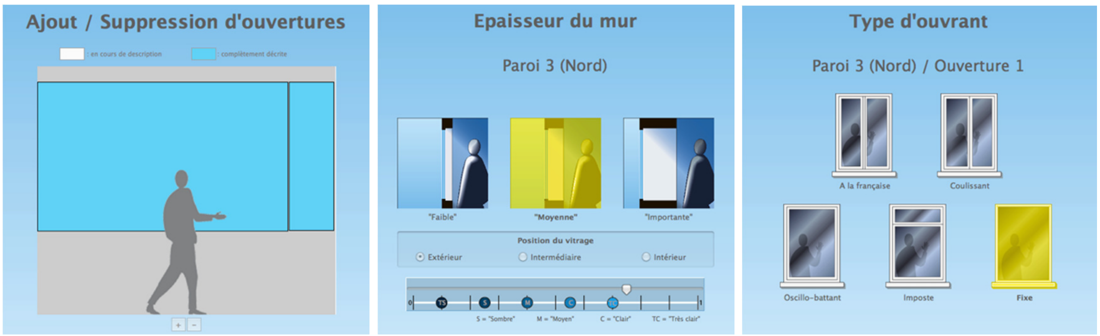


Figure 3: Some window characteristics.

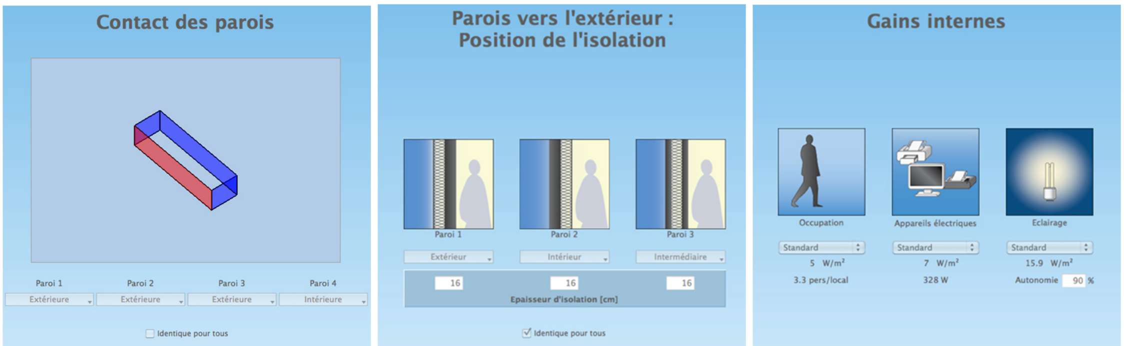


Figure 4: Some thermal characteristics of the building elements and some schedules of conditions of use

These are some of the interface windows showing the intuitive spirit of the software. For every quantitative value, a qualitative description or a picture help the users to introduce a correct model corresponding to reality, even when if they are not specialists.

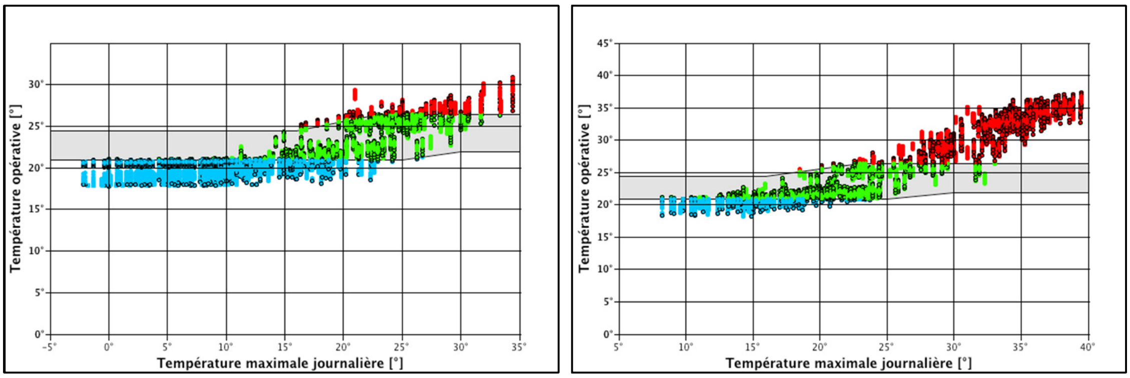
Thermal behaviour of the buildings simulated by Dial+

Both buildings are simulated in the Swiss climate and in the Cyprus climate. Simulations consider no special free cooling strategy except of opening the windows when it is too hot.

Overheating hours

Overheating hours are calculated without and mechanical cooling.

In Swiss climate In Cyprus climate



Ge: 240 overheating hours Ge: 1251 overheating hours

Ni: 191 overheating hours Ni: 1262 overheating hours

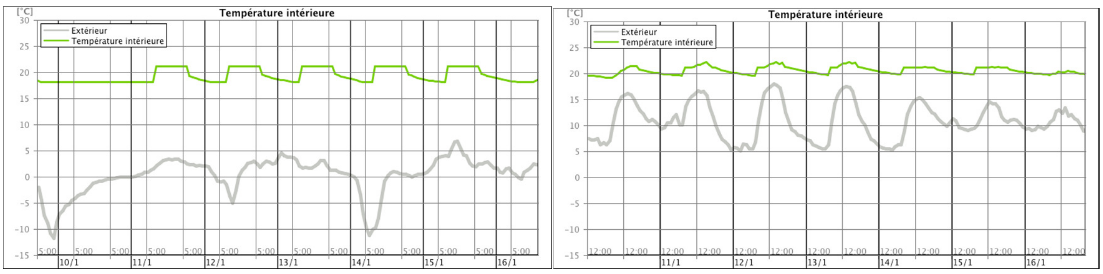
Figure 5: Temperature distribution of the buildings in Geneva and in Nicosia

The behaviour of the two buildings is similar for both climates. The Ni building presents less overheating hours than the Ge building because the glazed part is less in the south façade. Although the Ge building has external solar protection, the Ni building is better protected by the sun.

Heating demand

Heating demand depends on thermal insulation and on solar gains.

In Swiss climate In Cyprus climate



Ge: 56 kWh/m^{2}y of heating demand Ge: 7 kWh/m^{2}y of heating demand

Ni: 65 kWh/m^{2}y of heating demand Ni: 8 kWh/m^{2}y of heating demand

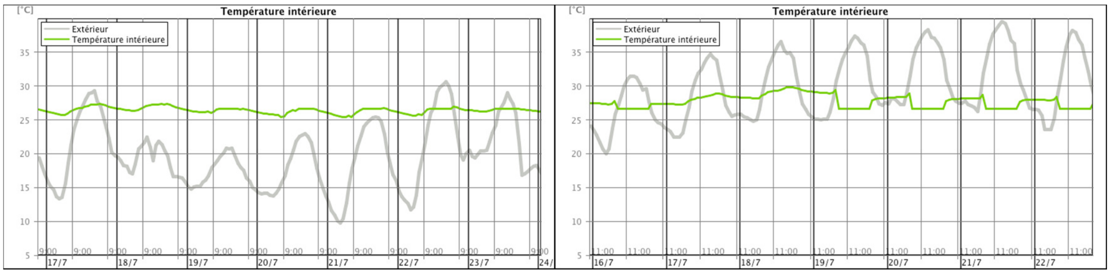
Figure 6: Heating demand of the buildings in Geneva and in Nicosia

The Swiss building with 16 and 20 cm of thermal insulation saves only 1 kWh/m2 in the Cyprus climatic conditions. This means that over the passive standards of 10 cm of insulation more insulation has very little effectiveness. The Ni building with 10 cm of insulation would consume 16% more than the Ge building. This means that additional insulation in the Swiss climate makes sense.

Cooling demand

Cooling demand is calculated with a conventional air conditioning functioning during working hours to bring temperature down to 26.5°C.

In Swiss climate | In Cyprus climate



Ge: 3 kWh/m²y for cooling demand | Ge: 65 kWh/m²y for cooling demand

Ni: 3 kWh/m²y for cooling demand | Ni: 35 kWh/m²y for cooling demand

Figure 7: Cooling demand of the buildings in Geneva and in Nicosia

These results show as for the calculation of the overheating hours that the Ni building is better adapted to the hot climate because of its smaller glazing at the south façade.

Optimisation strategies

Night cooling in Switzerland

Opening the window during working hours when it is too hot for Ge building is not a sufficient strategy. Although opening the windows reduces the overheating hours from 491 to 240 compared with a scenario with the windows closed, poor comfort is still present. The solution is a mechanical cooling, consuming 3 kWh/m²y of a free natural cooling by night ventilation. Night ventilation reduces to 5 the overheating hours, offering satisfactory comfort conditions.

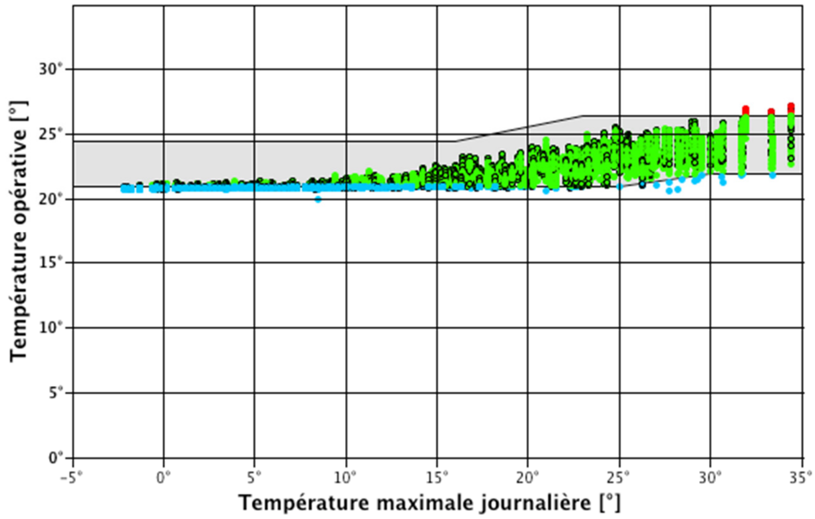


Figure 8: Overheating hours for Ge building with night ventilation strategy for cooling. Overheating hours are reduced to 5

Night cooling in Cyprus

Night ventilation without air conditioning reduces overheating hours from 1262 to 707. It is a considerable reduce but still uncomfortable. However, with the use of air conditioning, night ventilation reduces the cooling demand to 16.5 kWh/m^{2}y instead of 36.5. Cooling power of night ventilation goes up to 1500W or 32 W/m^2.

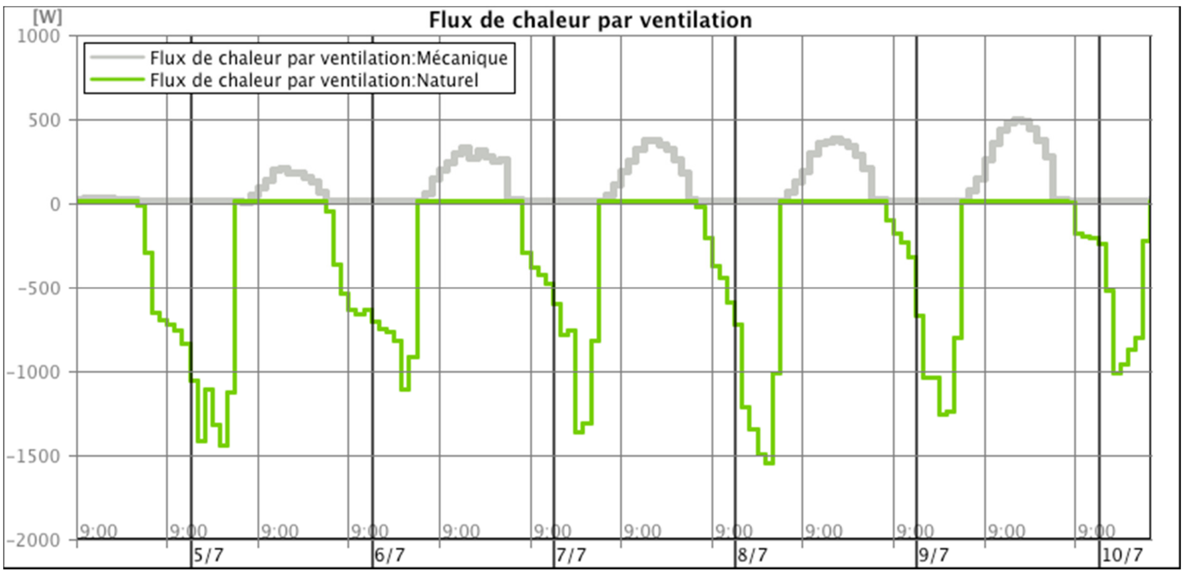


Figure 9: Free cooling power of night ventilation

The presence of a ceiling fan, rising the tolerated highest temperature to 28.5 °C, before switching on air conditioning, reduces cooling demand even more at 6 kWh/m^{2}y instead of 16.5 kWh/m^{2}y with set temperature 26.5°C.

Heat recovery in Switzerland and in Cyprus

In Cyprus a heat recovery system of 80% efficiency reduces heating demand to 4.5 and cooling demand to 33 kWh/m^{2}y instead of 8 and 35 kWh/m^{2}y respectively. The total reduction is 5.5 kWh/m^{2}y or 13%. However, this increases energy consumption to run the fans. The total balance of primary energy is not positive.

In Switzerland, the same system reduces heating demand to 35 kWh/m^{2}y instead of 56. Cooling demand change is not significant. The reduction of 21 kWh/m^{2}y or 37.5% is significant and may pay the primary energy necessary to run the fans, especially when the recovered energy if of high primary energy content.

Conclusion

In order to illustrate the use of DIAL+ software we compared the dynamic thermal behaviour of tow similar office buildings in Cyprus and in Switzerland. This comparison shows the potential of the software, but it also shows that passive technics pay differently in a hot and in a cold climates. For hot climates, instead of airtightness and heat recovery, common in north and central Europe countries, natural ventilation and night cooling are much more efficient. Without night cooling strategy it is difficult to meat passive house standards.

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Simplified hourly method to calculate summer temperatures in dwellings

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Abstract

The objective of this study was to develop a method for hourly calculation of the operating temperature in order to evaluate summer comfort in dwellings to help improve building design.

A simplified method was developed on the basis of the simple hourly method of the standard ISO 13790:2008 but with further simplifications. The method is used for calculating room temperatures for all hours of a reference year. It is essential that the simplified method is able to predict the temperature in the room with the highest heat load. The heat load is influenced by the solar load, internal load, ventilation loss, heat loss and the external temperature. The numbers of hours exceeding 26 °C and 27 °C are summarised in order to compare them with the requirements for summer comfort in the Danish Building Regulations.

The simplified method was qualified by comparison with simulation results from an advanced program for thermal simulations of buildings. The results are based on one year simulations of two cases. The cases were based on a low energy dwelling of 196 m^2. The transmission loss for the building envelope was 3.3 W/m^2, not including windows and doors. The dwelling was tested in two cases, a case with an ordinary distribution of windows and a "worst" case where the window area facing south and west was increased by more than 60%.

The simplified method used Danish weather data and only needs information on transmission losses, thermal mass, surface contact, internal load, ventilation scheme and solar load.

The developed method can calculate the number of hours above a given temperature limit. The limits are a prerequisite for the development of the simplified method, and a supplementary maximum temperature limit is suggested to ensure robustness. The setting of the ventilation rate is found to be essential for the fulfilment of summer comfort. Thus it is very important to address both opening areas and ventilation rates.

The developed simplified method makes it possible to test whether or not a building design for a dwelling will prevent excess of the summer comfort limits set by the building regulations.

Keywords

Hourly temperature calculation, summer comfort, simplified method

Introduction

The Danish Building Regulations 2010 (BR10) [1] set new requirements for the thermal comfort in summer. The Building Regulations applies to buildings in low energy class 2015 (LE2015) or in building class 2020 (BC2020). For larger buildings, the documentation must primarily be based on advanced thermal simulations, and it is up to the developer to specify requirements for temperature conditions in summer. For dwellings, the requirements are that 26 ° C must not be exceeded for more than 100 hours a year and 27 ° C must not be exceeded for more than 25 hours a year.

The purpose of this paper is to present a simplified method for calculation of the temperature conditions in dwellings during summer. The simplified method is developed on the basis of the simple hourly method of the standard ISO 13790:2008 [2], Annex C but with further simplifications.

BR10 contains a requirement that the energy performance framework of buildings must be verified by the program Be10, which is a method of calculating the energy demands of buildings [3]. In Be10, all energy-related data for a building is provided, but since the program calculates monthly average, it cannot be used directly for evaluation of summer comfort. However, the main idea of developing a simplified method is to connect it to the Be10 program, so that the building data provided for the energy calculation can be reused in the simplified method for evaluation of summer comfort.

Primary, the ability to develop a method that can be applied to the dwelling, as a whole, is preliminarily examined. This is a crucial point for developing a simplified method, which may be attributed to the program Be10 [3].

Additionally, it is investigated whether supplementary temperature requirements for dwellings can enhance robustness.

Methods and models

The full set of equations for a simple hourly method given in ISO13790: 2008 [2], Annex C, is the basis of this new simplified method. The method was simplified with assumptions about summer conditions in new low energy dwellings in Denmark.

The preconditions for further simplification of the method for determination of summer temperatures in dwellings were:

- All solar heat gains were assumed allocated in the air, i.e. small error when the sun shines directly on the wall or floor, but with internal blinds or other light surfaces the solar heat will actually be allocated in the air.
- It was primarily the internal room surfaces of the structures that interacted with the room air so the heat transfer from the deep mass to the surface was neglected.
- The operating temperature was a combination of air and surface temperatures but not calculated directly as in ISO: 13790:2008 [2].
- Ventilation with outdoor air but the heat recovery can be by-passed in summer.

The simplified method for calculation of temperature conditions in dwellings during summer was carried out in a spreadsheet by using the full set of equations in ISO 13790. To begin the calculation, the initial and boundary conditions should be given. These included initial air and surface temperatures and a minimum air temperature. In addition, some parameters were retrieved directly from Be10 while others had to be specified. The summer ventilation rate was determined based on the effective opening areas for respectively the dwelling and the most critical room. The ventilation loss was calculated by use of the ventilation rate.

The spreadsheet calculated operating temperature as a combination of air temperature and surface temperatures. The heat gain was found by adding solar load and internal load and the heat loss by adding ventilation loss and transmission loss. By subtracting the heat gain from the heat loss the resulting loss/gain was found. By this the heat transfer between the air and surface temperature was calculated and thus new air and surface temperatures were found and the next hours operating temperature could be calculated.

The monthly solar load was retrieved from Be10 and was adjusted to proportionality of the global radiation for each month.

Test cases

The test cases were based on a dwelling design by "Eurodan huse" where the building envelope has been adapted to comply with energy requirements for both LE2015 and BC2020. The requirement for transmission loss for a BC2020 new construction is that it should not exceed 3.7 W/m^2 and for the test case the transmission loss for the current building was 3.3 W/m^2.

The total floor area of the dwelling was 196 m^2 and a floor plan is shown in Figure 1. The living room had large window areas facing both south and west, and it was considered to be at highest risk of having too high summer temperatures. Therefore, the living room was found to be the most critical room and thus served as a reference room. The living room had a floor area of 29 m^2 and a room volume of 72 m^3. The described dwelling is referred to as Case 0 and is shown in Figure 1.

In this study a "worst case" was constructed by maximising the window areas of the south and west facing facades. This constructed variant is called Case 20 and is shown in Figure 2.

In both cases, the dwelling had balanced ventilation. There was exhaust from; utility room (10 l/s), kitchen (20 l/s) and bathrooms (15 l/s). The ventilation had heat recovery with 85% efficiency, which was bypassed in summer when heat recovery was inappropriate.

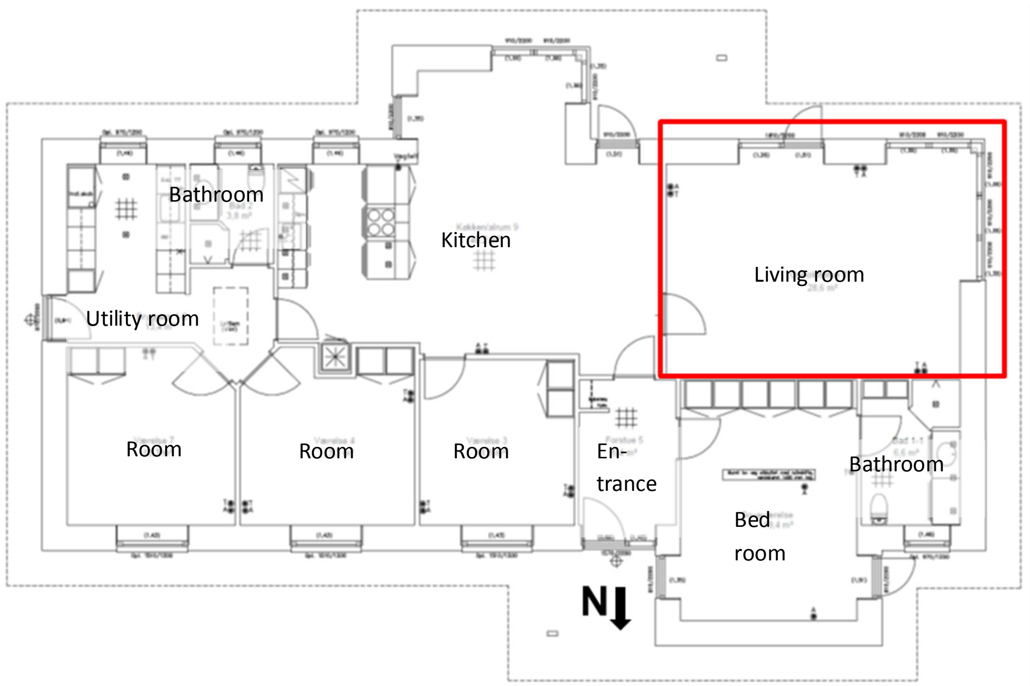

Figure 1: Plan of 196 m^2 low energy dwelling, corresponding to Case 0. The highlighted area is the living room.

Simulation tool

To test the simplified calculation method, a series of calculations was performed with BSim [4], which is an advanced program for building energy simulation. The program simulates operating conditions and the dynamic interaction between building installations and associated automatic control systems. This makes it suitable for indoor climate analysis and calculation of energy consumption in buildings. In analysis, this program is especially suited for parameter studies, for example of sunlight, use of passive solar protection, window sizes and orientation. In this study, BSim version 6, 11, 6, 20 was used.

Operating time

By definition, the operating hours of a dwelling are 24 hours a day. This means that occupants are always expected to be at home to regulate the indoor climate. In this study simulations were made for a time period of one year.

Venting

In BSim, it is possible to have advanced ventilation control settings. Since the simulations concerned a dwelling, the ventilation was expected to follow the operating time, which is always. However, in order to assess the robustness of the solution a simulation was also performed for a case where the dwelling was vacant and thus had limited ventilation.

Ventilation control

1. When the dwelling was always occupied, it was assumed that the window was opened whenever the air temperature exceeded 23 °C.

2. When the dwelling was vacant, the ventilation was restricted to passive venting or basic ventilation.

Sun protection

The robustness was assessed by simulation where the dwelling was vacant and venting therefore limited. As this easily gave high temperatures, it could be assumed that there were internal solar protection in rooms facing south and west, which was always activated. When the dwelling was vacated, it corresponded to the curtains being drawn.

Glass areas

In the simulations, two cases were investigated. The cases are shown in Figures 1 and 2 and have been described previously. Case 0 corresponds to an ordinary dwelling, while Case 20 is a constructed "worst case" variant of the same dwelling and more likely to experience problems of high temperatures in the summer due to enhanced window areas.

Models

A number of BSim models were created as the example shows for Case 20 in Figure 2, or part of it like Case 50. In the model shown in Figure 2, each room was designed as a separate thermal zone, but there were also models where all the rooms were combined into a single thermal zone. The model variants are described separately below.

Common to all models was that four simulations were performed on each model:

1. A simulation where the dwelling is always occupied and without solar protection
2. A simulation where the dwelling is always occupied and with solar protection
3. A simulation where the dwelling is always vacant and without solar protection
4. A simulation where the dwelling is always vacant and with solar protection

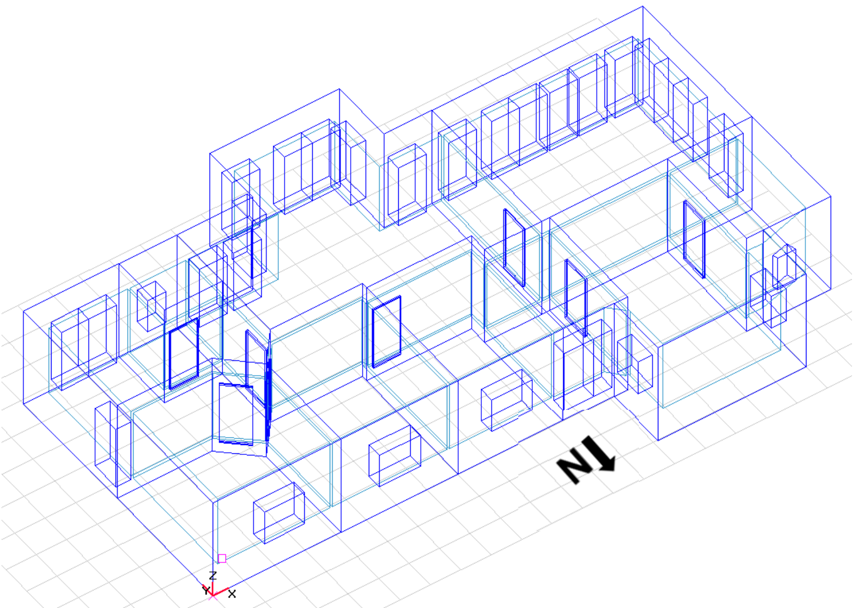

Figure 2: Model of Case 20, a constructed “worst case” variant of Case 0 with substantially increased window area in the South and West façade

Case 0

The geometry of Case 0 is shown in Figure 1. Each room has a separate thermal zone and there is no air mixing between the rooms, which correspond to closed doors between rooms.

Case 0X

In this model the geometry is as in Case 0. Opposite to Case 0 all rooms in the model is associated with the same thermal zone, corresponding to open doors and full air mixing between the rooms.

Case 20

The geometry of Case 20 is shown in Figure 2. The model is similar to Case 0, but with larger window areas to the south and west. Each room has a separate thermal zone and the doors are closed between the rooms.

Case 20X

In this model the geometry is as in Case 20. Opposite to Case 20, all rooms in the model is associated with the same thermal zone, corresponding to open doors between the rooms.

Case 40

This model only contains the living room with adiabatic interior surroundings. The model corresponds to the living room in Case 0 and the room is one single thermal zone.

Case 50

This model only contains the living room with adiabatic interior surroundings. The model corresponds to the living room in Case 20 and the room is one single thermal zone.

Simplified method simulations

By use of the simplified method, calculations were made for Case 0, see Figure 1 and Case 20, see Figure 2. Calculations were performed for every hour throughout the year using the simplified method and the total number of hours exceeding respectively 26 °C and 27 °C was summarized.

Basically, the simplified method can be used to calculate the temperature condition for every single room in the dwelling. However, if general data for the dwelling is used, as provided in the program Be10 (for energy calculation), the calculation results would correspond to a case with full mixing between zones and open doors between rooms. Thus the results should be comparable with BSim results for cases with just one thermal zone like e.g. Case 0X.

Therefore, to address the possible high temperatures in the living room, a simulation must be performed where the solar load corresponds to the most critical room, which in this case was the living room. The solar load in the most critical room was the maximum solar load.

The highest ventilation rate for the dwelling was proportional to the sum of the basic ventilation and the effective opening areas. As for the solar load, it was necessary to evaluate a ventilation rate for the entire dwelling as a whole, and one based on the maximum ventilation for the critical room. The lowest of the two ventilation rates per floor area should be used in the simplified method for calculation of summer comfort. When calculating the ventilation rate, it was allowed to assume cross ventilation for the most critical room through the door if the rest of the dwelling could be cross ventilated. A limit for a maximum air change rate should be provided or possibly converted into a maximum ventilation rate, e.g. 10 $l/s \cdot m^2$.

Additional temperature requirements

In order to ensure the robustness of the dwelling design an additional maximum temperature could be suggested for a vacant dwelling. A supplementary requirement would ensure that pets can stay in vacant dwellings without suffering too high temperatures. A proposal for an additional temperature requirement might be that the temperature in a vacant dwelling should not exceed the maximum outdoor temperature (32.1 °C) for more than a couple of degrees, and thus have a maximum temperature of no more than, e.g. 34 °C.

Results

The results of the simulations are shown in Tables 1 and 2. Table 1 gives the BSim results and Table 2 gives the results of the simplified method.

Temperatures	Case 0		Case 20		Case 0X		Case 20X		Case 40X		Case 50X	
Internal solar protection	+	–	+	–	+	–	+	–	+	–	+	–
Hours above 26 ° C	97	61	247	199	15	6	42	27	119	69	311	191
Hours above 27 ° C	46	23	148	117	3	0	14	6	60	29	205	110
Vacant dwelling												
Max. operating temperature, °C	*33.6*	*33.0*	*41.9*	*38.3*	*31.3*	*30.2*	*34.3*	*32.6*	*49.3*	*43.2*	*55.8*	*48.0*

Table 1: BSim results of simulations.

BSim results

The results in Table 2 show that Case 0 with internal shading comply with the temperature limits of maximum 100 hours over 26 °C and maximum 25 hours over 27° in the living room. In Case 0X and Case 20X, both with and without shading, the temperature requirements for thermal comfort were also fulfilled, but these cases did not evaluate the most critical room separately as the entire dwelling was one thermal zone.

For the proposed additional requirement of a maximum temperature of 34 °C when the dwelling is vacant, the results were that Case 0 fulfilled the proposed requirement whereas Case 20 did not. Furthermore, Case 0X with and without solar shading and Case 20X with solar shading were below the proposed minimum temperature.

Case 40X and Case 50X did not fulfil any of the requirements.

Results of the simplified method

Table 2 show results for the calculations with the simplified method. There are results for two different solar loads, one is an average for the dwelling (basic) and one for the most critical room (max). In the case with the basic solar load, a combination with minimum ventilation rate of 0.9 l/s·m^2 in day and evening hours and 0.6 l/s·m^2 at night was calculated as described in SBi Guidelines 213 [3], and a similar investigation performed for a doubling of the minimum ventilation rate. The other ventilation rates were based on the effective opening areas of the dwelling and the most critical room, for Case 0 it was respectively 4.3 and 10.6 l/s·m^2. In the calculations of the ventilation rates it was assumed that all windows could be opened.

Temperatures	**Case 0**						**Case 20**					
Solar load	basic	basic	max	max	max	max	basic	basic	max	max	max	max
Ventilation rate, l/s·m²	0.9	1.8	0.9	1.8	4.3	10.6	0.9	1.8	0.9	1.8	6.1	15.6
Hours above 26 ° C	199	70	908	396	107	41	342	129	1763	798	154	49
Hours above 27 ° C	62	9	611	227	46	20	176	47	1339	531	73	23
Max. operating temperature, °C	*28.1*	*27.3*	*33.8*	*30.8*	*29.1*	*29.1*	*29.5*	*28.0*	*38.4*	*34.1*	*30.0*	*29.6*
With internal solar shading ↓												
Hours above 26 ° C	185	66	825	354	102	37	320	123	1641	744	130	45
Hours above 27 ° C	49	6	547	203	38	17	164	42	1198	483	60	23
Max. operating temperature, °C	*28.0*	*27.3*	*33.4*	*30.5*	*29.0*	*29.0*	*29.3*	*27.9*	*37.6*	*33.5*	*29.9*	*29.6*

Table 2: Results of the simplified method

The results in Table 2 show that Case 0 both with and without internal shading complied with the temperature limits of maximum 100 hours over 26 °C and maximum 25 hours over 27 °C for the case with basic solar load and a doubling of the minimum ventilation rate and the case with maximum solar load and ventilation rate calculated on basis of the most critical room, living room. For Case 20, the only case to fulfil the requirements, the maximum solar load and ventilation rate was calculated on basis of the most critical room, living room.

According to the results, the influence of internal solar shading was quite small. However, the results differed for the overall image for cases without solar shading by reducing the number of hours above 27 °C.

Case 0 fulfilled the proposed requirement of a maximum temperature of 34 °C, whereas Case 20 did not. However, this was based on the ventilation rates given in Table 2 and not for a vacant dwelling with limited ventilation.

Comparing BSim results and results of the simplified method

For Case 0, the results indicated that it fulfilled the thermal comfort requirements for both the advanced BSim calculation and the simplified calculation method if the ventilation rate was sufficient. For the proposed additional temperature requirement of a maximum temperature of 34 °C, the results were the same that Case 0 fulfilled the requirement for both calculation methods.

For Case 20, the result indicated that it did not fulfil the thermal comfort requirements for the advanced BSim calculation, except in Case 20X. The same result was found for the simplified calculation method except in the case with maximum solar load and ventilation calculated on basis of the most critical room. For the proposed additional temperature requirement of a maximum temperature of 34 °C, the results were similar to the other Case 20 results. The maximum temperature limit for BSim results were exceeded for Case 20 but fulfilled for Case 20X, and for the simplified method the maximum requirement was met for the cases with maximum solar load and ventilation calculated on basis of the entire dwelling and the most critical room.

Discussion

Summer comfort

In BR10 contains requirements for thermal comfort in summer, which apply to buildings in low-energy class 2015 (LE2015) or in building class 2020 (BC2020). Specifically for dwellings, the requirements stipulate that 26 °C must not be exceeded for more than 100 hours a year and 27 °C not for more than 25 hours a year. Two calculation methods were tested on two dwelling cases. Both methods gave results that show that Case 0 complies with the summer comfort temperature requirements. Therefore, it is concluded that Case 0 is an example of a dwelling, which meets the requirements for thermal comfort in summer. Opposite, the results for Case 20 showed that the summer comfort requirements cannot be met. This matches our expectations, since Case 20 is a constructed “worst case” variant of Case 0 with substantially increased window area in the south and west façades.

In general there is agreement between the results of the two simulation methods, the advanced BSim program and the developed simplified method. The results for the influence of internal solar shading are quite small. The effect is that the number of hours above 27 °C is lowered for cases with internal solar shading. This is important, as this temperature limit seems to be the most restrictive requirement for summer comfort.

Supplementary temperature requirements

Occupants have different user behaviours, but in all cases it must be assumed that dwellings will be vacant for longer or shorter periods. This differs from the assumption in these calculations. Therefore, an additional requirement of a maximum temperature, which must be observed when the dwelling is vacant, will help to ensure that pets and flowers do not suffer from extremely high temperatures. The proposed requirement is a maximum temperature of 34 °C, and the results from both calculation methods show that Case 0 complies with the requirement, while Case 20 do not. Thus, an additional requirement of a maximum temperature when the dwelling is vacant will support the other summer comfort requirements for the thermal indoor climate.

Caution with calculated temperatures

As in every other calculation program, both Be10 and BSim give results that are highly dependent on the calculation parameters. Therefore, it should be emphasised that the results in terms of number of hours above 26 °C and 27 °C do not correspond to the number of hours that can be measured in an occupied dwelling. One assumption in the calculation is that dwellings are always occupied and thus that occupants can provide extra ventilation to prevent too high temperatures. However, the requirements for evaluation of summer comfort can help to prevent the construction of dwellings that are vulnerable to solar gain and have limited window openings that are unable to provide sufficient ventilation.

Future work

The simplified method for hourly calculation of the operating temperature in order to evaluate summer comfort in dwellings needs to be tested on other dwelling designs. This should also include block of flats and terraced houses.

Conclusion

A simplified method has been developed for hourly based calculation of temperatures in dwellings. The simplified method enables evaluation of the summer comfort in dwellings. The calculation method applies to dwellings that are by definition always occupied and can thus be vented. The highest temperatures will be experienced in the most critical room. Therefore, the calculation is performed on basis of the solar load for that room.

The method ensures that there is adequate ventilation by windows that can be opened in both the most critical room and in the whole dwelling. The simplified method can be implemented in the program Be10, so that program as an additional result can provide the summarised number of hours above respectively, 26 °C and 27 °C, which can be used to evaluate summer comfort.

The calculation method was tested on two dwellings, Case 0 and Case 20. The result was that Case 0 meets the thermal summer comfort requirements, while Case 20 does not. This result was expected, as Case 20 is a constructed "worst case" variant of Case 0 with substantially increased window area in the south and west facade.

Along with the development of the method, a proposal is made for a supplemental maximum temperature requirement for a vacant dwelling. Also in this case the results show that Case 0 can comply with the requirement, while Case 20 cannot. The proposed maximum temperature in a vacant dwelling is 34 °C.

The simplified method summarises the number of hours above 26 °C and 27 °C, but that does not necessarily mean that this number of hours will correspond to the number of hours with high summer temperatures in a real dwelling. The number of hours with high summer temperatures will largely be driven by user behaviour. However, it is expected that the simplified method can help to ensure that there is sufficient opportunities for ventilation of the dwelling and suitable solar protection.

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Addressing summer comfort in low-energy housings using the air vector: a numerical and experimental study

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Abstract

This article deals with summer comfort and room air distribution in low-energy housings. In such buildings, the efficient thermal insulation and air tightness make it crucial to efficiently dispose of the heat released by the internal gains. In this prospect, the comfort in a test room resulting from an integrated cooling and ventilation system is assessed both experimentally and numerically. The air is supplied into the room close to the ceiling through a wall-mounted diffuser of complex geometry composed of 12 lobed nozzles. Experimentally, the air velocity, CO_2 concentration, indoor air, wall and globe temperatures are monitored to assess the indoor environment quality. Numerically, CFD software Star-CCM+ is used to provide valuable information on the airflow patterns in the room. The CFD simulations are run in two steps in order to correctly integrate the complex diffuser's geometry. An excellent indoor environment is obtained in the studied conditions. Furthermore, a parametric study is performed in order to investigate the influence of the heat sources and of the supplying conditions on the airflow and on the resulting comfort.

Keywords

HVAC, air diffusion, summer comfort, experimentations, CFD

Introduction

Poor indoor air quality and thermal comfort have an important impact on health, well-being and productivity [1]. It is therefore necessary to supply fresh air to the building and dispose of heat, particles, humidity and other pollutants generated in the building. This issue is particularly crucial for low-energy buildings. In fact, they benefit from good thermal insulation and air tightness. Consequently, the heat released by internal gains (*i.e.* the occupants and electronic devices) cannot easily be disposed of. While this reduces to a great extent the energy required for heating in winter conditions, it also makes it particularly difficult to maintain an acceptable indoor temperature in summer conditions. Therefore, the air distribution system should be able to efficiently evacuate the warm, old air, and ensure a homogeneous cooling of the occupied zone. In this prospect, an integrated cooling and ventilation system is considered, with conditioned air blown through a wall-mounted mixing diffuser. The resulting summer comfort in a room is assessed both experimentally and numerically. The experimental method aims at assessing the thermal comfort and ventilation efficiency by measuring the flow parameters in the occupied zone of a test room. At the same time, it provides realistic boundary conditions for the CFD simulations, whose purpose is to give additional information on the airflow patterns inside the room and in the jet region. Additional CFD simulations are then performed for supplying conditions which are not tested experimentally, in order to pinpoint the influence of the supplying conditions on the airflow pattern.

Experimental method

Test room

The experiments are conducted in an airtight test room composed of steel wall panels (Figure 1a). An air treatment unit is used to blow the inlet air at the specified temperature and flow rate and the wall temperatures are controlled with a thermal guard enclosing the test room. In order to enhance the mixing of the fresh, cold air, with the indoor air, a wall-mounted diffuser is used. It is composed of 12 complex lobed nozzles with a nozzle to nozzle spacing of 8 cm (Figure 1b) and the first nozzle is located 16 cm below the ceiling. Two black cylindrical manikins of height 1.1 m and diameter 0.5 m are located inside the room to account for occupancy, with a CO_2 emission rate of 18 L/h. Lamps are placed inside the manikins to account for the sensible heat source, corresponding to a metabolism of 1 met.

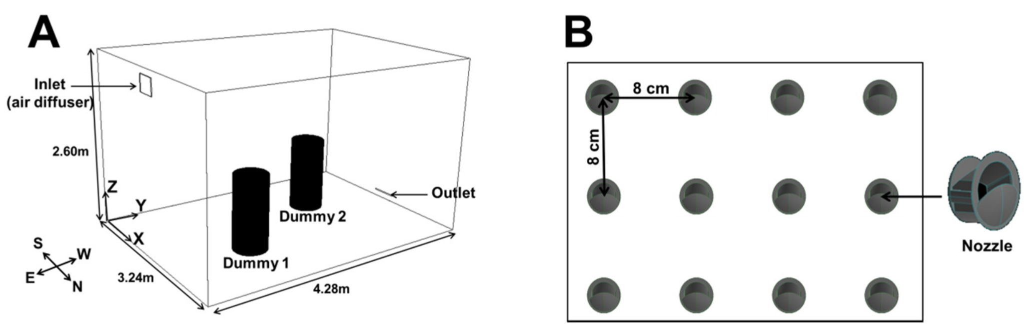

Figure 1: a) Room geometry, b) Diffuser geometry and detail of one nozzle.

Measurements

Measured variables include boundary conditions measurements at the air inlet, outlet and walls, as well as indoor variables measurements in order to assess the comfort. Air and globe temperature, air velocity and CO_2 concentration are monitored on 27 positions inside the occupied zone (Figure 2), corresponding to the ankle level (0.1 m), the neck level of a seated person (1.1 m) and the neck level of a standing person (1.7 m). The results are obtained when steady-state conditions are reached for all the measured values.

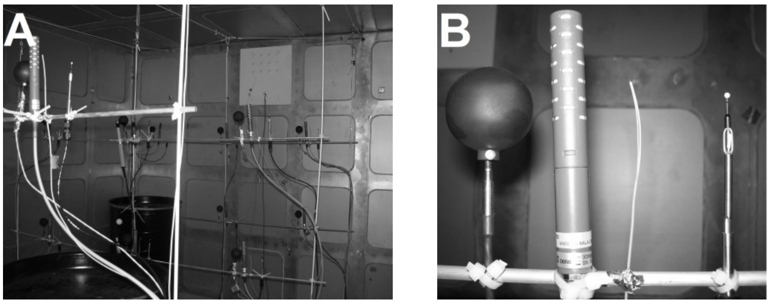

Figure 2: Experimental setup. a) Test room, b) Monitoring position in the occupied zone (globe thermometer, CO_2 probe, type-K thermocouple and omnidirectional thermal anemometer)

Numerical method

Numerical parameters

CFD code STAR-CCM+ is used to solve the 3D transport equations for mass, momentum, energy and CO_2 over the computational grid under steady-state conditions. The realizable k-ε turbulence model is chosen, which has been giving good results in room airflow simulations [2]. Two-layer wall functions are employed to model the heat transfer at the walls and buoyancy forces are taken into account through the variation of the density of the fluid considered as an ideal gas. Convergence is ensured by monitoring the residuals and the air and carbon dioxide mass balance evolution.

Diffuser modelling

The air diffuser is responsible for most heat and mass transfers inside the room, it is therefore crucial to correctly implement its geometry in the CFD simulation. However, the complex details of the diffuser make it too costly to directly include its complete geometry in the simulation. Several diffuser modeling methods have been developed to specify simplified boundary conditions either at the level of the diffuser

(Momentum method) or on a surface from a distance of the diffuser (Box Method). However, these methods require either precise measurements at the exit of the diffuser or characteristic equations associated with the diffuser geometry [3]. Since this information is not available, another approach is followed. A first simulation of only one nozzle of the diffuser is performed in a reduced computational domain (3M polyhedral cells), whose outlet boundaries are located 20 equivalent diameters away from the nozzle. The velocity components, turbulence quantities and temperature profiles are then collected at the very exit of the nozzle (Figure 3). Prior simulations have been run with several adjacent nozzles in order to ensure that the flow is not affected by the other nozzles at such a close distance. The collected profiles are then set as inlet boundary conditions in the full room simulation for each of the 12 nozzles (Figure 3c). A similar approach had been successfully used by Cehlin and Mosfegh[4] with a perforated displacement diffuser.

The test room calculation domain is then composed of 3M polyhedral cells (Figure 3c), the grid being refined in regions of high gradients (at the exit of the diffuser, close to the walls and to the dummies). The inlet boundary conditions correspond to the collected profiles, and an atmospheric pressure condition is specified at the exhaust. The other boundary conditions used for the CFD simulations were provided by the experiments.

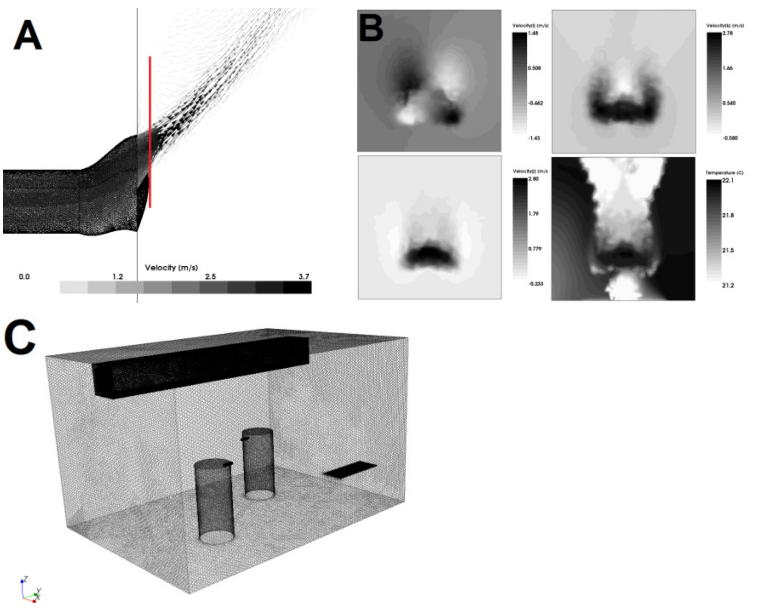

Figure 3: Diffuser modelling approach. a) Single nozzle simulation and profile collecting section, b) Collected velocity components and temperature profiles set as boundary conditions in the full room simulation c) Full room simulation (3M polyhedral cells).

Results and discussion

The obtained results are presented below, and analysed in terms of thermal comfort and ventilation efficiency. The airflow patterns are then investigated with the help of the CFD simulations in order to pinpoint the influence of the internal gains and supplying conditions on the airflow. The conditions of the studied cases are provided in Tab. 2, where T_{walls} corresponds to the average surface temperature of the six walls, and P_{inlet} is calculated as follows:

$$P_{inlet} = Q_0 c_p \left(T_0 - T_{OZ}\right) \tag{1}$$

Where Q_0 is the inlet air mass flow rate, T_0 the inlet temperature, and T_{OZ} the average temperature in the occupied zone. The air flow rate is the same for all cases (1,6 air change per hour), and is chosen in order to provide the cooling power while maintaining the supplied air temperature high enough. The Reynolds number (Eq.3), which qualifies the nature of the flow, and the Archimedes number (Eq.4), which is the ratio of the buoyancy force to the inertia force, are based on the equivalent diameter D_e of the free area of the 12 nozzles $A_{diffuser}$ composing the diffuser (Eq.2).

$$D_e = 2\sqrt{\frac{A_{diffuser}}{\pi}} \quad (2) \quad ; \quad Re_0 = \frac{\rho U_0 D_e}{\mu} \quad (3) \quad ; \quad Ar_0 = \frac{g\beta_T |T_0 - T_{OZ}|\, 2\sqrt{A_{diffuser}/\pi}}{{U_0}^2} \quad (4)$$

Where U_0 is the inlet velocity.

	Q_0			T_0	$T_{exhaust}$	T_{guard}	T_{walls}	P_{dummy}	P_0	Q_{CO2}	$T_0 - T_{oz}$	Ar_0	Re_0
	m^3/h	kg/h	ACH	°C	°C	°C	°C	W	W	l/h	°C	-	-
Case 0	55.6	66.7	1.62	21.5	24.9	28.0	25.7	-	69	-	-3.7	9.6E-04	16384
Case 1	57.0	69.4	1.66	17.0	25.3	28.0	25.9	107	162	18.0	-8.3	2.1E-03	17247
Case 2	56.6	69.4	1.65	14.9	24.8	27.2	25.4	213	200	36.0	-10.3	2.7E-03	17356

Table 1: Boundary conditions for the studied cases

Global comfort

The EN 15251 standard [5] was used to evaluate the indoor environment quality (IEQ). It provides a classification of the IEQ depending on the values of the operative temperature, Predicted Mean Vote as defined by Fanger [6], and carbon dioxide concentration level. Category I corresponds to an IEQ level expected in schools or buildings with sensitive persons. Category II is the level expected for new or refurbished buildings, while Category III is the value expected for existing buildings. In addition, the ISO 7730 standard [7] defines the Draught Rate as the percentage of dissatisfied for a given air velocity, air temperature and turbulence intensity.

Category	Maximum operative temperature (cooling season) (°C)	PMV	Maximum CO_2 level above outside concentration (ppm)
I	25.5	< 0.2	300
II	26	< 0.5	500
III	27	< 0.7	800

Table 2: IEQ categories according to the EN 15251 standard [5]

The contaminant removal effectiveness (Eq.5) and temperature efficiency (Eq.6) as defined by Sandberg [8] are also considered, and express the ability of the air distribution system to dispose of a pollutant, and to cool the indoor air, respectively:

$$\varepsilon_C = \frac{c_{exhaust} - c_0}{c_{OZ} - c_0} \tag{5}$$

$$\varepsilon_T = \frac{T_{exhaust} - T_0}{T_{OZ} - T_0} \tag{6}$$

The global results obtained for the experimental test cases are provided in Tab.3. The average, minimum and maximum values of the operative temperature, PMV and carbon dioxide concentration are provided, as well as the contaminant removal effectiveness and temperature efficiency.

MIN|AVG|MAX	Top (°C)			PMV			DR (%)			C(ppm)			ε_C	ε_T
Case 0	24.0	25.0	25.6	-0.12	0.20	0.40	0.0	0.2	2.8	-	-	-	-	0.91
Case 1	24.9	25.4	25.8	0.08	0.26	0.41	0.0	0.5	6.7	639	680	719	0.98	1.00
Case 2	24.8	25.3	25.7	-0.03	0.18	0.32	0.0	0.7	5.1	953	1001	1069	1.00	0.97

Table 3: Comfort values and ventilation efficiency obtained from the measurements for the tested cases

Overall, an excellent indoor environment quality is obtained for the studied supplied air conditions. A satisfactory operative temperature is obtained, as well as PMV value under 0.5. Furthermore, the draught risk is limited for all cases, not exceeding a maximum value of 6.7% of dissatisfied. The CO_2 concentration reaches an acceptable mean value of 1001ppm for Case 2, where two occupants are in the room. The contaminant removal effectiveness is close to 1, which means that the mixing strategy is efficient, and that the heat and CO_2 generated by the occupants are efficiently disposed of. The contaminant removal strategy is illustrated on Figure 4, where the streamlines from the CO_2 injection coloured in CO_2 concentration are displayed. The injected CO_2 is immediately entrained into the buoyant plume and brought to the upper part of the room, and is then entrained into the air jet where it is cooled down and falls in the occupied zone with a lower CO_2 concentration. In this case, the pollutant sources are associated with the heat sources, the ventilation strategy is hence efficient. However, many indoor air pollutants (VOC, aerosols…) are emitted from passive sources. It would be interesting to determine whether the studied system would be as efficient to remove such kind of pollutants.

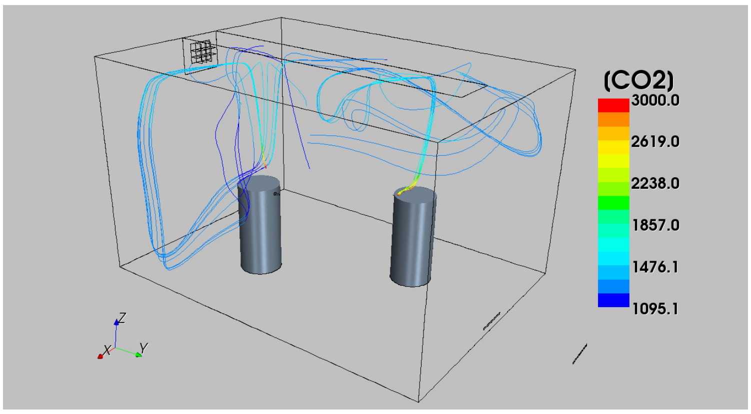

Figure 4: Streamlines colored in CO2 concentration for Case 2

Influence of the heat sources on the airflow pattern

The analysis of the global temperature and velocity values gives valuable information on the influence of the heat sources on the airflow in the room and on the resulting comfort. The mean, median, minimum and maximum values of the air temperature and air velocity for the experimental cases are plotted on Figure 5. All test cases are performed at the same air change rate. It appears that the air velocity in the occupied zone is higher with heat sources (Case 1 and Case 2), due to the entrainment of the ambient air into the buoyant plumes. It also stresses that an increase of the air velocity causes the distribution of the air temperature to be more homogeneous in the occupied zone, which is consistent with a mixing ventilation strategy.

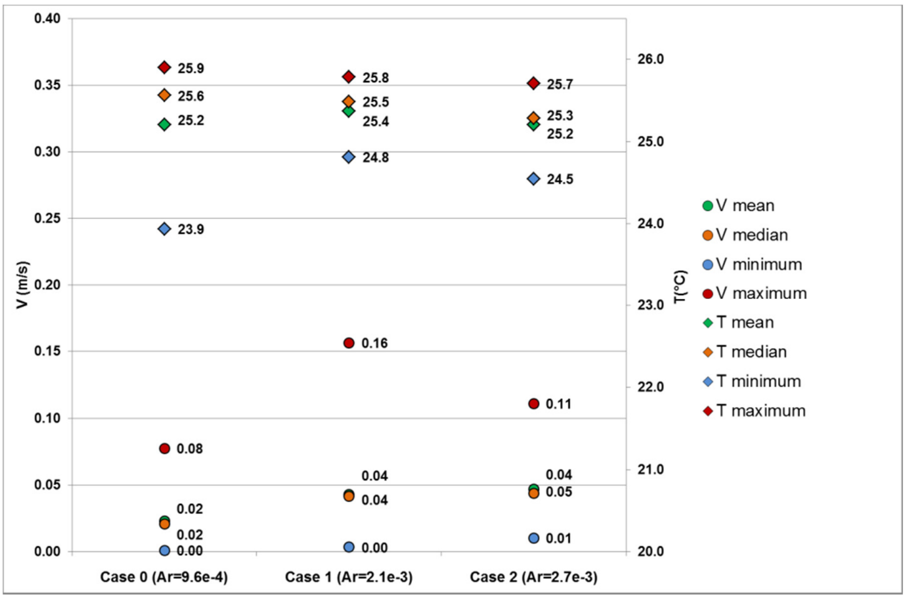

Figure 5: Mean, median, minimum and maximum values of the air temperature and air velocity for the experimental cases

However, the maximum velocity is higher for Case 1 than for Case 2, which indicates that the airflow maybe not be the same for both cases. The velocity streamlines colored in temperature and velocity vectors in the middle plane obtained from the CFD simulations for all three cases are presented in Figure 6 and provide interesting information about the airflow patterns in the room.

In Case 0, the jet is attached to the ceiling thanks to the Coandă effect and goes straight forward, thus reaching the opposite wall. In Case 1 and Case 2, the airflow is tridimensional and follows a clockwise path in the test room. This results from the non-symmetric placement of the dummies. The air jet issued from the diffuser is first deflected to the left by the buoyant plume of the dummy 1. The influence of the heat sources on the airflow is emphasized by the presence of recirculation bubbles above the dummies. The jet then separates from the ceiling and drops slightly in the occupied zone before hitting the opposite wall. It is then deflected backwards toward the occupied zone where it is again entrained in the buoyant plumes and in the air jet. The location and power of the internal gains thus greatly influence the air flow pattern in the room.

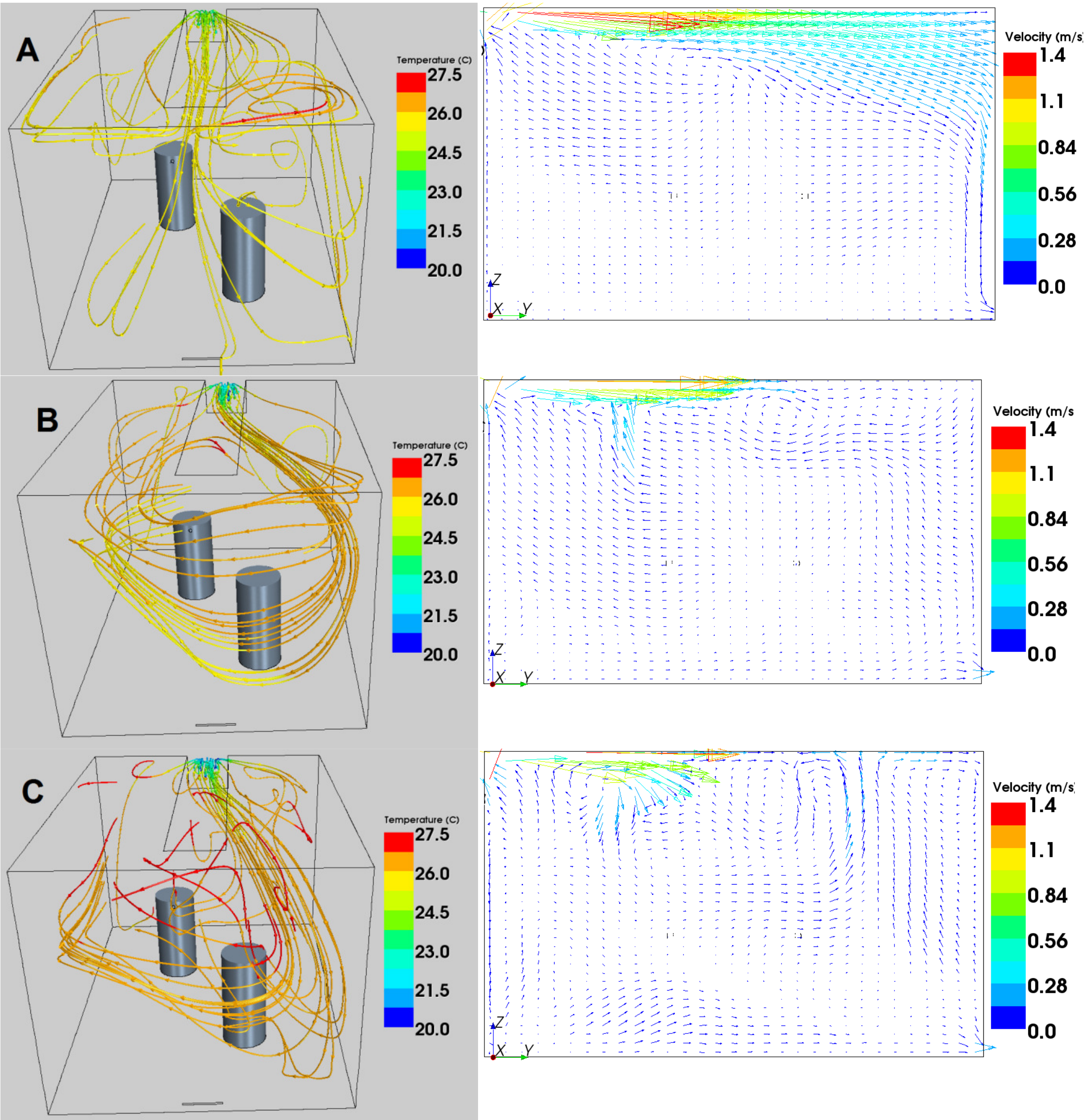

Figure 6: 3D velocity streamlines coloured in temperature and velocity vectors in the middle plane. a) Case 0, b) Case 1, c) Case 2.

Furthermore, it can be seen on the velocity vectors that the jet remains attached to the ceiling longer in Case 0 and Case 1 than in Case 2, which highlights the influence of the supplying conditions on the airflow. However, this influence is hard to pinpoint because of the varying occupancy in the different cases. Hence, a parametric study is performed on the supplying conditions where occupancy is kept constant.

Parametric study on the supplying conditions

The choice of the inlet air flow rate and air temperature is crucial during the room air distribution system design, in order to reach the desired IEQ level. Therefore, additional simulations are performed for cases that were not tested experimentally, in order to evaluate the influence of the supplying conditions on the airflow pattern and on the resulting IEQ. The cooling power brought by the supply, the internal gains, the wall temperatures and the CO_2 source are kept constant (corresponding to Case 1), while different inlet air flow rates and supply temperatures are considered. The conditions of the tested cases are listed in Table 4.

	Qinlet		**Tinlet**	**Toutlet**	**Toz-Tinlet**	**Ar0**	**Re0**
	m^3/h	ACH	°C	°C	°C	-	-
Case 1	57.0	1.66	17.0	25.3	8.3	2.1E-03	17247
Case 1^-	34.3	1.00	11.7	25.3	14.1	1.0E-02	10722
Case 1^+	85.7	2.50	19.7	25.8	6.2	6.8E-04	25497

Table 4: Supplying conditions for the parametric study

The air flow rate should be high enough to maintain an acceptable air quality, but low enough to not cause any draughts in the occupied zone. A way to define the maximum air flow rate is to choose a maximum velocity allowed in the occupied zone, or at the upper limit of the occupied zone (1,8 m), for instance 0.15 m/s [9]. Buoyant plumes may be responsible for draught as well, but the high air velocity resulting from the dummy is not taken into account in the following maximum air velocity values, the studied parameter being the supplied air conditions.

The contaminant removal effectiveness and temperature efficiency for the three tested cases are displayed in Table 5. Compared to Case 1, both Case 1^- and Case 1^+ display a better mixing in the occupied zone, which is translated into a better temperature efficiency.

	ε_C	ε_T	V_{mean}	$V_{max\_1.8m}$	V_{max_OZ}
Case 1	0.98	0.94	0.04	0.14	0.14
Case 1$^-$	=	+ 0.02	+ 0.01	+ 0.13	+ 0.13
Case 1$^+$	+ 0.05	+ 0.02	+ 0.04	- 0.01	+ 0.06

Table 5: Contaminant removal effectiveness, temperature efficiency, mean, maximum and maximum velocity et Z = 1.8m in the OZ, compared to Case 1.

However, a different phenomenon is responsible for the increased mixing in both cases. In Case 1$^-$, the cold air jet separates from the ceiling because of the high Archimedes number and drops into the occupied zone, as shown on Figure 7a. Consequently, the mixing is increased, but so is the draught risk with a maximum air velocity of 0.27 m/s in the occupied zone. In Case 1$^+$, the jet reaches the opposite wall thanks to the higher air flow rate. The latter also explains the increased mixing, caused by an higher entrainment of the ambient air into the cold air jet. But this is then responsible for a maximum air velocity at the ankle level of the occupied zone of 0.20 m/s, which results from the jet hitting the opposing wall and being deflected backwards. Therefore, it appears that a low enough Archimedes number which would ensure that the jet would not fall into the occupied zone is not a sufficient condition to guarantee that a high draught risk may not occur, caused by locally high values of air velocity.

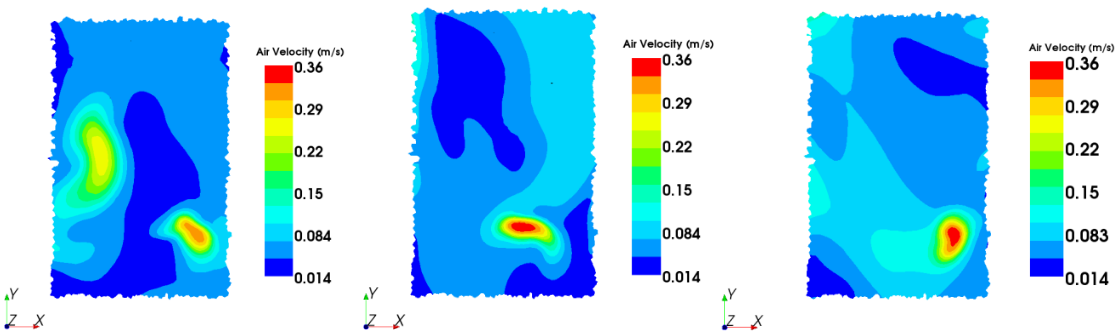

Figure 7: Velocity contours at the upper limit of the occupied zone : Z = 1,8 m. a) Case 1$^-$, Ar0 = 1.0E-2 b) Case 1, Ar0 = 2.1E-3 c) Case 1$^+$, Ar = 6.8E-4

Conclusion

The summer thermal comfort and the ventilation efficiency have been assessed in a test room using a wall-mounted mixing diffuser. Both experimental measurement and CFD simulations have been used. An excellent indoor environment quality has been obtained in the tested conditions. It has been found that the location of the heat sources in respect to the location of the air diffuser plays a major role on the airflow in the room and on the resulting IEQ. This emphasizes that the location and power of the heat sources should be taken into account during the design of an air distribution system. Furthermore, it appeared that predicting the throw of the cold air jet to ensure that it does not fall in the occupied zone, or predicting the average air velocity in the occupied zone with the Archimedes number may not ensure that there would not be locally high values of air velocity responsible for local discomfort by draught. It would therefore be advised to perform new CFD simulations to predict the airflow whenever a new room geometry is considered.

Acknowledgements

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Uncertainties in airflow network modelling to support natural ventilation early design stage

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Abstract

Despite a lot of Integrated Design Process guidelines and procedures have been developed in the last few years, more specific energy design procedures are needed to push the implementation of passive design techniques.

As natural ventilation design influences strongly the building shape and aspect, it has to be considered since the early design stages and its effect should be correctly predicted and proved by means of suitable tools and methods. In this respect, airflow network models seems a promising modeling techniques as they are already integrated in the existing dynamic building simulation tools and they have a quick solver. On the other hand, the simplicity of these models implies uncertainties in a lot of input parameters, first of all the wind conditions. The urban wind environment is a stochastic phenomenon, and as consequence the ventilation performance in a naturally ventilated building changes. An accurate wind analysis should be supported by weather data collected on site and by an external CFD simulation at urban scale. Discharge coefficients and external convection coefficients could be accurately estimated by laboratory tests. This would mean additional design costs (and time) and need of expertise in the field. A key issue of this work is to assess the thermal-airflow model reliability in airflow prediction when accurate estimation of input data is not feasible.

This paper presents an uncertainty analysis performed on a dynamic simulation model of a new office building naturally ventilated during the night.

Full-factorial parametric analysis have been performed to assess the influence of the input parameters on the model reliability. Possible input ranges have been estimated and organized in a tree-structure to investigate the effect of dependent parameters like wind velocity profile, wind pressure coefficients, discharge coefficients, and external convection coefficients.

Significant variations in air change rates are shown that reflect an uncertainty of +/- 2% on total cooling loads in respect to the base case simulation result. No direct correlations between outdoor environmental conditions and air change rates have been found as the discharge coefficients affect significantly the results.

The design proposal supported by quantitative analysis and results prediction uncertainty assessment can be taken into consideration more seriously by the design team. The obtained results can be generally extended to airflow networks with similar airflow paths.

Keywords

Airflow network, uncertainty analysis

Introduction

Since 1993 a lot of Integrated Design Process guidelines and procedures have been developed [1]. They are mainly focused on team work methods, design evaluation and design strategy and, to the knowledge of the authors, no pragmatic energy design procedures have been yet developed to push the implementation of passive design techniques (among which natural ventilation strategies) and to model their effectiveness.

As natural ventilation design influences strongly the building shape and architectural impact, it has to be considered since the early design stages and its effect should be correctly predicted and proved by means of suitable tools and methods. To face this issue, airflow network models seems a promising modeling technique as they are already integrated in the existing dynamic building simulation tools and they have a quick solver.

However, the model implies assumptions on pressure distribution on the building and its interaction with the internal airflows, that cannot be accurately assessed unless additional design costs (and time), that need expertise and possibly laboratory facilities, are invested. A key issue of this work is to assess the thermal-airflow model reliability in airflow prediction when accurate estimation of input data is not feasible.

The paper presents an uncertainty analysis performed on a model of a new office building naturally ventilated during the night, simulated with a dynamic Energyplus. The obtained results can be generalized for further airflow network models.

Building model

Building description

The case study of the present work is one of the new office buildings placed in the new Technology Park in Bolzano (Italy). The building is architecturally conceived as a black monolithic block with a nearly-square plant. It has five above ground floors and an underground floor. The main entrance is on the north side of the ground floor and on the south side there is the expo area. The upper floors will host offices, meeting and service rooms, whereas in the underground floor there will be several conference rooms. In the centre of the building and through the full height, a green patio is designed as a buffer zone to improve indoor comfort and daylighting.

The envelope is a metal-glass façade with a solar shading system in the south façade and a black surface with different strips of horizontal windows in the other facades. The horizontal windows on north, east and west façade are positioned on the inner side of the external wall. In this way, the deep reveal due to the wall thickness and the low height of the windows work as a sun shading system and the glazed part of the façade will not be visible from the outside perspective.

Natural ventilation strategies

A night stack driven cross ventilation was chosen as the most effective configuration that balances performances needs with constrains given by fire compartments, acoustic comfort and privacy needs in the offices during the working hours [2].

To increase the height difference between inlet and outlet openings, connecting floor vents will be applied. This solution fulfils the architect's requests by reducing the movable part in the façade and by keeping free the internal layout of the spaces. Inlet and outlet openings are automatically controlled top hung windows. The floor vents are closed during the working hours to avoid acoustic discomfort and maintain privacy between offices.

The foyer is directly connected to a lightwell and to the hall of every floor and is ventilated through a stack driven cross ventilation to avoid overheating situations.

Due to safety reasons underground floor and expo areas are mechanically ventilated. A small office area in the centre part of the building is single-sided ventilated and connected with the green patio.

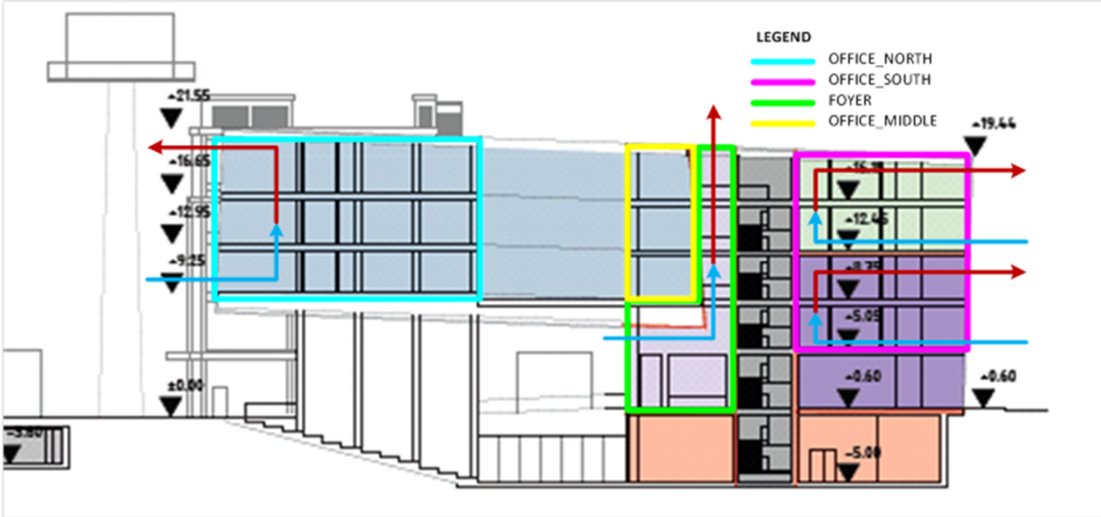

Figure 1: Cross section of the building fire regulation plan with fire compartments, model zones and a scheme of the selected stack-driven cross ventilation configurations for the considered zones.

Energyplus modeling to study airflows

The original Energyplus building model zoning has been re-thought to introduce an airflow network. Particular care has been taken to reduce computational time as the uncertainty analysis requires to perform parametric analysis running several simulation.

The building has been divided into thermal zones with the same temperature and pressures, occupation patterns, internal gains, major exposure, cooling set points. Thermal zones have been further on divided depending on the planned airflow paths and linkages. Building airflow zoning can be more detailed than building thermal zoning as Energyplus airflow network allows only one temperature node per thermal zone.

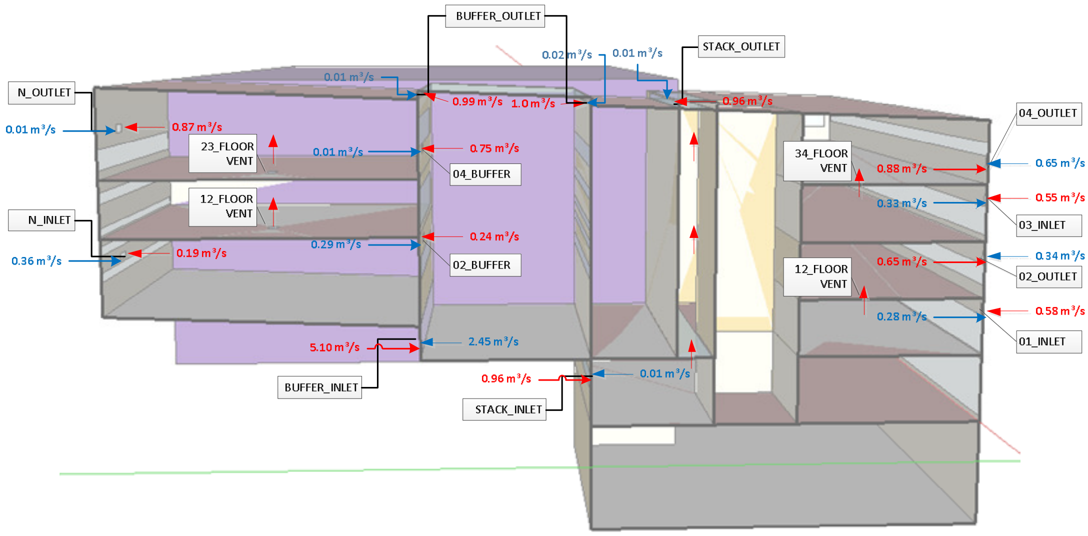

Figure 2: Section of the Energyplus geometry model in sketchup with average airflow rates during summer season. Red and blue arrows represent respectively positive and negative airflow directions.

According to fire compartments and natural ventilation strategies, three airflow networks are set: a first network involves the office block on the south side of the building, the second network involves the atria zones and the stack and a third network involves the office block on the north side of the building.

It is important to detail the window area and the height from the ground level (Table 1). Horizontal window series and floor vents series have been modeled using multiplier as they have the same size and the same height from the ground level. In this way window shape effects are taken into account properly and less input objects are necessary to set window controls.

Zone group	Airflownetwork component name	Opening type	Height [m]	Area [m^2]	Nr of windows	Tot. opening area [m^2]
OFFICE_ SOUTH	01_INLET	Top hung window	7.5	0.65	16	10.40
	02_OUTLET	Top hung window	10.9	0.65	18	11.70
	03_INLET	Top hung window	14.6	0.54	26	14.04
	04_OUTLET	Top hung window	16.8	0.80	26	20.80
	12_FLOOR VENT	Horizontal opening	8.2	1.60	9	14.40
	34_FLOOR VENT	Horizontal opening	15.6	1.60	10	16.00

OFFICE_ NORTH	N_INLET	Top hung window	10.6	0.54	10	5.37
	N_OUTLET	Top hung window	19.0	0.74	10	7.39
	04_BUFFER	Top hung window	17.6	0.71	4	2.85
	02_BUFFER	Top hung window	11.7	0.71	4	2.85
	N23_FLOOR VENT	Horizontal opening	12.4	0.60	9	5.40
	N34_FLOOR VENT	Horizontal opening	16.1	0.60	9	5.40
BUFFER ZONE	BUFFER_INLET	Top hung window	21.5			8.00
	BUFFER_OUTLET	Top hung window	21.5			8.00
STACK	STACK_INLET	Top hung window	3.4	-	-	8.70
	STACK_OUTLET	Top hung window	22.0	-	-	8.70

Table 1: Airflow network surface components area and height from the ground.

An opening factor of 0.5 has been set for top hung window assuming a maximum opening angle of 45°.

This may cause inaccurate shadowing and daylight distribution inside the building, but the computation time decreases. However, daylight study is not one of the purposes of this work and opening area is only a small percentage of the whole glass area. For the same reason a full exterior solar distribution with no reflections has been set. Reflections would have required also no-convex zones, that would have meant setting more zones [3].

Energy Managment System objects are used to introduce simple controls on windows and vents opening [4]. Indoor and outdoor temperature variables have been set as sensors and venting opening factors as actuators. A program was written to activate natural night ventilation between 8 pm and 8 am if the following conditions are fulfilled:

- indoor temperature is higher than 24°C;
- indoor temperature is higher than outdoor dry bulb temperature;
- outdoor dry bulb temperature is higher than 10°C.

External nodes at every floor height and with different orientations have been set to take into account more accurately wind pressure conditions and model stack effects.

Floor vent interzone surfaces references an horizontal opening component with a 90° sloping pane angle. Horizontally pivoted detailed openings are used to model inlet and outlet top hung windows.

An ideal loads air system template, with cooling setpoint temperature of 26°C during working hours, has been implemented to evaluate cooling loads without taking into account the plant system. Infiltration rates have been neglected, as the building tightness required by the local standard is restrictive and will be proven by blowerdoor test.

Simulations have been run from June to September. Average airflow network volume flow rates are shown in Figure 2. Simulation results of the base case model have showed that the airflow direction do not always follow the positive direction of the airflow path: it works in the 86% of activation hours on the upper floors and in the 46% of activation hours in the lower floors. This could be due to the lower opening area in the 1st and 2nd floor. The graph in Figure 3 shows the airflow frequencies through the inlet-outlet components in the south office block airflow network. It could be noticed that low airflow rates are more frequent in the negative airflow direction.

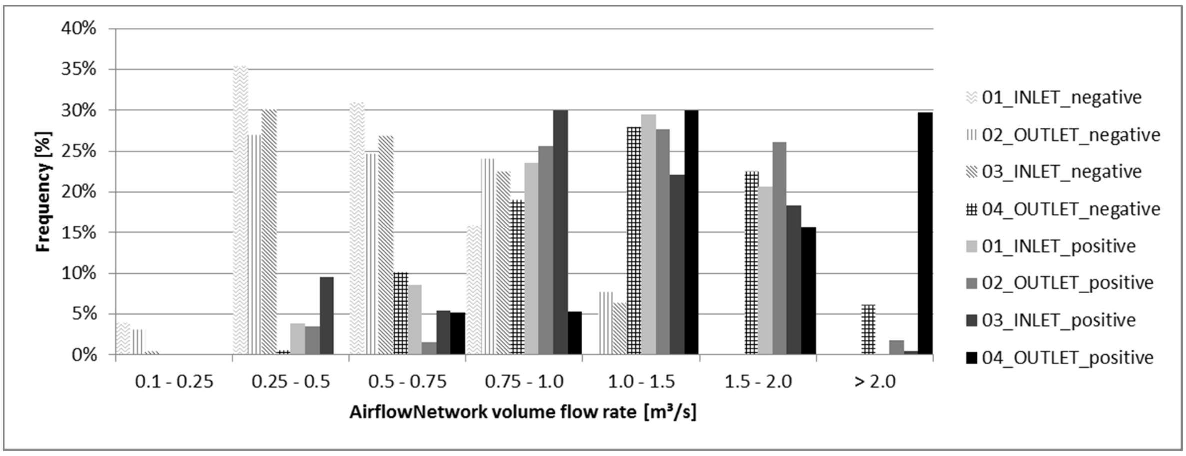

Figure 3: Natural volume flow rate frequency along the airflow network paths in the south office block.

Main uncertainty sources on airflow prediction

Apart uncertainties derived from model assumptions and approximations, the main uncertainty sources in airflow prediction can be ascribed to the input data:

- Leakage's geometry, position and flow characteristics (discharge coefficient)
- External and internal temperatures

- Wind pressures

Opening position determines buoyancy pressures and external wind pressures. Discharge coefficients are considered in the airflow network models as a fixed property of an opening, which depends on its shape and Reynolds number. However, when an opening is installed in an envelope, the actual discharge coefficient may differ from the fixed one. Etheridge D. (2012) [3]provides some boundaries that can be placed on the uncertainties.

Whereas external temperatures are supposed to have low uncertainty, internal temperatures are affected by uncertainties due to the heat transfer - airflow model coupling. Hensen J.L.M. et al. (1995)[6] stated that coupling problems can be overridden by setting a proper time step. Therefore temperature uncertainties are affected mainly by convection coefficients. The external convection coefficients depend on inside-outside temperature difference, wind speed and direction and surface roughness. The internal convection coefficients depend on inside-outside temperature difference, zone airflow regime, surface orientation and heat flow direction. Furthermore vertical temperature profiles (stratification) cannot be reproduced by a multizone airflow model, as the temperature distribution is considered uniform in every zone.

The stochastic features of the urban wind environment affect the naturally ventilated building performances. An accurate wind analysis should be supported by weather data collected on site and by an external CFD simulation at urban scale, to assess wind pressure coefficients on the building envelope.

Parametric studies have been performed on Energyplus Airflow Network model by means of jEPlus program [7] to analyse the sensitivity of the model to different combination of dependent and independent parameters. Possible input ranges have been estimated and organized in a tree-structure (Figure 4) to investigate the effect of these input parameters on airflow rate prediction.

The presented parametric analysis aims at assessing the model uncertainty by:

1. assigning a range and a discrete distribution for each input parameter;
2. executing the model in full-factorial design mode[2];
3. assessing the airflow rates prediction variation rates and its effect on cooling need calculation.

[2] A full factorial experiment is an experiment whose design consists of two or more factors, each with discrete possible values, and whose experimental units take on all possible combinations of these levels across all such factors.

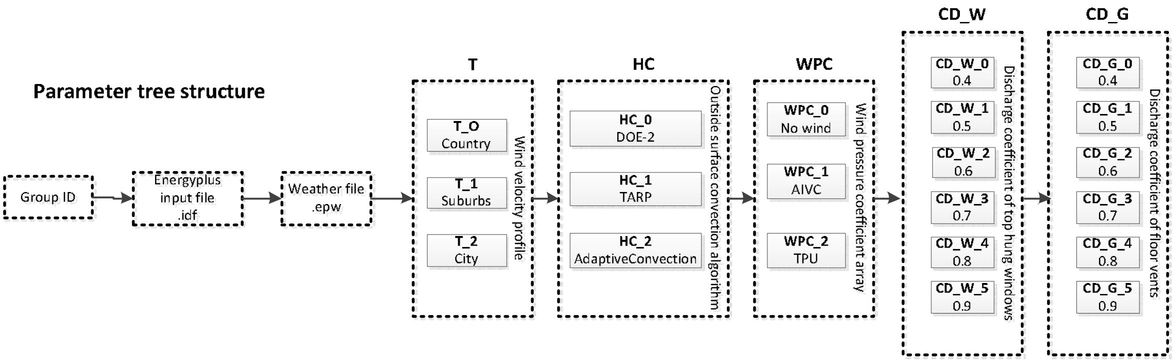

Figure 4: Parameter tree structure set in jEPlus. Each simulation job is a path from the root node of the tree to a leaf of the tree, with each node containing an optional value of the corresponding parameter. As a result, the total number of jobs encoded in the tree corresponds to the total number of paths from the root to the leaves.

Wind velocity profiles

EnergyPlus converts wind velocity weather data through numerical method that considers the differences between the weather station location and the building site, according to:

$$U_{\infty} = V_{met}\left(\frac{\delta_{met}}{z_{met}}\right)^{a_{met}}\left(\frac{z}{\delta}\right)^{a} \tag{1}$$

where z is the height from the ground, z_{met} is the height of the standard meteorological wind speed measurement, and a and δ are terrain-dependent coefficients that determine the wind velocity profile.

Terrain type field in the building object associates different values of wind speed profile exponent and height. Three different wind velocity profiles (Table 2) have been taken into account in the parametric analysis.

Terrain type	**Exponent, α**	**Boundary layer thickness, δ (m)**
Country	0.14	270
Suburbs	0.22	370
City	0.33	460

Table 2: Wind speed profile coefficients. Source: energyplus engineering reference

Outside surface convection algorithm

Energyplus users can select which model equations or values to apply for the exterior convection coefficients calculation. There are five models based on ASHRAE correlations and different flat plate measurements: SimpleCombined, TARP, MoWitt, DOE-2 and AdaptiveConvectionAlgorithm.

Simple combined model returns higher values as also radiation to sky, ground and air is included in the exterior convection coefficient, whereas all other algorithms yield a purely convective heat transfer coefficient. MoWitt model can be applied only to very smooth vertical surfaces. DOE-2 model is a combination of MoWitt and BLAST. TARP model is very similar to BLAST and detailed convection models. The adaptive convection algorithm assigns default equations to surfaces depending on their outside face classification, heat flow direction and wind direction.

The graph in Figure 5 compares the exterior convection coefficients for a vertical surface calculated by TARP and DOE-2 model depending on wind speed, façade position respect to wind direction and different surface roughness. A fix temperature range has been assumed. DOE-2 model is more affected by wind speed than the TARP one.

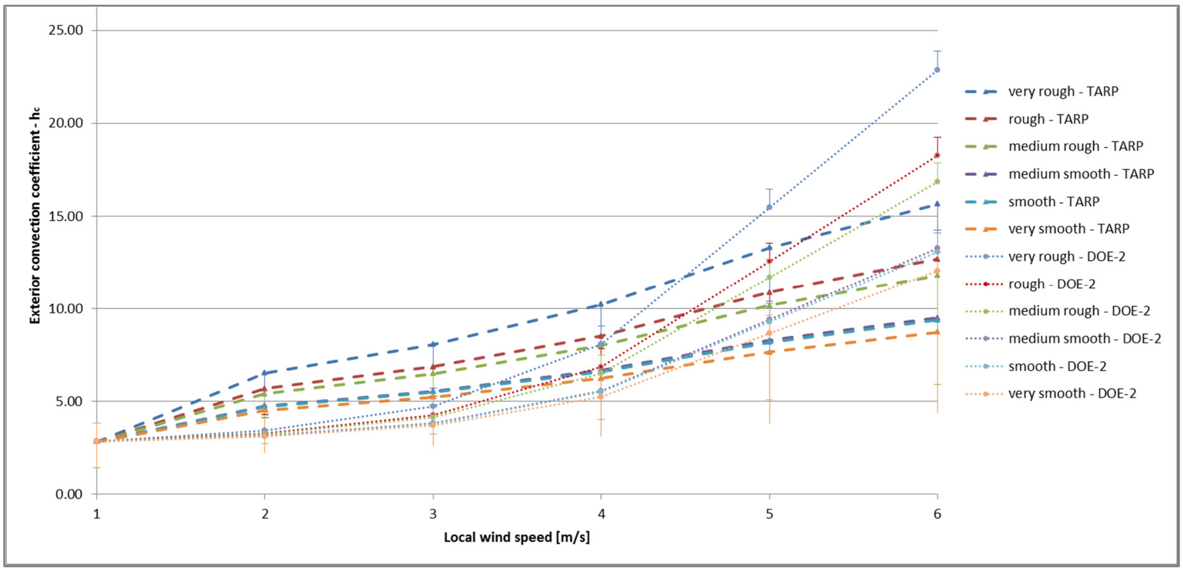

Figure 5: Vertical wall exterior convection model comparison depending on wind speed and surface roughness at a fixed temperature difference

Wind pressure coefficients

Wind pressure coefficients are used to calculate wind-induced pressures on each façade surface during simulations and are defined as the ratio of static pressure to dynamic pressure at a given point on the façade:

$$c_p = \frac{P_x - P_0}{P_d} \tag{2}$$

where P_x is the static pressure at a given point on the building façade (Pa), P_0 is the static reference pressure (Pa), P_d is the dynamic pressure (Pa).

They can be evaluated in detail according to different approaches:

- full-scale measurements
- reduced-scale experiments in wind tunnels
- computational fluid dynamics (CFD) simulations

However these approaches are not suitable to early stage design simulations as they are time-consuming and requires additional costs. Therefore, analytical models and database are most often used in building energy simulation and airflow network tools [8].

For the present case study, surface-averaged c_p have been avoided as more openings per façade are present and the magnitude of their uncertainty is high [9].

C_p coefficient used in the parametric analysis are extracted from two main existing database.

The Air Ventilation and Infiltration centre (AIVC) database [10] is based on the interpolation of data from published material. Wind pressure coefficient datasets are available for low rise buildings with different exposure conditions and with different length to width ratio. The coefficient sets used for the analysis are extracted from the low rise building database with length to width ratio of 1:1 and surrounded by obstructions equal to the height of the building.

The Tokyo Polytechnic University (TPU) aerodynamic database [11] is a web based database on wind tunnel experiments on 12 test cases including low and high rise buildings with varied eaves. The coefficient sets used for the analysis are extracted from the low rise building database with flat roof, height to width ratio of 1:4 and length to width ratio of 2:2.

A wind pressure coefficient array of null values has been introduced in the parametric analysis to evaluate the influence of wind pressures on output results.

Discharge coefficients

The discharge coefficient is required as input in airflow models and is defined by Equation 3.

$$c_d = \frac{q}{A}\sqrt{\frac{\rho}{2\Delta p}} \tag{3}$$

where q is the volume flow rate across the opening (m^3/s), A is a defined open area (m^2), ρ is the air density (kg/m^3) and Δp is the airflow difference across the opening (Pa).

Airflow predictions are linearly dependent on discharge coefficients and therefore they imply directly proportional results uncertainty.

The Equation 1 is applied to openings installed in a surface separating two much larger spaces in still-air conditions, with uniform and equal densities. In practice flows through openings are generated by wind and buoyancy forces. The wind modifies the external flow field, whereas the buoyancy forces cause different air densities. As Etheridge D. (2012) stated [4], the installation effects are negligible for air vents and small windows in case of low velocity ratio and for large stacks in case of inward flow only. In particular, this effect depends primarily on the magnitude of V/u_m (typical values range from 1.5 to 9), where V is the cross flow velocity and u_m is the spatial mean velocity through the opening.

As much as opening angles is smaller the uncertainty is higher, because leakages along all other sides of the opening account for a relative large part of the opening area.

It is to underline that these effects are dependent on wind velocity and direction. Therefore the parametric analysis has to be applied simultaneously to discharge coefficients, wind velocity profile and wind pressure coefficient sets on the façade.

Johnson et al. (2012) compared airflow network predictions with measured airflow values and found that the discrepancy in the predicted value is due to inaccuracy in the discharge coefficient, which was an estimated value [12].

In the case of buoyancy driven cross ventilation, some measurements were performed on scale model tests [13] but no full-scale model have been yet analysed, especially in the case of horizontal openings.

Two different discrete variables have been set for the floor vents discharge coefficient and for the top hung windows discharge coefficients in the parametric analysis.

Stated Heiselberg's experiments [14], it has been assumed that discharge coefficients have a value bigger than 0.4.

Results

The simulation model has been run in full-factorial mode for the airflow network involving the office block on the south side of the building.

Analysing the resulting air change rates at fixed outdoor wind and temperature conditions (Figure 6), significant variations with standard deviations up to 0.6 occur.

The lower values in the graph are referred to samples with null wind pressure coefficients. The upper values in the graph are referred to samples with wind pressure coefficients from AIVC database. The smaller variations are due to different discharge coefficient combinations. In presence of wind, discharge coefficient effect is higher than in absence of wind.

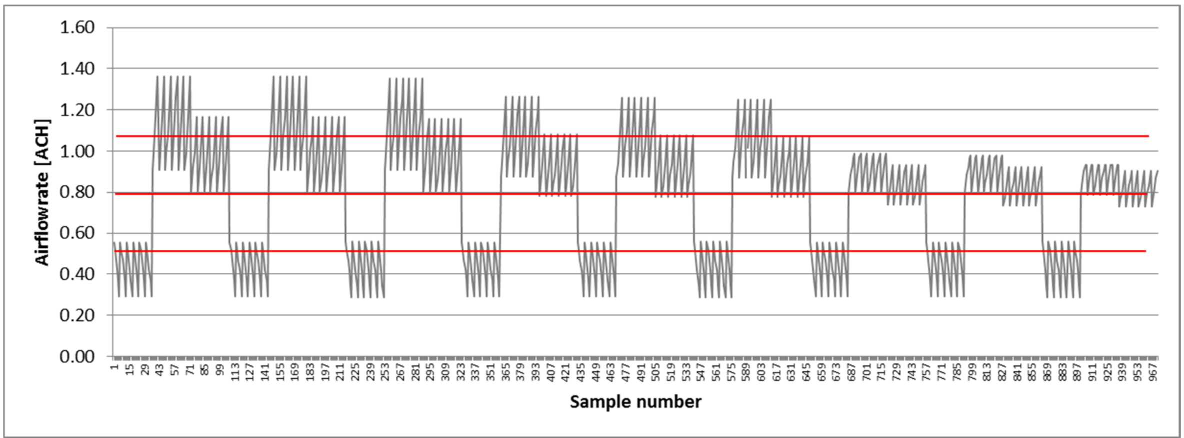

Figure 6: Resulting airflowrates at 22/06 h 21:00 from parametric analysis with 967 samples. Row lines represents average ACH +/- standard deviation.

Figure 7 shows the airflow rates calculate for the base case with error bars that represent the variations calculated through the parametric analysis.

Total cooling loads of the office block in the south part of the building have been calculated with the result that the variations in air change rates reflect an uncertainty of +/- 2% on total cooling loads.

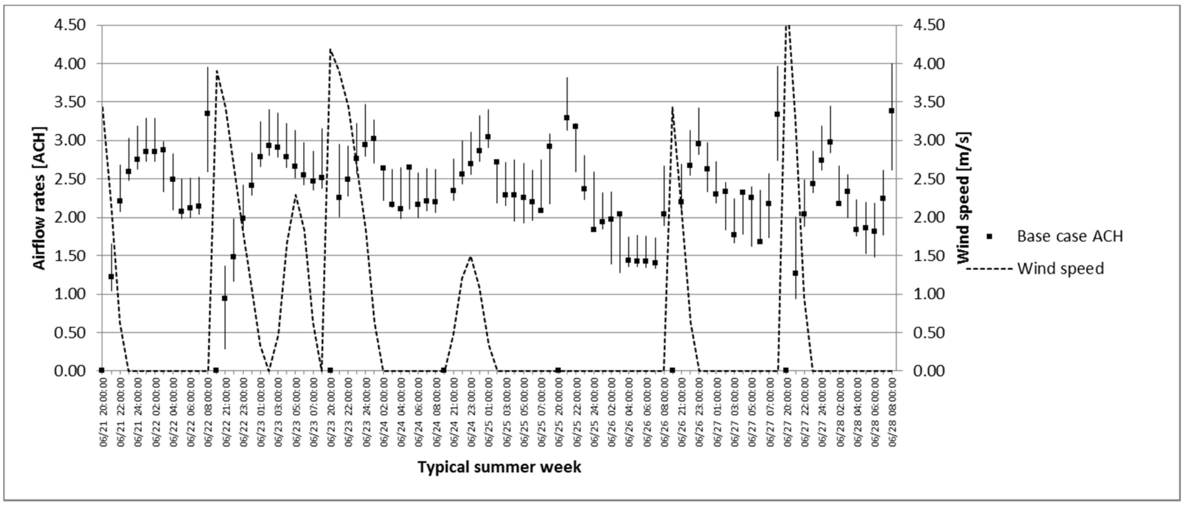

Figure 7: Airflowrates during the typical summer week in the first floor of the building south office part. Points represents the base case results and bars represent the airflow rate possible uncertainties.

Scatter plots in Figure 8 have been performed to find correlation between external environmental conditions and airflow rates. No direct correlation has been found, but it can be noticed that in absence of wind or in case of low outdoor temperatures, the range of standard deviations is larger. This means that, discharge coefficients may affect significantly the results.

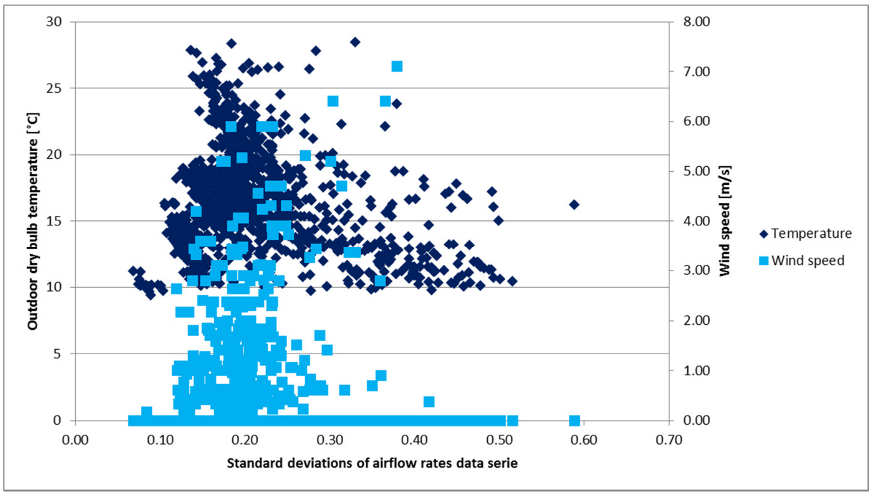

Figure 8: Scatter plot of standard deviations of airflow rate data serie related to wind speed and outdoor dry bulb temperature.

Conclusion

These results would be of particular interest for the natural ventilation design support during early design phases and can be generally extended to airflow networks with similar airflow paths.

Significant variations in air change rates are shown that reflect an uncertainty of +/- 2% on total cooling loads.

No direct correlations between outdoor environmental conditions and air change rates have been found as the discharge coefficients affect significantly the results. However, in absence of wind or at low temperatures the standard deviations have a wider range of variability.

Airflow network simulation results are also useful to control the efficacy of the natural ventilation strategy proposed. Simulation results of the base case model have showed that the airflow direction follow the positive direction of the airflow path in the 86% of activation hours on the upper floors and in the 46% of activation hours in the lower floors. Inlet and outlet opening area at 1st and 2nd floor should be increased.

Further developments of the process are planned to optimize opening area and opening controls to prevent natural ventilation strategy dysfunctions.

Thanks to this quantitative analysis support, the mentioned natural ventilation strategies can be evaluated in a more rigorous way by the design team, for the building and architectural choices in the early design phases.

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Hybrid ventilation – the ventilation concept in the future school buildings?

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Abstract

Hybrid ventilation (HV), as a combination of automated natural ventilation (NV) and balanced mechanical ventilation (MV), provides opportunities to use the advantages of both ventilation systems during the seasons in order to reduce energy demand and at the same time obtain comfortable indoor climate.

For this study a comparison of NV, MV and HV systems applied to an existing school building has been made by means of detailed modeling in the widely adopted simulation program IESVE. The energy demand for heating and ventilating the building in the different ventilation modes was calculated for three key European cities; Munich, Copenhagen and London. Control strategies were set to obtain the same indoor climate for all ventilation systems, and the indoor climate classification was made according to EN15251 [1] .

The overall ambition in the study has been to make a very realistic modeling of state-of-the-art MV and NV systems and apply these systems to an existing school building renovated to fulfill expected energy performance requirements for 2015 buildings.

It is noteworthy that the results show that the energy performance of the MV and NV systems are nearly the same in terms of primary energy, while it is showed that HV enables energy savings of 44-52% compared with MV and NV. This corresponds to a total primary energy consumption for the NV and MV of 18-21 kWh/m^2 per year in all three locations, while the HV consumption was only 9-10 kWh/m^2 per year.

Calculation of the total investment of the different systems including maintenance, operation (electricity and heating) and capital cost (products and installation) showed that in the first year MV was found to be 2.5 to 4 times more expensive than NV. By selecting HV 25% of the

total investment could be saved compared to a MV system. The differences between the systems were found to be the same in a 20 year timeframe

Keywords

Hybrid ventilation; natural ventilation; mechanical ventilation; ventilation in schools; comparison of different ventilation strategies; Energy; IAQ.

Introduction

From the literature it is known that the combination of automatic NV and MV is a promising opportunity to achieve significant energy savings in buildings while maintaining a comfortable indoor climate. HV might as such be a key technology for designers in need of means to fulfil the ever stricter energy requirements while at the same time provide the user with a healthy and comfortable indoor climate.

Many studies of HV are available in the literature. The Reshyvent project [2] investigated HV in residential buildings, while the Hybvent project [3] studied the applicability for non-residential buildings. A number of case studies investigated in the international project IEA ECBCS-Annex 35 [3] show that there can be achieved significant energy savings in hybrid ventilated buildings, especially through reduction in fan and cooling energy demand. The case studies for school buildings show that the HV system saves 17-55 % in a year compared to a solitary mechanical system. Examples also include Cron & Inard [4] who investigated classrooms and found the best results for HV in the warmer regions, compared to MV without heat recovery, and in the warmest regions, compared to NV with heat recovery. Emmerich [5] found that heating demand is reduced most in cold regions, and fan electricity is reduced to a maximum in warm regions. Sowa & Karas[6] found a reduction of about 60 % for heating demand and about 40 % for fan electricity in a HV simulation for a real school. Heikkinen et al. [7] who investigated ventilation concepts for a school in Finland, only found a very little possible reduction in heating demand, but also a reduction of 70 % in fan electricity.

Thus, the literature contains a number of publications and in general, HV is demonstrated to result in significant energy savings. The present study investigates whether this conclusion will also be valid in schools in the near future (2015), using state-of-the-art MV and NV systems and a school building fulfilling expected 2015 performance requirements. In order to give a true comparison between the energy performances of the systems, nearly identical indoor air quality and thermal climates in the buildings have been created. The simulations are carried out for three large European cities with different climates; Copenhagen, London and Munich. The study also calculates the expected CO_2 emissions and the economical costs from selecting the different systems.

School Building geometry, properties and location

Building layout

The school building has one storey with eight classrooms, four on each side of the corridor and is oriented North/South. The ceiling height varies from 2.8 to 4.5 m for all ventilation types and the area of one classroom is 76 m^2, the volume is 278 m^3. The geometry of the building can be seen in Figure 1.

Construction properties

Some of the construction properties attached to the models are listed in Table 1.

The glass ratio of the outer façade is 45 % and the g-value is 0.63 and light transmittance is 0.74. Only the lager windows have an external sun screening with a shading coefficient of 0.2.

Figure 1: Layout of the school building.

Building Element	U-Value [W/m^{2}K]
Ground Slab	0.08
Exterior Walls	0.12
Roof	0.08
Windows	0.9-1.1

Table 1: Construction properties

Internal heat loads

There are 28 students and one teacher in each classroom, resulting in an occupancy density about 2.6 m^2/person, which is a typical occupant density for schools. There is assumed 95 % occupancy during lessons (Monday to Friday from 8 am to 2:50 pm) taking into account absent persons. Vacation time is 12 weeks per year in total (week 7, 14, 20, 26 - 31, 42 and 51 - 52). The occupancy during vacation is set to 10 % from 8:00 am to 2:50 pm from Monday to Friday, which takes into account summer courses or maintenance. There is no occupancy during weekends.

Each person has a heat load of 75 W sensible heat and 50 W latent heat corresponding to an adult with an activity level of 1.2 met. This assumes a heat emission of 70 W/m^2 skin surface and a skin surface of 1.8 m^2. Children with a lower body mass normally also have a higher level of activity of about 1.4 met (81 W/m^2skin surface). Assuming a skin surface of 1.5 m^2 per child, the heat emission for all persons is quite the same.

Each student and teacher is expected to have a computer, which is switched on 50 % of the time during occupancy. There are typically not so many computers in a classroom today, but it is expected that the use of computers will increase in the future.

The lighting (fluorescent lighting) shall provide a luminance intensity of 300 lux at the table and has a maximum heat load of 15 W/m^2, which corresponds to an effective lighting system.

Outdoor climatic conditions

The locations chosen for the comparison are Copenhagen, Munich and London. These three cities are typical European cities with different climates and therefore different possible opportunities for HV. Copenhagen has a cold winter and a cool summer, whereas Munich has a colder winter and a warm summer. London, located close to the sea, has a maritime climate with a mild winter and a cool summer.

Requirements on indoor air quality and thermal comfort

Many studies have shown that existing schools have a poor indoor climate with CO_2 levels sometimes exceeding 2-4,000 ppm [8]. These levels are clearly adversely affecting the learning ability of the school children [9,10,11] and these levels must be improved. However, general adoption of the current Category II requirements in EN15251 with a maximum CO_2 concentration of 900ppm seems unrealistic in schools for two key reasons: First, the air exchange rate for a typical 60 m2 class room with a room height of 2.8m and 29 persons needs to be at least 6-7 ACH. This will create problems with air speeds in the comfort zone in the majority of existing schools.

Second, it is also noted that the financial abilities of the public authorities in most countries does not support such strict requirements. In fact, they could prove to be a barrier against improving the indoor climate in existing schools simply because the systems become too expensive.

The classification of the thermal comfort and indoor air quality in the buildings are based on EN 15251. For the school building Category III with an acceptable level of expectation is applied for the assessment of indoor climate.

Dimensioning of ventilation systems

To maintain the air quality according to Category III of EN 15251 the necessary air flow rate was calculated by the air flow rates per m^2 given in the standard for persons in a classroom and low emissions from the building. This results in a flow rate of 2.4 l/sm^2 and a total air flow rate of 180 l/s, 648 m^3/h. The total flow rate for all 8 classrooms is 1,440 l/s or 5,148 m^3/h. For maintaining temperature different air change rates were tested in the simulation. Due to these results a maximum air exchange of 4.6 per hour was chosen for summer and night ventilation. This is a flow rate of 360 l/s or 1,296 m^3/h for one classroom and 2,880 l/s or 10,368 m^3/h for all classrooms.

Natural ventilation

For the NV every second high level window on both sides of the room can be opened with motors to realize cross ventilation. The resulting openable window area for the automated windows is 4.1 m^2 and 5.4 % of the room area.

A temperature difference of 1 K and 5 K between inside and outside is resulting in a nearly 4 and 9 fold air exchange, respectively. A wind speed of 0.5 m/s and 1 m/s is resulting in a 5 and 10 fold air exchange rate, respectively (calculated according to the British Standard Method [12]. Outdoor conditions with 0.5 m/s wind speed and a temperature difference above 1 K should be available during most time of the year for all three locations and are also adequate for temperature maintaining ventilation in summer.

Mechanical ventilation

For the MV in the school building four smaller decentralized units are utilized. The system was dimensioned for the maximum air flow rate according to air quality and indoor temperature (10,368 m^3/h). The specifications of the four units are selected amongst available market products, resulting in a total flow rate of 15,680 m^3/h.

The pressure loss for the supply and the exhaust pipe system is only 80 Pa for the supply system and 80 Pa for the exhaust system. The filter classes were F7 for supply air and F5 for exhaust air causing an additional pressure of 40 Pa due to dirt. The SFP value for the whole system is 993 J/m^3 - which probably is among the best available on the market currently. There were assumed no additional heating unit and no cooling unit. As the 'Demand Controlled Ventilation' is working with a constant pressure loss in the main pipes the setting for the external pressure was hold constant.

The heat recovery system is a state-of-the-art counter-flow plate heat exchanger with low-energy deicing function and the sensible heat effectiveness in the simulation was set to 92 %, which is the temperature efficiency including the effects of the motor heat at 1800 m^3/h, 75 % of the design flow rate.

Hybrid ventilation

For the HV there are applied two decentralized units dimensioned for the air flow rate only according to air quality (5,148 m^3/h). To maintain indoor temperature in summer NV is utilized with a flow rate of 10,368 m^3/h.

The pressure loss in the MV system for the supply and the exhaust pipe system is then about 132 Pa for the supply system and 143 Pa for the exhaust system. Pressure loss from filters, sensible heat effectiveness and heating/cooling unit is the same as the MV. SFP value for the whole system is 1135 J/m^3.

Calculation method

The energy demand and the indoor climate of the building were simulated in the widely adopted simulation the program VE-Pro (version 6.4.0.7, Integrated Environmental Solutions Limited, Glasgow, UK). This program has a special device for calculating more complex HVAC systems (ApacheHVAC) and also a very reliable calculation tool for NV (MacroFlo), which is able to calculate NV and effects from wind turbulence on air exchange, considering special features like the aspect ratio and sash type of the opening. The calculation was done in 1 min steps to achieve realistic results for natural and especially natural pulse ventilation. The results are derived from 6 min averages of the calculation.

For the assessment of indoor climate, CO_2 levels inside the building were used as indicator for indoor air quality and operative room temperature was used as indicator for thermal comfort. The values were obtained during occupancy in one representative room, and the requirements to the thermal comfort and indoor air quality were based on EN15251.

Control strategies

The transfer of the control strategy to the simulation models was done as close as possible to WindowMaster's control strategy. Sometimes changes were necessary due to the restrictions of the simulation software or to obtain a similar thermal comfort and indoor quality.

Natural ventilation

NV in this investigation is defined as automated NV through high level windows on both sides of the rooms. The windows are opened and closed to a specific amount with small motors. The opening width is defined by a controller, which uses indoor and outdoor climatic parameters to calculate the appropriate opening width. This precise opening is necessary, because the resulting air flow rate is not only depending on the climatic parameters, but also very much on the opening width of the windows. A precise air flow is necessary to avoid too high ventilation rates, which cause additional heat losses or a bad thermal comfort due to low temperatures or a high draught ratio, and to provide a good air quality at the same time.

There is implemented three different opening strategies; continuous ventilation with a varying opening degree, pulse ventilation with the maximum opening degree calculated due to weather for a short time and night ventilation. The first strategy is utilized for control of air quality during the whole year and indoor temperature in summer. The second strategy is only for additional control of indoor air quality during winter and transient times, where the opening width for continuous ventilation is restricted due to comfort reasons. The third strategy is utilized for an additional cooling of the rooms in summer. In addition, the windows are opened to maximum after occupancy to ventilate the rooms completely with fresh air until outdoor air quality is reached.

Mechanical ventilation

The flow volume of the MV is defined due to improvement of indoor air quality and reduction of overheating. Therefore the maximum flow volume is utilized, when either the flow rate due to carbon dioxide level or the indoor air temperature rises above a certain point. Furthermore night ventilation is only active during the warmer periods

Hybrid ventilation

The HV control strategy is a combination of the natural and mechanical control strategy. The main strategy is to use the best aspects of both systems in order to reach the best possible values for energy consumption and indoor quality.

During winter season only MV is activated as the heat recovery of the system helps saving energy to heating. NV has best results during summer where good indoor air quality and temperatures easily can be reached. In addition, the motors of the windows need much less electricity than the fans for MV and the flow rate can be raised only by opening the windows a little more without using additional energy. This is also the benefit from NV during night time.

During the transient season it depends most of the times on internal situations, if MV or NV is the best solution. Therefore the system chooses automatically between the two systems depending on indoor temperature as indicator for heating or possible cooling demand.

Heating, shading and lighting

The heating is activated from October to May. The heating is set to avoid too low temperatures in accordance to Category III during occupancy (8 am 2:50 pm). During hours with no occupants in the building the set-point is 18°C.

The setting for the shading devices is controlled due to outdoor and indoor parameters. This is done to avoid overheating, which may affect indoor temperature and thermal comfort also a few days later. The device rises with a wind velocity above 12 m/s and/or an outdoor temperature below -6°C to avoid damages on the devices. The device lowers with a solar radiation above 100 W/m^2 and if the indoor temperature is above 23.5°C.

The dimming of the artificial light is controlled due to occupancy and to keep 300 lux in the rooms.

Results

Temperature and CO_2

The results of thermal comfort and indoor air quality was evaluated for one south facing classroom as there only were found very small differences between a north- and south facing classrooms. Figure 2 shows the relative frequency of indoor temperatures during occupancy in accordance to the categories in EN15251. This is displayed for all three ventilation types in each location.

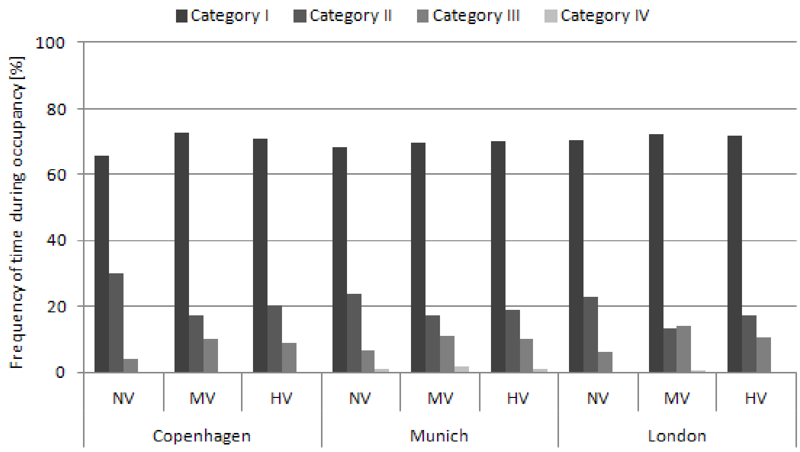

Figure 2: Relative frequency of indoor temperatures during occupancy in accordance to EN15251

Only very small temperature variations were found for the different locations. This implies that each of the three ventilation systems has the same thermal comfort level. A similar picture was found when comparing the CO_2 levels. The result showed that Category II could be reached in 45-55% of the time depending on the ventilation system, while the remaining time was fulfilling the requirements to Category III.

Primary energy

For the calculation of the total primary energy demand (sum of heating and fans electricity demand multiplied with primary energy factors) the nationally adopted primary energy factors have been used for the different locations; Munich (district heating: 0.7; electricity 2.6), Copenhagen (0.8; 2.5) and London (1.2; 2.92).

Figure 3 shows the primary energy consumption. Comparing the primary energy demand it can be seen, that heating demand can be reduced by nearly 70 % for HV compared to NV. Fans electricity can be reduced with 75 % for HV compared to MV. Total primary energy is almost the same for MV and NV, but can be reduced up to 50 % for HV.

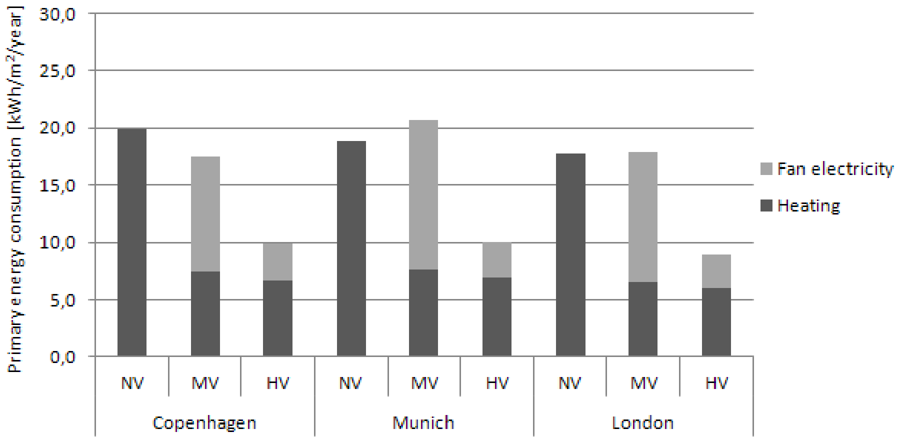

Figure 3: Primary energy consumption

CO_2 emission

Calculation of the CO_2 emissions are based on the following figures; Munich (district heating 200 g/kWh and electricity 606 g/kWh), Copenhagen (104; 425) and London (206; 517). CO_2 emission due to electricity and heating is almost the same for NV and MV. Depending on the location the CO_2 emission ranges from 2.6-5.4 kg CO_2/m^2 per year. HV makes it possible to reduce this CO_2 emission up to 50%.

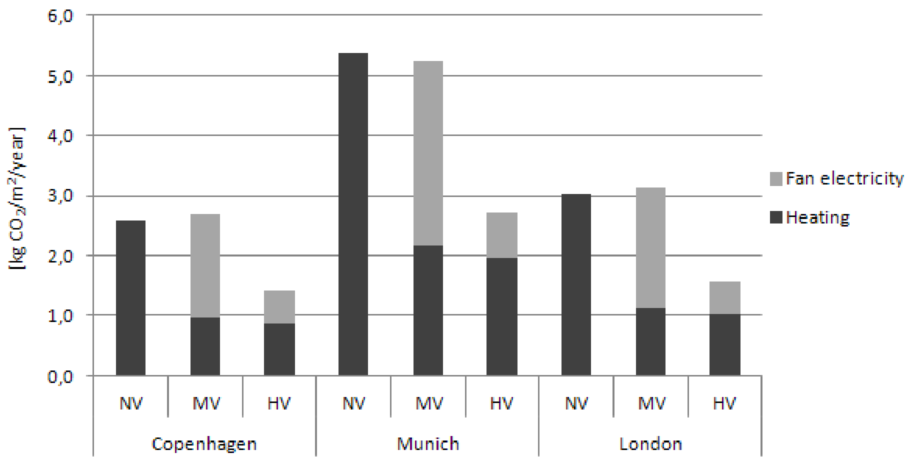

Figure 4: CO_2 emission

Cost

The total investment of the different systems has been evaluated including costs for maintenance, operation (electricity and heating) and capital cost (products and installation) for the first year of operation (Figure 5) and during a period of 20 years

(Figure 6). The prices are calculated by WindowMaster in close collaboration with a Danish ventilation contractor [13].

The maintenance cost for HV is almost the same as MV. Choosing NV this cost could be reduced with 70%. No significant difference was found between NV and MV for the operation cost during the first year. However, using HV this cost could be reduced with 50%. One of the major differences of the three systems is the capital cost. Here it was found that a MV system is more than four times as expensive as a NV system. For HV this was only a factor three. As a result of this big capital costs the total investment the first year is still in favour of NV with a factor four compared to MV and a factor three compared to HV. HV was found to be 25% cheaper than a full MV system.

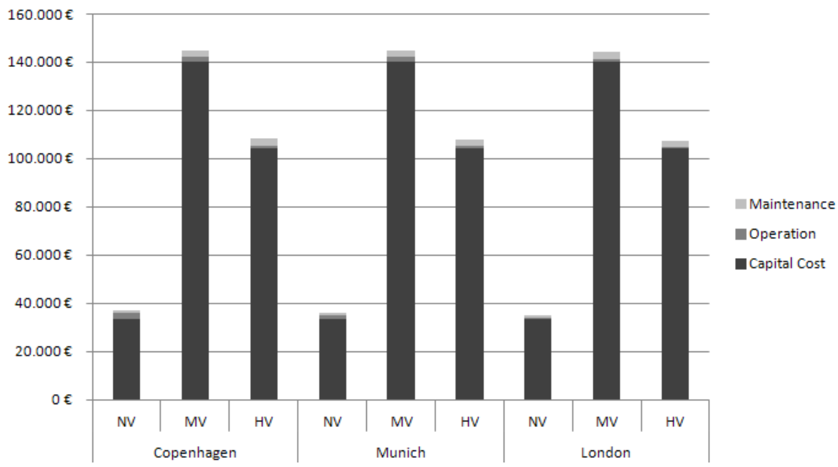

Figure 5: The total investment during the first year of operation

The total investment over a time period of 20 years showed almost the same pattern as the first year of operation. MV was found to be 2.5 to 3 times more expensive than NV on the total investment during a 20 year period. 25% could be saved choosing a HV system compared to a MV system.

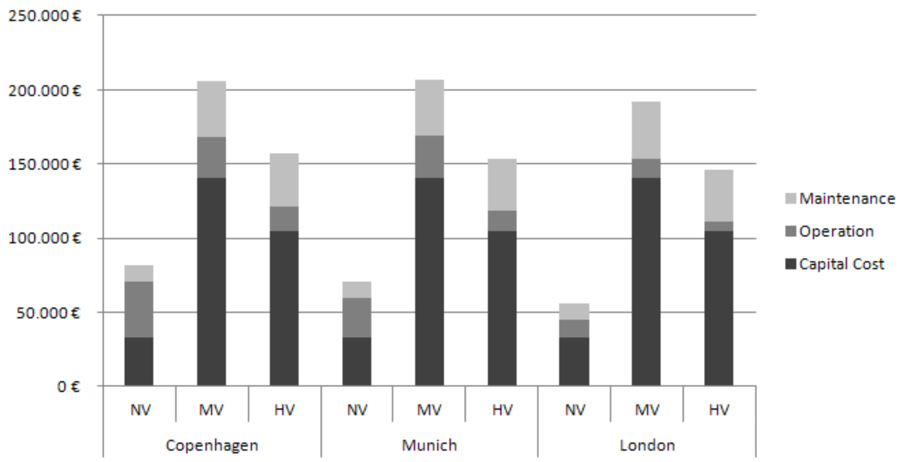

Figure 6: The total investment during a period of 20 years

Discussion

The main ambition for the control strategies is to obtain the desired thermal comfort and indoor quality defined in EN15251 Category III and to obtain very similar indoor air quality for all three ventilation types. This is necessary for a useful comparison of energy demand resulting from the ventilation types. It should be noted that better indoor air quality and thermal comfort could have been obtained for all of the three ventilation systems if other requirements for CO_2 and temperature was chosen.

The transfer of the control strategy to the simulation models was done as close as possible to WindowMaster's control strategy. Sometimes changes were necessary due to the restrictions of the simulation software or to obtain a similar thermal comfort and indoor quality. It is believed that these simulations still are in accordance to the WindowMaster control system.

HV control strategy is a combination of the NV and MV control strategy. The main strategy is to use the best aspects of both systems, in order to optimise the balance between indoor climate and energy consumption. This is possibly the greatest challenge with HV and it is therefore necessary to have a control strategy that can take this into consideration.

During winter and summer the HV control strategy is almost straight forward. MV has the best results during cold periods, when there is a heating demand. The heat recovery of the system then helps saving energy of the building. During the summer period is it NV that has the best results. Good indoor air quality and thermal comfort can easily be reached without using any fan energy. The flow rate can be raised only by opening the windows a little more whiteout using additional energy. NV has also the ability to benefit from night ventilation without using any additional energy.

The transient season is, however much more complicated and the most of the time it depends on the internal situation, if MV or NV is the best solution. Therefore it is very important to have an automatic system that can choose between the two systems depending on indoor temperatures as an indicator for heating or possible cooling demand. This is perhaps not that complicated. The complicated part is to make for instance MV stop and then start up the NV system due to the fact that the internal environment has change throughout the period where MV has been used. This strategy has been developed in these calculations.

Conclusion

The total primary energy demand (sum of heating and fans electricity demand multiplied with primary energy factors) for the NV and MV systems ranges from 18-21 kWh/m^2 per year in all three locations. For HV the total primary energy demand was only 9-10 kWh/m^2 per year.

The result shows that HV enables energy savings of 44-52% compared with MV. Compared to NV an energy saving from 46-50% could be reached. The heating demand can be reduced by nearly 70% for HV compared to NV. Fans electricity can be reduced with 75% for HV compared to MV. Total primary energy is almost the same for MV and NV, but can be reduced up to 50% using HV.

One of the major differences was to be found in the total investment of the different systems including maintenance, operation (electricity and heating) and capital cost (products and installation). Looking at the first year of operation and during a period of 20 years MV was found to be 2.5 to 4 times more expensive than NV. By selecting HV 25% of the total investment could be saved compared to a MV system.

The results demonstrate clearly that HV should be considered for schools in addition to NV and MV. Overall the HV makes it possible to save money for heating and electricity during operation time and to save up to 50% of the CO_2 emissions.

Acknowledgements

First and foremost, we would like to thank WindowMaster for the valuable guidance, advice and knowledge sharing. Besides, we would like to thank the *Fraunhofer Institute for Building Physics* for providing a good environment and facilities to complete this project.

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The discharge coefficient of a centre-pivot roof window

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Abstract

Accuracy in estimation of airflow through windows is the key parameter for modelling and designing of naturally ventilated buildings. The flow through windows is usually described by the orifice flow plate equation. This equation involves the discharge coefficient. In practice, often a constant value of discharge coefficient is used. The constant value of discharge coefficient leads to deceptive airflow estimation in the cases of centre-pivot roof windows. The object of this paper is to study and evaluate the discharge coefficient of the centre pivot roof window. Focus is given on unidirectional flows i.e. inflow and outflow. CFD techniques are used to predict the airflow through the modelled window. Analytical orifice flow equation is used to calculate the discharge coefficient. Results are compared with experimental results. It is concluded that the single value of the discharge coefficient leads to ambiguous estimation of airflow rates. The discharge coefficient decreases with increase in flap opening area. The discharge coefficient also depends upon the flow direction

Keywords

Centre-pivot roof window, discharge coefficient, CFD

Introduction

The amount of air entering through natural ventilation (or in hybrid systems) is extremely difficult to predict accurately as airflow depends on unknown wind and buoyant effects. In practice, the orifice flow equation is used to compute the airflow through the intentional openings and windows. The discharge coefficient (C_D) in this equation is usually taken as constant. The constant value of the discharge coefficient is valid only for constant opening areas [1, 2]. Hence, the use of constant value of C_D for operable windows leads to deceptive results. The exactness of the C_D can have a significant impact on the ability of a mathematical model to predict the airflow rates [2, 3].

There is a need of evaluation of C_D of operable (i.e. with flap) windows. Operable windows are broadly used in residential buildings for ventilation. Scientific literature on façade windows is somehow available. However, the literature on roof windows (especially center-pivot roof window) is not much discussed. This paper focuses on the discharge coefficient of a center-pivot roof window.

Airflow rate through the opening is the integral of velocity over the opening area i.e. $q = \int v dA$ [4]. In practice it is difficult to do this integration. Therefore an alternate way has to be adopted. The airflow passage through the opening acquires the shape of a jet [5]. Therefore the volume flow rate at the vena contracta of the jet is the actual volume flow rate through the opening. Velocity (v_c) in the vena contracta is defined in terms of a theoretical (frictionless flow) velocity (v_{th}) and the velocity coefficient (C_v). The area (A_c) of vena contracta is defined in term of the opening area (A) and the contraction coefficient (C_c). The velocity in the vena contracta is constant therefore the flow rate (q_c) in vena contracta is:

$$q_c = A_c v_c = C_c C_v A v_{th}$$

The theoretical velocity (v_{th}) is mainly due to pressure difference (ΔP) and the product of C_c and C_v is called the discharge coefficient (C_D).

$$v_{th} = \sqrt{\frac{2\Delta P}{\rho}} \quad \text{and} \quad C_D = C_c C_v$$

The C_c and C_v are mostly discussed in literature but in practice, especially with operable windows, they are extremely difficult to estimate. Therefore, the discharge coefficient is usually used to define the flow. From above mentioned correlations, the airflow through an opening is defined as:

$$Q \equiv C_D A \sqrt{\frac{2\Delta P}{\rho}} \tag{1}$$

This is generally referred as orifice flow plate equation. A colossal literature is available for estimation of C_D. However, a constant values of C_D is predominantly used in practice. These constant values are derived from the data used to estimate the flowrate in pipes [1].

Bot et. al. performed [6] a full scale measurements of flowrate through one side mounted casement windows (façade window). The author defines the resistance coefficient/friction factor in terms of aspect ratio of the window and opening angle. *The resistance coefficient (ζ) is a coefficient that defines the pressure drop due to friction in the opening and flow. Theoretically, it is equal to:*

$$\Delta P_{fr} = \frac{1}{2}\zeta\rho v_c^2$$ [3].

The authors use the cross sectional opening area, and from the results of their research it can be concluded that the overall discharge coefficient of the top hinged window increases with the increase in opening angle.

P. Heiselberg et. al. [1] uses the minimum opening area to estimate the discharge coefficient of a façade window with movable flap. Experimental results showed that the discharge coefficient is not constant for different flap opening angles. The authors conclude that the value of discharge coefficient is approaching to 1 with the decrease in flap opening angle (corresponding to minimum opening area). Only for large opening angles the value of 0.6 can be used as a discharge coefficient of a window with movable flap.

Andersen [5, 7] discussed theoretically, friction and contraction coefficients of openings with movable flaps. The author use artificial as well as pure resistance coefficients along with artificial and real opening angle to calculate the contraction coefficient. The author concludes that (for centrally hinged flap) the contraction coefficient decreases with increase in the flap opening angle. Furthermore, the resistance coefficient (both artificial and theoretical) increases with the increase in flap opening angle. The real opening angle is dependent on aspect ratio of the window. Therefore, the contraction coefficient, and consequently the discharge coefficient, is also dependent on the aspect ratio of the window. For sharp edged openings, the discharge coefficient is about 0.61 [5, 7].

Hult et. al. [2] determines the discharge coefficient of the façade window using CFD. The authors conclude that the discharge coefficient of a façade window is reliant upon aspect ratio and window opening angle. However, for larger opening angle it approaches to the commonly used value of 0.6. According to their research, the CFD results suggest that the actual flow through a top-pivoted window may be as much as twice the flow predicted by EnergyPlus software.

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[3] *The definition is from theoretical books and from (Andersen 1996)*

Methods

CFD techniques was used to test the dependence of discharge coefficient (C_D) on the flap opening angle(α). The CFD domain was defined in such a way that on right side of the domain the outflow through the window could be examined, and the inflow through the window could be examined from the left side of the domain. For reducing the processing time, only half part of the window was examined by using symmetric boundary condition. Height and width of the domain was selected in such a way that the size of the domain had no influence on the local velocities and the pressure distribution around the window. The model room was defined as shown in Figure 2. InVent was the opening in the model room with the window and OutVent was the opening without any window. Both InVent and OutVent were on the roof with slope/pitch of 45°. The window and flap geometry were kept simple because the details of window (minor bends on flap) has insubstantial effect on overall discharge coefficient [8].

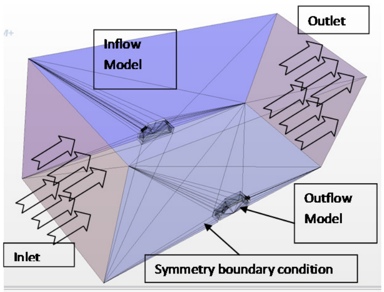

Figure 1: CFD Domain

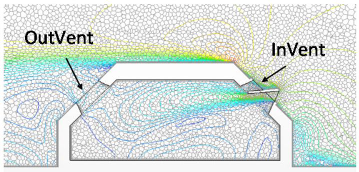

Figure 2: Model room with InVent and OutVent

The polyhedral meshing scheme was used with 8 prism layers mesher at the boundaries. The base size was 1 m. The prism layers were 8% of the base cell size. The surface growth rate was 1.3. Allowable skewness for cells was 85°. Several parts of the domain had customised surface mesh size to ensure proper mesh quality. The Inlet was the velocity-inlet, the Outlet was the pressure-outlet. The domain top, left and the right boundaries were symmetric boundaries and all other boundaries were walls with no slip conditions.

Physics:

A body interacts with the surrounding fluid through pressure and shear stress, and the resultant force in the direction of stream is the drag force. The drag coefficient D_f is used to define the drag force when the detailed information about pressure and shear stress is not known i.e. it is a ratio between the drag force and the wind pressure force [4]. The minimum number of cells was selected in such a way that by increase in the number of cells, there is no effect on the D_f of the model room. This means that the further decrease in cell sizes had no effect on the local velocity and the pressure distribution around the building.

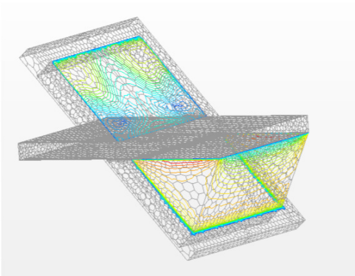

Figure 3: Outside pressure (average)

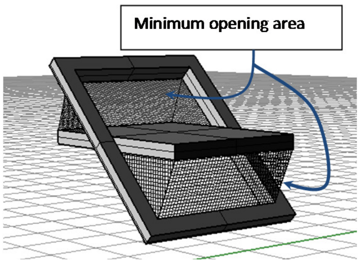

Figure 4: Centre-pivot roof window and minimum opening areas

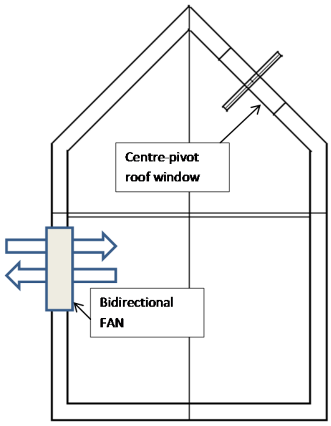

Figure 5: Schematic of on-site measurements

The "Two layer realizable k-ε turbulent" model was used to compute the airflow and its physical behaviour [3]. The working fluid was incompressible ideal gas. The flow was 3D steady state. Flow and energy were both modelled using the segregated approach. The second-order upwind discretisation scheme was used for both flow and energy. Under relaxation factors for velocity, pressure and energy were 0.7, 0.3 and 0.9 respectively. Inlet condition (Inlet -Figure 1) for the turbulent kinetic energy ($k_o = 1.5(T_iU_o)^2$) and the dissipation rate ($\varepsilon_o = k_o 1.5/l_o$) were according to Nielsen [9] recommendations. Where, T_i is the turbulence intensity and was taken as 4% with inlet temperature of 293K. U_o is inlet velocity in {m/s}. l_o{m} is length scale and was taken as one-tenth of the height of the inlet. The inlet velocity was selected in such a way that for each simulation the airflow through the window was fully developed turbulent flow. The C_D value for fully developed turbulent flow does not vary with Reynolds number [3]. Equation 1 was used to find out the C_D for the window (Figure 4).

The flowrate at InVent (Figure 2) was estimated by the flowrate at the OutVent (Figure 2Figure). The flow Q{m^3/s} at OutVent was calculated by integration of velocity over the area of OutVent i.e. the face area of the cells at the interface (OutVent and external region) times the perpendicular component of the velocity i.e.

$$Q \equiv \sum_{i=1}^{n} A_i v_i = \frac{1}{\rho} \sum m_i \qquad (2)$$

Where, A_i\{m^2\} is the face area of cell at the interface and v_i\{m/s\} is the perpendicular (to A_i) component of the velocity. The airflow rate can also be calculated by the mass flow rate divided by the density as shown in Equation 2. The density ρ is constant i.e. 1.2 kg/m^3, m_i is mass flow rate in kg/s.

Pressure difference:

To measure the pressure difference (ΔP) across the InVent one probe was measuring the pressure inside the room (in the centre of the room). The outside pressure was an area weighted average of outside pressure at the InVent opening as shown in Figure 3.

Opening area:

The C_D was also dependent on the opening area. Therefore it was evaluated for two different opening areas. One was the minimum opening area (A_{min}). The minimum opening area is shown in Figure 5. The sum of two minimum opening areas is the total minimum opening area (Figure 5). For flap opening angles of 50^o and greater, the minimum opening area is the sum of two face cross sections of the window.

Another way to define the opening area is the gross face area (A_{face}) of the opening i.e. 1.14 x 1.4.

On-site measurements:

On-site measurements were performed in the Energy Flex House (EFH) of the Technological Institute of Denmark (Copenhagen). The house was equipped with the VELUX centre-pivot roof window. Blower door test, for infiltration/exfiltration, were perform before the measurements. Only outflow measurements were compared in this study, because this study is mainly focused on CFD, and measurements were performed only for validation purpose. The flow through the window was the sum of fan flow and infiltration/exfiltration. Figure 5 shows the schematic diagram of the measurements setup. The discharge coefficient was calculated by using equation 1 and minimum opening area. The outside pressure was the area weighted average pressure around the window. Inside pressure was the average inside pressure of the house. The centre-pivot roof window in the EFH was fully automatic and it was not possible to open the window more than 17^o. Therefore, discharge coefficient only for very small (α<17^o) opening angles was measured.

Results

Figure 7 illustrates the discharge coefficients ($C_{D,min}$) of a centre-pivot roof window when minimum opening area is used in equation 1. The CFD results for $C_{D,min}$ of inflow through the window (flow from outside to inside) is represented by the blue line. The CFD results for $C_{D,min}$ of outflow (flow from inside to the outside) is represented by the red line. The green line is $C_{D,min}$ that is obtained by the on-site measurements. As mentioned earlier, it was not possible to measure (onsite) the discharge coefficient for angles greater than 17°. Therefore, measurement data is available only for 17°, 14°, 9.3° and 4.6°. It should be noted that the minimum opening area is taken from the manufacturer catalogue. On the contrary, with available computing power for CFD calculations, it was very time consuming to go below 15° of opening angle. That's why the CFD simulation was performed only for 15°, 25°, 35°, 45°, 50° and 90°. It should be noted that the minimum opening areas were calculated through the CAD drawings.

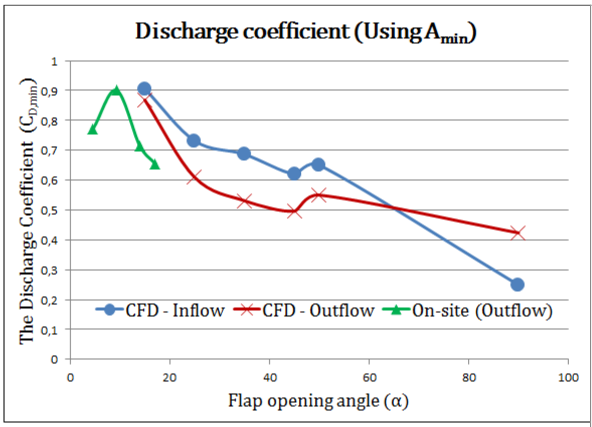

Figure 6: Discharge coefficient of the centre-pivot roof window using minimum opening area

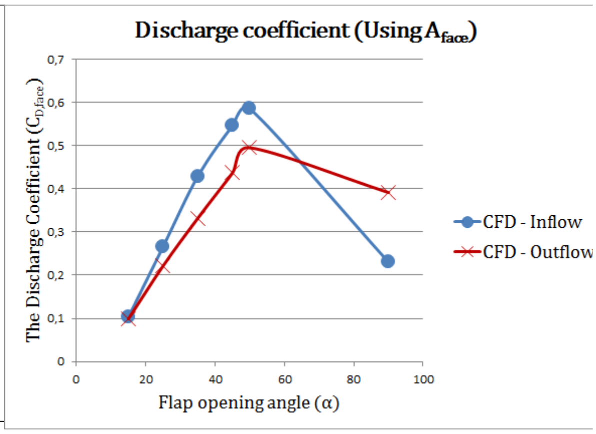

Figure 7: Discharge coefficient of the window using face area

Figure 6 illustrates the discharge coefficients ($C_{D,face}$) when the gross face opening area is used in equation 1. The blue line shows the $C_{D,face}$, obtained from CFD calculations, when flow is directed inward. Whereas, the redline is the $C_{D,face}$ (CFD result) when flow is directed outward.

Discussion

From Figure 7, apparently there is a difference between the $C_{D,min}$ values evaluated by the measurements and by the CFD. One of the reasons is ideal versus real condition. However, the trend of decrease in $C_{D,min}$ is almost the same. Ergo, the problem might also be in calculation of the minimum opening area. Therefore, another quantity, C_D x A (so called effective area) of both, measurements and the CFD are compared. This comparison is shown in Figure 8. The difference in measurements and the CFD results are now minimum. The CFD results are in sound accordance with the measurements at the angle of 15° and around. The concurrence, of CFD and the measurements, at low opening angles are taken as benchmark for higher opening angles. Hence, the realizable k-ε turbulent model predicts the flow (across the window) in a good agreement with the measurements.

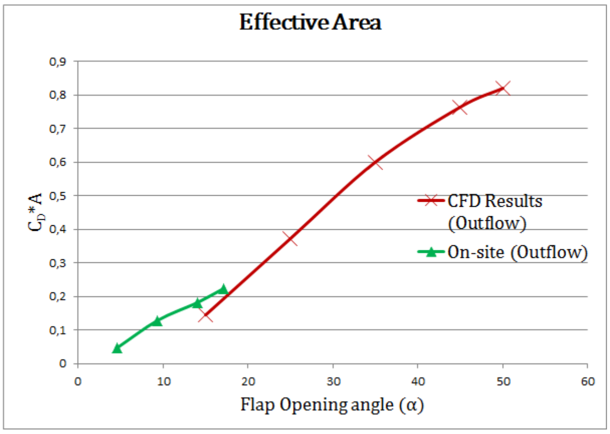

Figure 8: Comparison of CFD results with On-site experimental results

The $C_{D,min}$ curve represents the change in the discharge coefficient due to (mainly) flow effect. Whereas, the $C_{D,face}$ curve represents the change in the discharge coefficient due to both flow and area effects.

The $C_{D,min}$ (as shown in Figure 7) decreases with the certain pattern until it reaches to 50° of the flap opening. At this point the A_{min} shifted from the position shown in Figure 4 to the face cross sectional area. Then the pattern of decrease in $C_{D,min}$ also changes. This change in $C_{D,min}$ is due to the change in A_{min} and due to the fact that pressure field on the roof surface (closed to the window) is disturbed.

In the case of inflow through the window, the flow in each section of the window is not evenly distributed, except for very small opening angles. Flow in the lower section is much higher then in the upper section. At the angle of 90° there is no flow in the upper section of the window. In the case of outflow through the window, the flow in each section of the window is somewhat evenly distributed. Therefore, the $C_{D,min}$ inflow is higher than the $C_{D,min}$ values of outflow. This phenomenon negates the assumption of stagination condition at the inlet in determination of C_D. However, the slope/trend of the outflow curve is same as of inflow curve. After 50° of flap opening, the slope of outflow become lower than the inflow. Therefore, the $C_{D,min}$ at 90° of flap opening of outflow direction was higher value than that of inflow direction. The reason for difference in $C_{D,face}$ curves (for inflow and outflow direction) is also the same.

For more concrete conclusion, the results have to be evaluated for several openings with different aspect ratios. However, these results are only subjected to the particular window type and for natural ventilation caused by wind effect.

Conclusion

It is concluded that the k-ε turbulent model predicts the flow in sound agreement with the measurements. It is also concluded that in the orifice flow plate equation, the discharge coefficient is not a constant quantity. The $C_{D,min}$ decreases with flap opening angles. The trend of decrease in $C_{D,min}$ changes after 50^o of flap opening angle. Likewise, the discharge coefficient also depends upon the flow direction. The $C_{D,min}$ is higher for inflow and the $C_{D,min}$ is lower for outflow. For higher opening angles (e.g. higher than 65^o) the criteria is swapped.

Acknowledgements

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Improvement of summer comfort by passive cooling with increased ventilation and night cooling

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Abstract

The present study describes the potential improvement of summer comfort and reduction of energy consumption that can be achieved by adopting passive cooling solutions, such as daytime comfort ventilation with increased air velocities and night cooling, in domestic buildings. By means of the IDA ICE based software EIC Visualizer, the performances of ten ventilation and cooling strategies have been tested in four different climatic zones across Europe (Athens, Rome, Berlin and Copenhagen). Thermal comfort and indoor air quality (IAQ) have been evaluated according to the standard EN15251 for the summer period of the year only. The study revealed that thermal comfort can be achieved by passive means in all four locations. It was also found that, with the exception of Athens, the initially investigated combination of ventilative and night cooling is too aggressive, causing overcooling and increasing the energy consumption. A moderate strategy performed well without overheating and overcooling in Rome, Berlin and Copenhagen. In general the natural ventilation turned out to be capable to achieve a very good IAQ and a reduction in energy consumption in all locations, when compared with mechanical ventilation or mechanical cooling.

Keywords

Natural ventilation, ventilative cooling, night cooling, increased indoor air velocity, residential buildings.

Introduction

The 2007 report of the Intergovernmental Panel on Climate Change (IPCC) [1] stated that the warming of the climate global system is an ascertained problem. For this reason in 2008 the EU set the target of reducing by 20% the total energy consumption within 2020 [2]. According to the Promotion of the European Passive House (PEP)[3], buildings account for 40% of this total energy consumption, and through the application of the Passive House concept, a relevant reduction of the energy consumption, quantifiable in a CO_2 emission reduction between 50% and 65%, can be obtained. The aim is to lower the buildings' energy demand without affecting the thermal comfort or the IAQ. The comfort condition is function of different parameters and thermal comfort can be provided within a range of air temperatures. When the air temperature increases, the warm thermal sensation can be restored from warm to neutral by decreasing the mean radiant temperature or by increasing the convective heat exchange between the body and the surrounding ambient [4]. The reduction of the radiant temperature is achieved by mean of the *night ventilation*: cold air is circulated through the building during night, the building structure is then cooled, providing a thermal sink and a lower radiant temperature during the next day. The increase in the convective heat exchange is the basic idea of the *ventilative cooling*: thermal comfort is obtained by increasing the air velocity in the room through natural or mechanical ventilation.

The effectiveness of increased air velocity and night cooling in reducing the energy consumption has been proven by means of both field surveys and dynamic simulations. In particular E. Gratia et al. [5] showed it is possible to reduce the cooling needs by about 30% using the ventilative cooling strategy. S. Schiavon and A. K. Melikov [6] demonstrated that increased air velocities can improve the comfort and allows a cooling energy saving between 17% and 48% and a reduction of the maximum cooling power between 10% and 28%. Furthermore Yun et al. [7] stated that opening the windows at the ambient temperature higher than the indoor temperature does not help to cool down the office, but can still improve the thermal comfort providing direct cooling over the occupants.

According to many authors, night ventilation appears to be one of the most promising passive cooling techniques. The work of Böllinger and Roth [8] revealed that in Frankfurt a nighttime air flow rate of 3 ach can compensate for a specific load of about $35W/m^2$, while for a 6 ach the compensated specific load rises up to $41W/m^2$. Similarly Santamouris et al. [9] showed that night ventilation applied to residential buildings may decrease the cooling load up to 40 kWh/m^2. Santamouris [10] also found that, under free floating condition, the night ventilation decreases the next day peak indoor temperature up to 2.5°C and reduces the expected number of overheating hours between 64% and 84%. According to Shaviv et al. [11] depending on thermal mass, air flow rate and temperature swing, the night cooling can achieve a 3 – 6°C temperature reduction in the hot and humid climate of Israel.

The purpose of this project is to determine, by means of dynamic simulations with the EIC Visualizer software, under which climatic condition the passive cooling techniques are capable to reduce the energy need without compromising the occupants' thermal comfort and the IAQ, during summer. The tested scenarios include, beside a fully mechanical system regarded as a reference case, both natural and hybrid ventilation and cooling solutions.

The methodology

The building

A 1½-storey, single-family house with a 8x12 m footprint, corresponding to a 175 m^2 floor area, has been selected for the investigation (Figure 1). The dwelling has no internal partitions and has then been studied as a single-zone building. The building tightness allows an infiltration rate of 0.15 ach for an outdoor-indoor pressure difference of 50 Pa. The internal surface of roof, walls and floor is an exposed concrete layer whose thickness has been obtained by empirically optimizing the thermal mass.

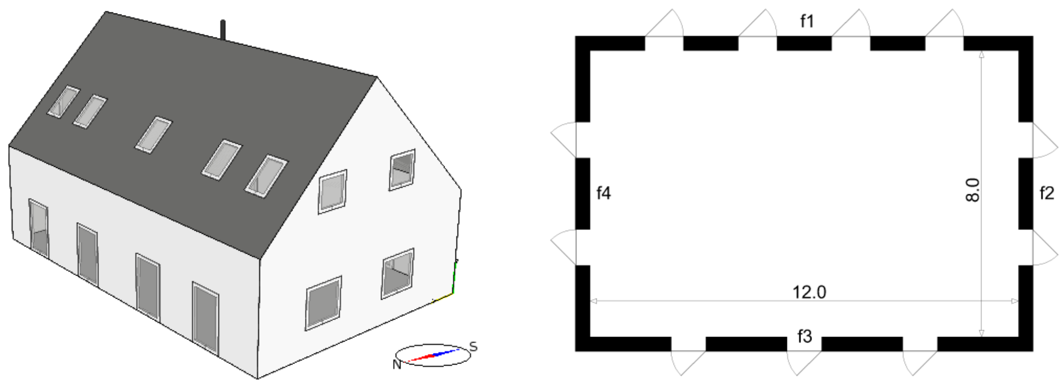

Figure 1: Visual representation (left) and footprint (right) of the building selected for the investigation.

To achieve a potential for sufficient natural ventilation and good daylight conditions requires a large windows area; the building has then a 30.4 m^2 glazed surface (corresponding to 17% of the internal floor area): 23.1 m^2 are façade windows and 7.3 m^2 are roof windows. All windows are operable and equipped with an external sunscreen.

The thermal properties of opaque and glazed surfaces are summarized in Table 1.

	U-value [W/(m^{2}K)]	Solar heat gain coefficient (g)	Solar transmittance (T)	Solar transmittance for the visible spectrum (T_{vis})	Internal/external emissivity
Roof	0.23	-	-	-	-
Wall	0.34	-	-	-	-
Floor	0.32	-	-	-	-
Glass	1.47 (0.90)	0.6 (0.1)	0.54 (0.05)	0.77	0.84

Table 1: Thermal properties of opaque and glazed surface (in brackets the multiplier due to the sunscreen).

In the AHU a supply fan introduce fresh air from outside the building through a grid at the ground level and an extraction fan extracts the exhausted air through a grid located 2.5 m from the ground. Both the supply and the extraction fans give a pressure rise of 200 Pa with a 0.8 electricity-to-air efficiency and a 0.25 kW/(m^3/s) specific power. A 5 Pa pressure loss has been assumed for the grid. No heat recovery has been used: the air is then supplied at the outdoor thermodynamic condition.

The heating system is an ideal heater with a 17500 W maximum power and a 0.9 generation efficiency, the cooling system is an ideal cooler with a 35000 W maximum power and a COP of 2.4. The distribution losses have been assumed to be equal to 1% of the heat delivered by the plant for the heating system and to 0.10 W /m^2 of the internal floor area for the cooling system.

Standard EN15251

Thermal comfort, IAQ and energy consumption are the three parameters used to quantify the performance of the tested ventilation and cooling strategies. The thermal comfort and the IAQ (the CO_2 level has been chosen as indicator of the IAQ) have been evaluated according to the standard EN15251 [12]. For the thermal comfort the standard prescribes two models. In buildings equipped with a mechanical cooling system the comfort categories are defined according to the non-adaptive model of the ISO7730. In naturally ventilated buildings the standard prescribes an adaptive model comparable to the one developed by de Dear and Brager [13]. In addition the EN15251 adopts the temperature offset that can be obtained by means of increased air velocities under summer comfort condition proposed in the ISO7730. Since only a graphical representation of the temperature offset is reported in the standard, an approximation had to be made. Four easy-to-identify points ((0.2;0), (0.3;1), (0.9;2.75) and (1.2;3.3)) have been isolated and interpolated with a logarithmic trend line. The equation of the trend line (1) has then been used to calculate the temperature offset.

$$\Delta T = 1.777 \cdot \ln(v) + 2.9782 \quad (1)$$

Where ΔT is the temperature offset and v is the indoor air velocity.

Thermal comfort and IAQ have been evaluated only during the *natural ventilation period*, which is defined as the period of the year that starts the day during which natural ventilation is used for the first time (i.e. the conditions for the window opening are met for the first time since the beginning of the year), and ends the last day of application of natural ventilation (i.e. the conditions for the window opening will never be met again for the rest of the year).

Assumptions and system setup

The simulations have been run with the software Energy and Indoor Climate Visualizer (EIC Visualizer), which is based on the commercial software IDA Indoor Climate and Energy 4 (IDA ICE).

The dwelling is occupied by four people who leave the house at 8:00 in the morning and return at 17:00 in the afternoon every weekday. During the weekend they spent 24 h/day inside the building. The occupants' clothing and activity levels have been set equal to 0.5 ± 0.2 clo (the clothing level is automatically adjusted between limits to obtain the best comfort) and 1.2 met respectively. Other contributions to the internal load are the constant 4 W/m^2 due to the equipment and the load from electrical lighting with an installed capacity of 4 W/m^2. The lights are switched on only when the average daylight level is below 50 lux and with a percentage of lighting on simultaneously equal to 75% (2.4 W/m^2). The sunscreens used to reduce the solar gain are operated by a PI controller and are automatically activated when the indoor air temperature rises above 23°C.

The windows controller is intended to simulate human behaviour [14]. During daytime (from 7:00 to 22:00) the window opening is modulated to maintain an air temperature set point by cooling when the outdoor temperature is lower than the indoor by 2°C. During night the windows are opened between 22:00 and 7:00, if at 22:00 the indoor air temperature is higher than the outdoor and it is above the given threshold.

When window opening is not required with regards to thermal comfort, the mechanical system supplies a constant air flow rate of 0.29 l/s/m^2 calculated, according to EN15251, as sum of the amount of fresh air needed to compensate for the pollution from the occupants (4 l/s/person for category III) and the amount of fresh air necessary to remove the building emission of pollutants (0.2 $l/s/m^2$ for a very low polluting building).

The set points for the heating and for the cooling systems are 21.0°C and 24.6°C respectively.

An approximated method has then been used to calculate the mean indoor air velocity. The air flow rate through the windows has been divided into two contributions according to the direction, axial or transversal, with respect to the building footprint. For each direction two values of air velocity have been calculated: one on the windows threshold and one on the building cross section. Averaging the threshold air velocity and the cross-section air velocity the two components, namely axial indoor air velocity and transversal indoor air velocity, of the indoor air velocity have been obtained. Finally, averaging[4] those two components, an approximation of the indoor air velocity value has been obtained. The procedure just described has been adopted to determine the air velocity, and from it the temperature offset, hour by hour.

The temperature offset has then been subtracted from the operative temperature to calculate the *perceived operative temperature* (2), a parameter whose purpose is to express the temperature actually experienced by the body.

$$T_{op,\,perceived} = T_{op} - \Delta T \qquad (2)$$

The climatic zones

The analyses have been performed by testing the ventilation and cooling strategies in four different climatic conditions. The selected cities are Athens, Rome, Berlin and Copenhagen. According to the Köppen-Geiger climate classification system [15], both Athens and Rome have a Mediterranean climate, Berlin has a humid-continental climate and Copenhagen is in the oceanic climatic zone.

In Athens the climate is hot and dry, with a maximum monthly average high temperature of 33.1°C in July. Rome is hot and humid; the highest monthly average high temperature (30.6°C) is reached in August. Berlin has a temperate climate; the monthly average high temperature rises up to 24°C in July. Lastly Copenhagen has a

[4] The temperature offset has been calculated referring to the velocity-offset curve valid when the mean air temperature is equal to the mean radiant temperature. In our case the mean radiant temperature is lower than the mean air temperature for most of the time, then, for the same air velocity, the offset prescribed by the standard is lower. To compensate for it, the two contribution have been averaged and not summed as vectors.

cold climate and the monthly average high temperature presents a maximum of 20.4°C in July.

Rome presents very good night cooling potentials because of a 12°C day-to-night temperature swing. Also in Athens the night cooling is expected to perform well. Berlin and especially Copenhagen present a lower day-to-night temperature difference. All the locations are windy enough to provide an adequate air flow rate. In particular Athens and Copenhagen preset a prevailing wind blowing from North and from West respectively.

The climatic data used in the simulations are obtained from the ASHRAE's International Weather for Energy Calculations (IWEC) database.

Case studies

The analysis can be divided in two steps. First a set of preliminary simulation has been run to empirically optimize the thermal threshold for the night ventilation, the orientation, defined referring to façade 3 (f3 in Figure 1) and the building thermal mass, with respect to the climatic condition of each location. After, ten different ventilation strategies have been tested. The examined cases are described in Table 2.

Case studies	Ventilative cooling	Night cooling	Mechanical ventilation	Mechanical cooling
N_02_H	Non-increased air velocities	Open all night from 22:00 to 7:00	When windows are closed	Not equipped with a mechanical cooling system
N_02_H_A	Non-increased air velocities	Modulated according to comfort requirements	When windows are closed	Not equipped with a mechanical cooling system
N_02_HC	Non-increased air velocities	Open all night from 22:00 to 7:00	When windows are closed	Daytime: when the temperature is above the set point. Nighttime: when the temperature is above the set point and the windows are closed
N_I_H	Increased air velocities	Open all night from 22:00 to 7:00	When windows are closed	Not equipped with a mechanical cooling system
N_I_H_A	Increased air velocities	Modulated according to comfort	When windows are	Not equipped with a mechanical cooling system

		requirements	closed	
N_I_HC	Increased air velocities	Open all night from 22:00 to 7:00	When windows are closed	Daytime: when the temperature is above the set point. Nighttime: when the temperature is above the set point and the windows are closed
M_HC	Never used	Never used	During the entire year	When the temperature is above the set point
M_HC_N	Never used	Open all night from 22:00 to 7:00	When windows are closed	Daytime: when the temperature is above the st point. Nighttime: when the temperature is above the set point and the windows are closed
M_HC_N_A	Never used	Modulated according to comfort requirements	When windows are closed	Daytime: when the temperature is above the set point. Nighttime: when the temperature is above the set point and the windows are closed

Table 2: Case studies

Results

For every location the performances of the analyzed cases are graphically represented by mean of the *individual signature*. In a 3D graph the thermal comfort, the IAQ and the energy consumption have been correlated. The data used to plot the individual signatures and to compare the performances of the different strategies are: for the thermal comfort the percentage of hours in category II (the static or adaptive model has been used depending on whether the mechanical cooling system had been used or not), for the IAQ the percentage of hours in category I and for the energy consumption the energy used on a year long period expressed as a percentage of the energy consumption of the M_HC scenario.

Athens

The tested night thresholds for Athens range from 23.0°C to 25.5°C with a 0.5°C increase step. The sensitivity analysis proved the 25.0°C threshold to be the best performing: when compared with the 23.0°C one, the number of hours of comfort increases by 16.7% and the energy consumption is reduced from 43.7 kWh/m^2 to 1.2

kWh/m^2 because of the overcooling prevention. Also it shows a 4.0% increase in the thermal comfort and the same energy consumption if compared with a scenario where the night ventilation is not used (the overheating is avoided). For the strategies which combine mechanical cooling and night ventilation the threshold has been lowered to 24.5°C.

Eight orientations have been tested (N, NE, E, SE, S, SW, W, NW). The SW orientation, exposing to NE the façade with the largest glazed surface, allows to achieve a very good thermal comfort (for 98.5% of the time the building is in category II) by reducing the solar gain. The price to pay is that, with its 2.2 kWh/m^2, the SW presents one of the highest energy consumption among the tested orientations (the solar gain is reduced during winter as well, the heating system must then supply more heat to the dwelling). Also, with a 0.25 m/s average indoor air velocity, the SW orientation has the largest potential for ventilative cooling.

The thermal mass optimization evaluated the building performances for concrete layer thicknesses ranging from 0.08 m (415 kg/m^2 of floor area) to 0.24 m (1183 kg/m^2) with a 0.02 m increase step. Comparing the 0.24 m with the 0.08 m thickness, the thermal comfort increases by 1.5% (both overheating and overcooling are reduced) and the energy demand is decreased by 37%. Increasing the building mass is beneficial up to a 0.20 m concrete layer thickness (991 kg/m^2), a further increases gives only a negligible performance improvement.

For the daytime ventilation a 24°C threshold has been chosen.

With the selected parameters the mean air velocity for the non-increased air velocity cases is 0.12 m/s, while for the increased air velocity ones is 0.25 m/s. The natural ventilation period goes from April 20th to October 30th, that is 194 days of natural ventilation (53% of the year).

The individual signatures (Figure 2) show that the N_02_H_A scenario is the best performing.

When compared with the fully mechanical system N_02_H_A grants only a slight improvement in the thermal comfort (1.2%), but an 83% decrease of the energy consumption. The IAQ is much higher as well: the mechanical system supplies 0.5 ach only, while the natural ventilation strategy provides 4.8 ach during night time and 1.7 ach during daytime, which results in a 26% increase of time in category I. It is also true that the N_02_H_A requires an automatic controller for the windows opening and such systems are not very common in domestic buildings. If we limit the choice to the manually operated systems the best performing is the N_I_H. In fact during the transition seasons the increased velocities of the N_I_H scenario are capable to maintain the indoor temperature within an acceptable range, thus limiting the use of night ventilation and, with it, the overcooling (task that in the non-increased air velocities scenario required the installation of the automatic controller). The N_I_H, when compared with the M_HC case, presents a slight decrease in the thermal comfort (0.6%), an improvement in the IAQ (28%) thanks to the increased air flow rates (4.1 ach during night and 5.6 ach during day), and a reduction in the energy consumption (83%).

Among the mechanically cooled building the N_I_HC ensures the best comfort conditions: the thermal comfort is increased by 0.6%, the IAQ by 5.9% (the hybrid ventilation system supplies 2.6 ach during night and 1.5 ach during day) and the energy demand is reduced by 20% (the consumption is lowered by 8.3 kWh/m^2, of which 8.0 kWh/m^2 for cooling needs).

This proves that if the mechanical system is assisted by passive ventilation and cooling strategies, even based on a very simple logic, the result is a relevant reduction in the energy consumption and a potential increase in the quality of the indoor environment.

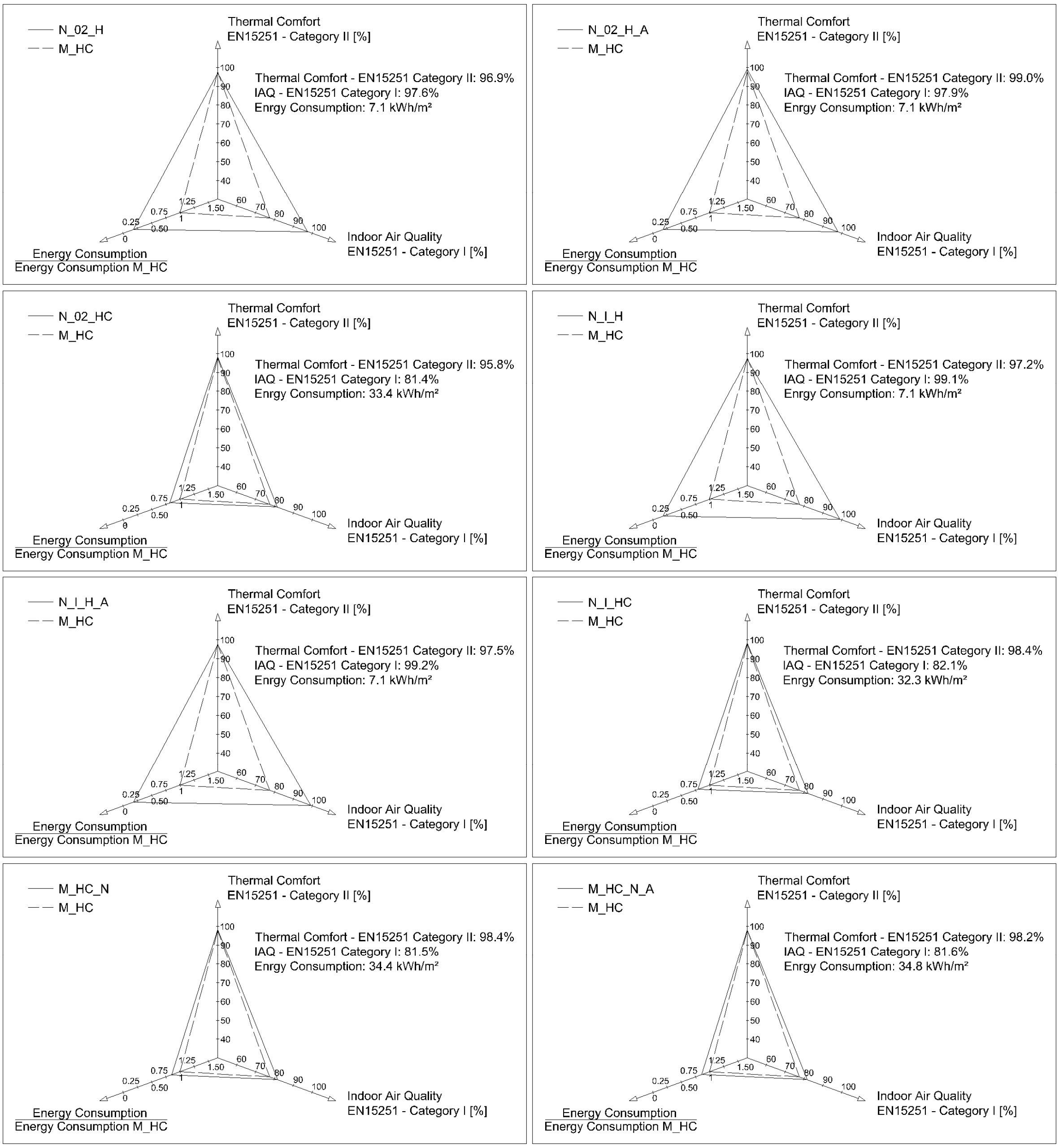

Figure 2: Individual signatures for the case studies in Athens, reference case: M_HC (Thermal Comfort – EN15251 Category II: 97.8%, IAQ – EN15251 Category I: 77.5%, Energy consumption: 40.6 kWh/m2).

Rome

The same night cooling thresholds used in Athens have been tested in Rome (i.e. 23.0°C to 25.5°C with a 0.5°C increase step), where the best thermal comfort has been obtained when the night cooling strategy is not applied. In all cases the discomfort is caused by the overcooling of the building, due to the large day-to-night temperature swing mentioned before. Nonetheless a 24.0°C night ventilation threshold has been chosen since the prevention of the summer overcooling is a priority.

The E orientation has been considered the best performing.

For the thermal mass analysis the considerations made for Athens are valid for Rome as well, i.e. increasing the thermal mass is beneficial up to a 0.20 m concrete layer thickness.

For the daytime ventilation a 24°C threshold has been chosen.

With the selected parameters the mean air velocity for the non-increased air velocities cases is 0.16 m/s while the mean air velocity for the increased air velocities ones is 0.28 m/s. The natural ventilation period starts on April 7th and ends on October 25th, the natural ventilation strategies are then applied for 202 days over 365 (55% of the year).

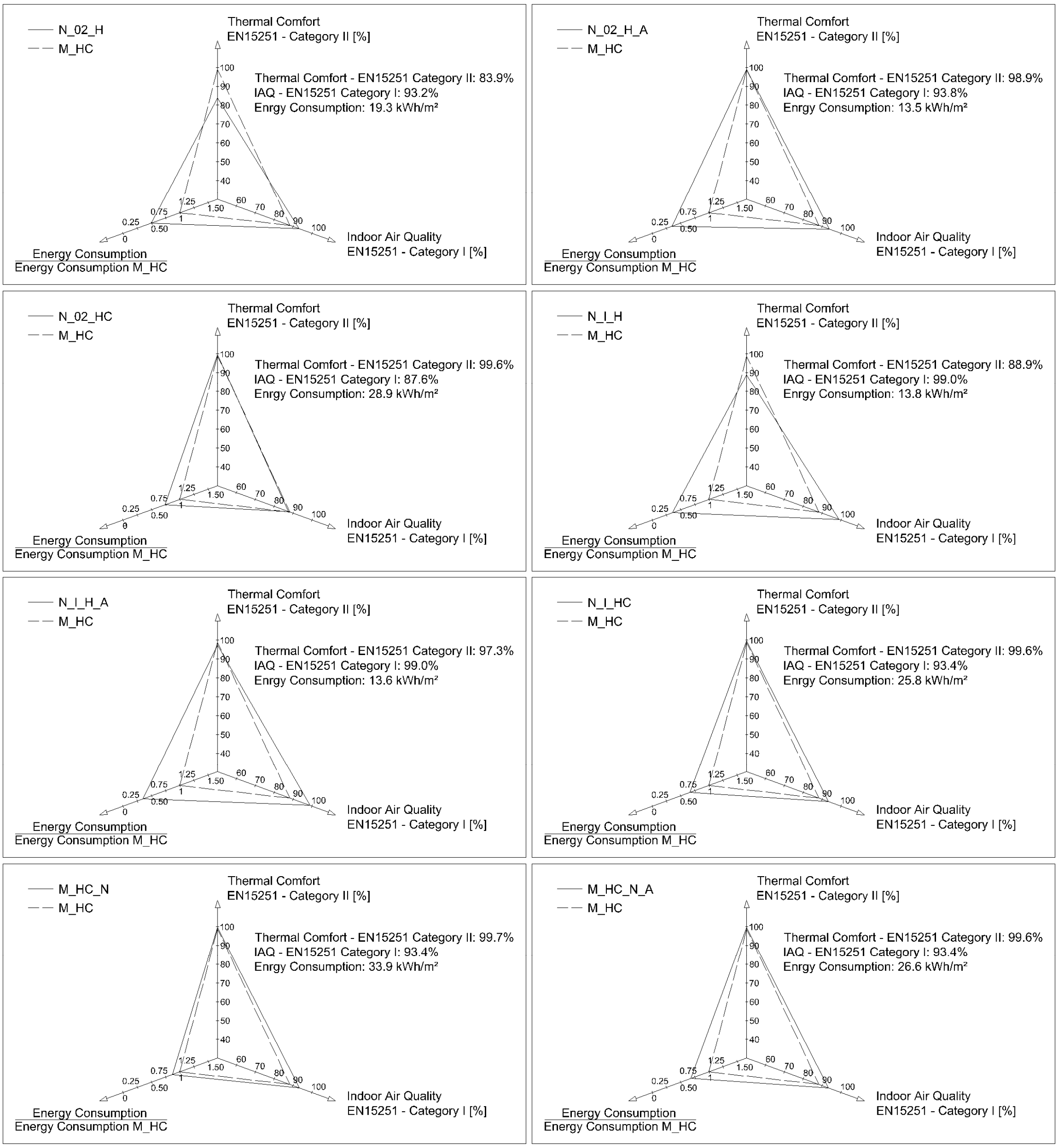

Figure 3: Individual signatures for the case studies in Rome, reference case: M_HC (Thermal Comfort – EN15251 Category II: 99.0%, IAQ – EN15251 Category I: 88.2%, Energy consumption: 39.0 kWh/m²).

For Rome in the mechanically cooled building the thermal comfort is generally higher than in the naturally ventilated and cooled ones. The cause is the overcooling of the building during night time and indeed the introduction of the automatic controller, creating a constrain on the night air flow rate, improves the thermal comfort in both the N_02_H (17.9%) and the N_I_H (9.4%) solutions. Among the passive cooling systems the one that give the best results is the N_02_H_A. From a comfort point of view the solution performs as good as the M_HC system (there is a negligible 0.1% decrease), the IAQ is increased by 6.0% because, again, the natural ventilation provide much higher air flow rates (4.2 ach during night and 1.5 ach during day), and the energy consumption is reduced by 65%. In Rome when the mechanical cooling system is assisted by an automatically controlled night ventilation strategy, the improvements are relevant: the consumption is reduced by 31.8% and the indoor environment is more comfortable (+0.6% the thermal comfort and +5.9% the IAQ). The energy demand can be further decreased if the mechanical system is assisted by ventilative and night cooling (-33.8%), in which case the automatic control becomes unnecessary.

Berlin

In Berlin the upper limits for the comfort categories defined according to the adaptive model are quite lower than in Athens and Rome, then the tested night ventilation thresholds have been proportionally decreased. For the sensitivity analysis the potential thresholds go from 22.0°C to 24.5°C with a 0.5°C increase step. As in Rome, the solution that generates the best thermal comfort is the one without night cooling and, as in Rome, the overcooling prevention has been considered a priority. Then a 23.5°C threshold has been adopted.

In Berlin the influence of the orientation on the building performance is negligible, then the N orientation, being capable to provide a 0.26 m/s average indoor air velocity, has been chosen.

A 0.20 m concrete layer has been considered sufficient (2.3% increase in the thermal comfort and 4.3% reduction of the energy consumption when compared with the 0.08 m thickness).

For the daytime ventilation a 23°C threshold has been chosen.

With the selected parameters the mean air velocity for the non-increased air velocities cases is 0.12 m/s while the mean air velocity for the increased air velocities ones is 0.26 m/s. The natural ventilation period starts on May 7th and ends on October 5th, the natural ventilation strategies are then applied for 152 days over 365 (42% of the year).

In Berlin the energy demand for cooling accounts for only 5.4% of the total consumption, then the passive cooling strategies are not particularly beneficial from an energetic point of view: the highest reduction in the energy consumption is indeed equal to 5.6% (N_02_H_A). If we limit the choice to the solutions without mechanical cooling, N_02_H_A is the scenario with the highest performances: the 0.2% reduction in the thermal comfort is compensated by the improved IAQ (1.7%) and by the decreased energy consumption. Among the mechanically cooled buildings the optimum is reached by the N_I_HC system, which increases the IAQ by 3.6% and reduces the consumption by 2.6%, maintaining the building in category II for 100% of the occupancy time.

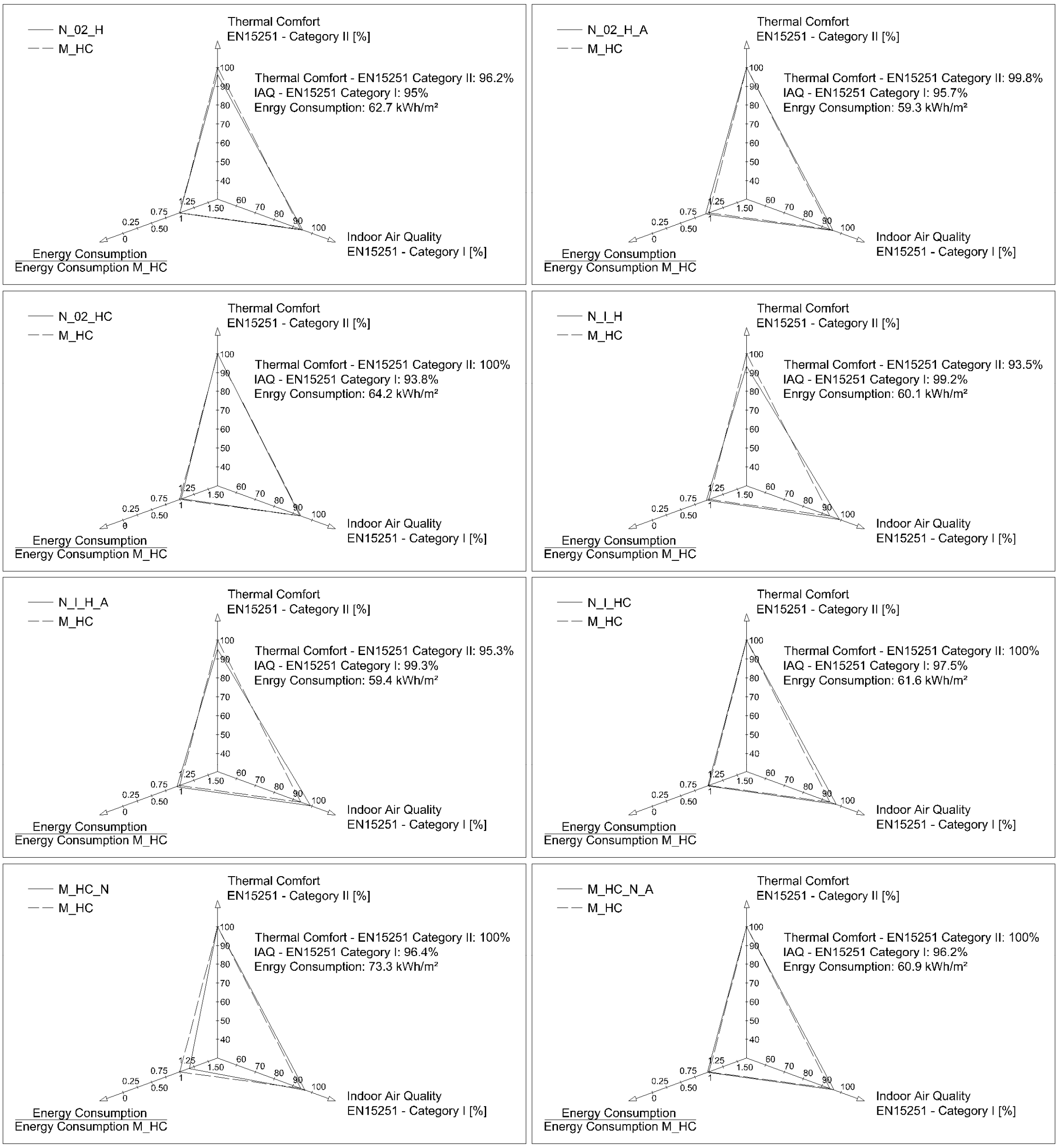

Figure 4: Individual signatures for the case studies in Berlin, reference case: M_HC (Thermal Comfort – EN15251 Category II: 100%, IAQ – EN15251 Category I: 94.1%, Energy consumption: 62.8 kWh/m2).

Copenhagen

For the night thresholds optimization the temperatures taken into account range from 22.0°C to 24.5°C with a 0.5°C increase step. The analysis revealed that for thresholds higher than 23.5°C (included) the night cooling strategy is never applied during the year, while for thresholds lower than 22.5°C (included) the use on night ventilation increases the discomfort, causing the overcooling of the dwelling, and the energy consumption for heating. The 23.0°C threshold can be then considered the best option, allowing the application of the night ventilation only when strictly necessary.

In Copenhagen the risk of overheating during summer is limited to very few days, during most of the year the discomfort is caused by the overcooling of the building. Thus the NE orientation has been selected since it maximize the solar gain, improving the thermal comfort and reducing the energy consumption. During summer, when needed, the solar gain can be decreased by activating the solar shading.

As the windows are almost always closed during night, no improvement at all in the thermal comfort have been obtained by increasing the thermal mass, then, in accordance with the other locations, a 0.20 m concrete layer thickness has been chosen.

For the daytime ventilation a 23°C threshold has been chosen.

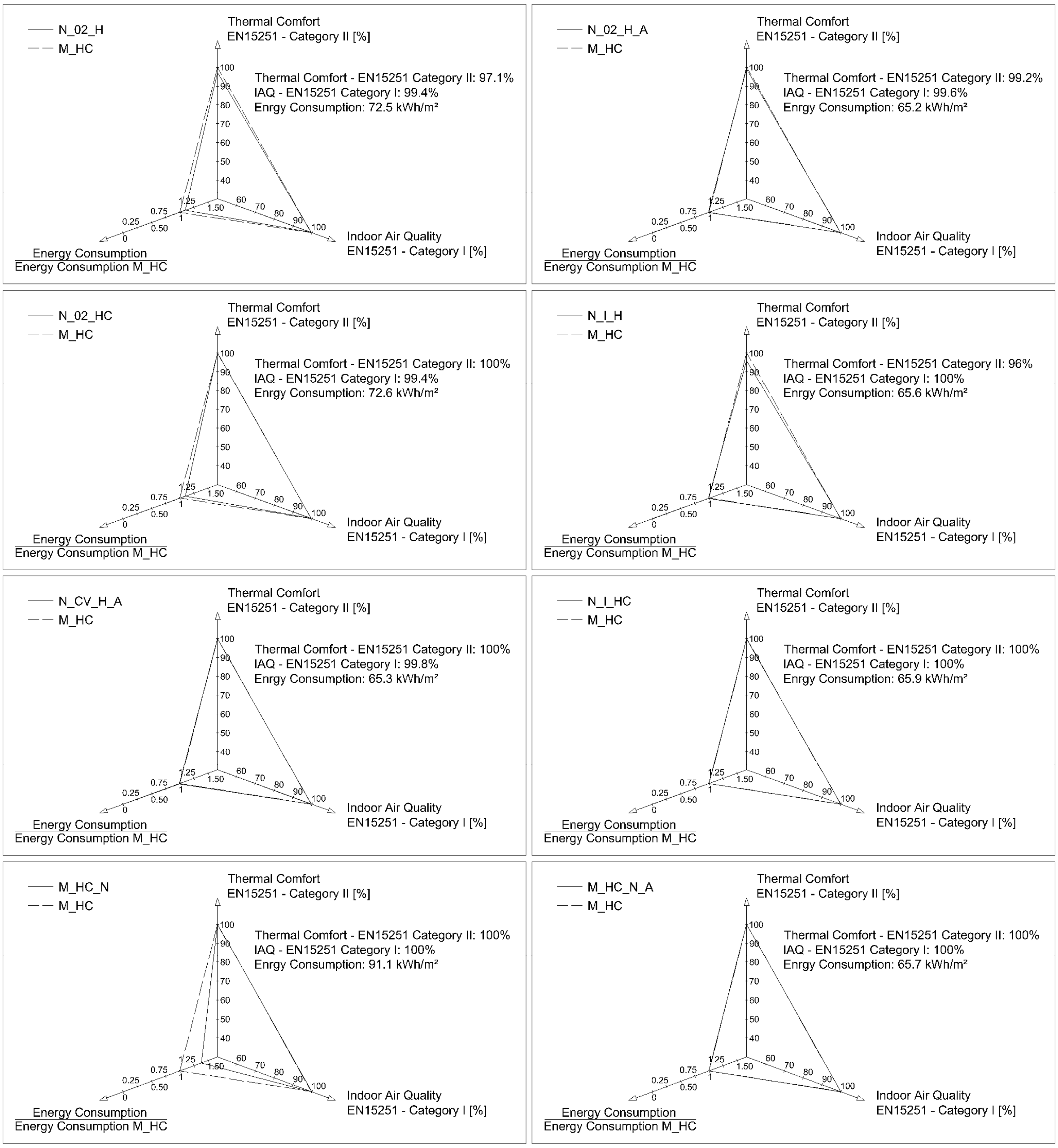

Figure 5: Individual signatures for the case studies in Copenhagen, reference case: M_HC (Thermal Comfort – EN15251 Category II: 100%, IAQ – EN15251 Category I: 99.7%, Energy consumption: 66.1 kWh/m2).

With the selected parameters the mean air velocity for the non-increased air velocities cases is 0.16 m/s while the mean air velocity for the increased air velocities ones is 0.30 m/s. The natural ventilation period starts on April 30th and ends on September 17st, the natural ventilation strategies are then applied for 141 days over 365 (39% of the year).

As a matter of fact the night ventilation is almost never used: over the entire natural ventilation period the windows are opened for 13% of the nights in the N_02_H scenario and for 1.5% of the nights in the N_I_H one. Indeed in Copenhagen only 0.6% of the energy consumption is for cooling, it means that the cooling load is almost zero. This is the reason why all the passive cooling strategies, being too efficient for the local climate conditions, reduces the thermal comfort (from a minimum of 0.8% for N_02_H_A, to a maximum of 4.0% for N_I_H), sometimes increasing the thermal comfort (e.g. by 9.7% for N_02_H).

The M_HC_N_A solution has been designed as a thermostatically controlled one, but the energy consumption reveals that the mechanical cooling system is never turned on and there is no extra heating demand: the night cooling seems then capable alone to reduce the cooling load to zero without causing overcooling. The observation suggested us to test one more ventilation strategy in Copenhagen, namely N_CV_H_A. The strategy is based on daytime mechanical ventilation and automatically controlled night cooling, but the windows can be opened, according to the occupants comfort, for a short period of time (15 min.) in the early morning (8:00 a.m.) and when the occupants go back home (17:00 in the afternoon) for airing the dwelling. The daytime natural ventilation achieved in this way does not provide ventilative cooling, but allows the occupants to better control the indoor environment. The solution performs excellently: the building is for 100% of time in category II (thermal comfort), for 99.8% of time in category I (IAQ) and requires 1.3% less energy than the thermostatically controlled one. Beig capable to prevent both the night time overcooling that affects the non-increased air velocities scenarios, and the draft sensation that affects the increased air velocities ones, it is the only passive cooling solution which does not decrease the thermal comfort.

Discussion and conclusions

The project here reported investigated the potential energy saving and summer comfort improvement that can be achieved by mean of passive cooling strategies such as solar shading, ventilative cooling and night cooling. In general the passive approach seems capable to ensure a good indoor environment in terms of high IAQ and prevention of both overheating and overcooling, as well as a reduction in the energy consumption.

In Athens the increased air velocities, being capable to maintain the mean air temperature within an acceptable range, are efficient in limiting the use of night cooling without the need of an automatic controller. The combination of the two strategies seems then capable to achieve a very good thermal comfort.

For Rome, Berlin and Copenhagen the same combination of daytime increased air velocities and night cooling turned out to be too aggressive, causing some overcooling and an increase in the energy consumption for heating that ranged from the 1.6% of Rome to the 9.7% of Copenhagen. A moderate approach showed good results.

In Rome and Berlin a constraint to both the daytime air velocity and the nighttime air flow rate was capable to provide a very good indoor environment, even more comfortable than the one obtained by mechanically cooling the building. For colder climates, such as the one of Copenhagen, the best performance on thermal comfort (100% of the time in category II) was obtained with the use of the night cooling strategy only.

The hot Mediterranean climate of Athens and Rome presents a very high cooling load, thus the adoption of the passive technique leads to a consistent reduction in the energy consumption (83% for Athens and 65% for Rome). In Berlin and Copenhagen the reduction in the energy demand (5.6% and 1.3% respectively) corresponds entirely to the energy used by the chiller, showing that the strategy is capable to lower the cooling load, in particular in Copenhagen the selected strategy is capable to reduce the cooling load to zero.

When the IAQ is looked at, the natural ventilation performs much better than the mechanical one in all cases. The air flow rate ranges from 1.1 ach (Copenhagen) to 4.8 ach (Athens) during nighttime and from 0.7 ach (Copenhagen) to 1.7 ach (Athens) during daytime.

In general, the scenarios that performed best on all three parameters (thermal comfort, IAQ and energy), were the ones based on natural ventilation.

Acknowledgements

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Hybrid ventilation and cooling technics for the new Nicosia Townhall

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Abstract

The new Nicosia Town-hall is a very particular building. On the site where it is built, important antiquities were discovered during the first day of construction and the whole design was completely modified to fit to the new situation. The archaeologists continued to excavate 2/3 of the entire site and created an archaeological park in the centre of the town. The building area was constraint to the remaining land, and co-exists with the uncovered findings.

As a consequence, the building was split into 5 smaller units, 4 office and public service buildings and a municipal hall. Foundations were changed to a combination of piling between findings and large raft slabs sitting above the level of undisturbed ground. The design of office buildings followed the rules of bioclimatic architecture to meet the passive standards and the building is on process for Minergie® (Swiss) labelling. Massive buildings, naturally ventilated and cooled, offer a natural comfort with minimum energy consumption.

The hall follows completely different design principles. Above a large slab, sits a light structure and glazed façades, allowing maximum view and contact between the interior- where the municipal council meet- and the surrounding archaeological park.

According to good practice design rules, this building would be a bad building, especially in a hot climate. A completely glazed cube would certainly overheat and consume a lot of energy. To avoid this, the design proposes a hybrid ventilation system using a sophisticated natural air path to cool naturally the building. Several distinct air streams using smart stack effect path ventilate the building differently according to the time, to the use of the building and to

the external climate, in order to reduce mechanical ventilation and air conditioning hours of use to the strict minimum. Ventilation system shifts automatically from natural to mechanical offering maximum comfort with minimum cooling, heating and fan energy consumption.

This original hybrid ventilation and cooling system made possible the particular architectural expression of the building with low energy consumption. The whole building complex makes a harmonious eco-neighbourhood with low-energy-consumption, comfortable interiors and friendly shaded, wind-protected public spaces, open to the town, where urban life meets cultural heritage.

The article explains the ventilation concept, bioclimatic principles and the simulated comfort and energy performances.

Keywords

Potential for ventilative cooling strategies; design approaches for ventilative cooling and case studies; summer comfort and ventilation; innovative ventilation.

Introduction

The new Nicosia town hall (Cyprus) is not a simply green building showing several bioclimatic architecture principles. It is the first contemporary building in the island applying all the bioclimatic principles, which are necessary to meet the passive building standards (primary energy consumption for heating, ventilation, air conditioning and hot water production less than 30 kWh/m^{2}y).

The "town hall" is not a single building. Archaeological findings restricted the available land to the 1/3 of the initial available surface and the unique initial building is split to smaller units in order to fit in the remaining complicated site. 4 office buildings and a municipal hall, able to receive the council meetings in presence of 250 people, form a neighbourhood in Nicosia old town, just 100 m from the green line, where the war divided the city several decades ago.

Bioclimatic and sustainable architecture starts from the site use. The buildings respect the old town scales and they are integrated in the archaeological site not only preserving cultural heritage, but also making it available to the population, through walk paths, squares, and shaded patios. They create a public space with a social environment, where urban life meets culture and municipal services, in a marginalised district of the city, where social life is stopped for many years now. Orientation and disposition of the buildings group similar uses, separate polluting and noisy activities from office spaces, create natural shading to public space and neighbouring buildings.

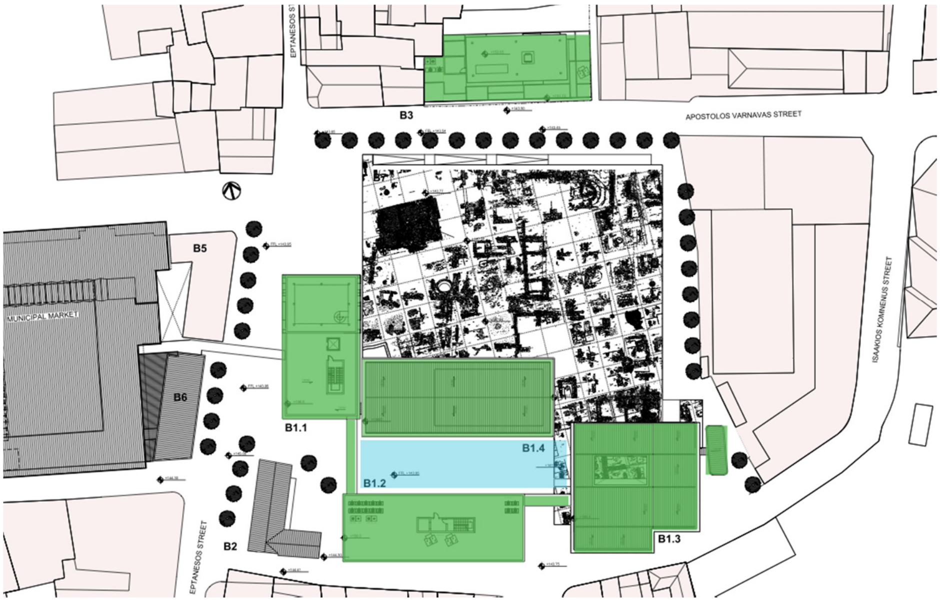


Figure 1: Instead of a single building occupying the whole land, a family of small buildings around the archaeological findings create interesting bioclimatic potential

Figure 2: Panoramic virtual view of the building complex from the green roof of building 1.3; view of the municipal hall from the antiquities, view of the shaded patio between buildings B1.2 and B1.4. Only building B1.3 is finished. The other buildings are under construction

Buildings B1.1, B1.2, B1.3 and B3 are massive and well-shaded office and public service buildings. Building B1.4 is the light structure, fully glazed 10 m height municipal hall. The square between buildings B1.2, B.3 and B 1.4 is shaded, providing solar protection to the three buildings and especially to the south glazed façade of the municipal hall. Building B1.3 is sitting on 10 pillars over the archaeological site and it has an interior yard making available natural light and ventilation to the core of the building.

Regarding ventilation strategies, office buildings function with purely natural ventilation with specially designed vents. The municipal hall runs with a sophisticated hybrid system, with natural ventilation assuring air movement for free night cooling and mechanical ventilation distributing heat and mechanical cooling.

Bioclimatic design, of office buildings

The basic condition for a comfortable thermal environment of offices is good insulation and solar protection. In south climatic conditions, with very hot summers and relatively cold winters, energy performance is necessary for both winter and summer seasons. A well-insulated building, with reasonable glazing orientation and solar protection, consumes 15-25% of the total thermal demand for heating and 75-85% for cooling. In the past, where buildings where not insulated, this ratio was inversed, with heating demand representing more than 75% of the total demand. This is illustrated on table 1 and figure 3.

Insulation	Heating demand	Cooling demand	Total demand
A. 0 cm, single glazing	149 (78%)	43 (22%)	192
B. 4 cm, double glazing - 3.5 W/m^{2}k	21 (25%)	64 (75%)	85
C. 10 cm, double glazing - 1.3 W/m^{2}k	8 (19%)	34 (81%)	42

Table 1: Heat and cooling demand of 3 building scenarios (building B3) simulated dynamically with DIA+ software: A-construction as it was usual before the entry into force of the energy Low (without insulation and with single glazing), B-with insulation and glazing respecting the minimum requirements of the local energy Law, C-with optimised insulation depth, high energy performance windows and with static solar shading, meeting passive standards. For all scenarios, there is no special free cooling strategy

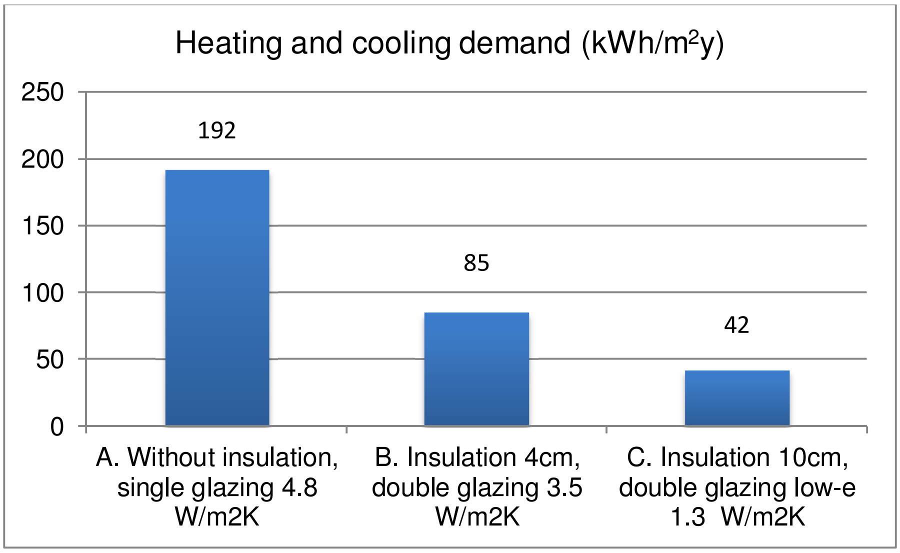


Figure 3: The graph shows heating and cooling demand, according to the 3 basic scenarios. Cyprus Energy Law reduces the energy needs to the 44% of those of a building, without any care for thermal insulation. Additional thermal insulation (10 cm instead of 4) and more insulated windows (U value 1.3 instead of 3.5), with 60 cm passive solar protection on the south façade, reduces the energy needs to the half of those of a building meeting the minimum legal insulation values. Passive buildings (C) have only 22% of the energy needs of non-insulated buildings (A)

Thermal insulation

After several optimisation dynamic simulations with DIAL+ software, we decided that the optimum insulation characteristics to meet the passive standards are 10 cm of rockwool for the roof and the facades, 5 cm for the periphery of the building and a U value of the windows of 1.3 W/m^{2}K. Thermal insulation on the ground does not change anything, as the mean ground temperature in Cyprus is high. A careful analysis and treatment of every joint between constructive elements minimises thermal bridges and heat losses in winter. External insulation gives the advantage of thermal mass inside the building.

Figure 4: Outside thermal insulation with minimised thermal bridges is composed mostly by 10 cm rockwool. Foundations are insulated with 5 cm xps. In some special cases it was necessary to use internal thermal insulation with 10 cm rockwool

Thermal mass

An apparent cladded concrete ceiling and a floor composed with 4 cm anhydride screed over the concrete slab and rough concrete screed, offer a high thermal mass, absorbing excess heat during the day and restoring it during night. This optimises the use of internal heat gains during winter and reduces the pick temperature during summer.

Figure 5: Floor and ceiling are composed by massive materials with high thermal mass

Solar shading

As building B3 is north - south oriented, static solar protection of 60 cm is sufficient. As we see from table 1, comparing scenario B with scenario C, solar protection reduces cooling demand in summer by nearly 50% (additional wall and window insulation plays a small role for the cooling demand).

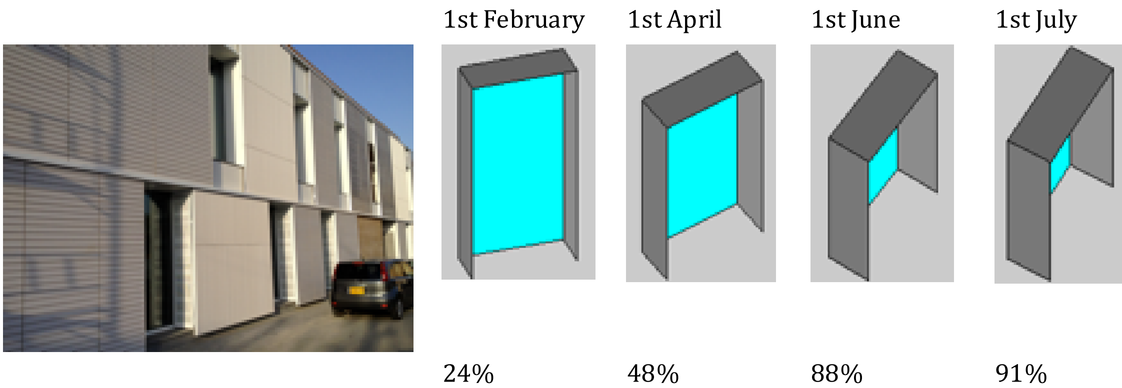


Figure 6: The south façade (15° South – West) at 14:00 is shaded with 60 cm static top and side solar protection. The choice of a glazing with g value of 0.4 is a good compromise between summer solar protection and winter useful solar gains. The façade white ceramics is a robust, inert, self-cleaning material with low sun absorption, offering low façade temperature rise to prevent overheating of incoming air during summer. The table shows the solar shading during the year. In winter solar shading is minimum and in summer maximum

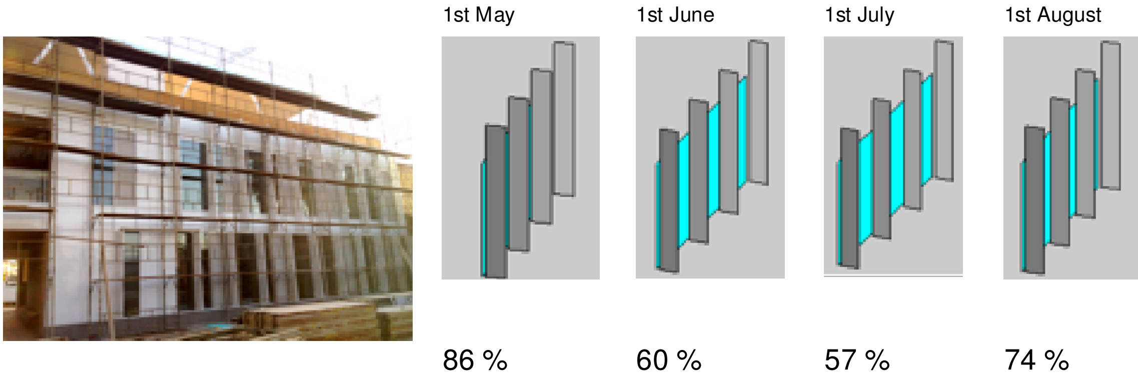


Figure 7: As the north façade is 15° oriented east, early in the morning during June and July, sun hits the façade during 2 hours – 05:00 – 06:00. Some solar shading is necessary to protect north offices early in the morning. The table shows the shading efficiency of a 60 cm vertical static solar protection at 6 h in the morning

Natural and artificial lighting

In a context where air conditioning represents ¾ of the total energy demand, internal gain control is capital. Lighting counts for around 50% of the installed electric power. The question is how many hours the users will need to turn it on. High natural lighting autonomy is easy to obtain in a climate with 300 days of sunshine per year. The designer should even pay attention so that there is not excessive light coming from windows. A delicate equilibrium should be found between daylight needs, winter solar gains for passive heating and summer solar gain control. A north office of 4 m large has 2 modules of glazing of 140 X 300 (70% glazing), while a south oriented one a single module of the same dimensions (35% glazing).

Figure 8: Natural light autonomy, simulated with DIAL+ software between 75 and 85% during working hours

Several optimisation measures increase up to 30% the natural light autonomy: white colour walls and ceiling; reduction of glazing frames and rise of the window top, up to the ceiling; white colour external shading; glazing with high luminance transmission (g value 0.4, LT 0.7); light wash of the lateral white wall with the glazing moved to the side; These measures provide a natural light autonomy between 70 and 85% during working hours

In addition to natural light maximisation, artificial lighting uses high efficiency luminaires with installed power < 12 W/m^2. Light is automatically switched off when the office is empty.

Ecological materials and health

The objective is not only a high-energy performance and comfortable building but also a building using environmentally friendly materials, creating a healthy interior environment. Concrete for the structure, plain wood for the façade structure and the window framing (non treated larch), gypsum panels and rock wool for interior partitions, rock wool for the thermal insulation and ceramics for façade exterior facing, are the main materials used in the building. Joint synthetic substances are avoided. Instead of surface treatments and painting, the natural material colours create the chromatic synthesis and the aesthetical language. Wood is just oiled with linseed/turpentine/TiO mixture, avoiding varnishes and synthetic substances. The life cycle of the building elements is high, with low maintenance needs. Cyprus climate is very difficult for the exterior materials exposed to sun, dust and rain. The exterior façade facing with ceramics, white for façades with high sun exposure and coloured for other orientations, is a robust high life-span solution, adapted to the local environment.

There is no material in the building emitting VOC particles. The floor is done with anhydrite liquid screed, which is mineral and inert, offering high thermal mass and avoiding VOC emissions.

Ventilation and cooling strategies of office buildings

Before we adopt a ventilation strategy, we put on the balance 4 aspects: air flow necessary to assure air quality and occupant's health, energy consumption by fans or by thermal losses or gains because of excess ventilation, occupant's wishes / well-being. Some people, influenced by good practice in the North and Central Europe countries, concentrate on the possibility of heat recovery. They a priori consider that mechanical ventilation with heat recovery is a good practice for every climate and for every building use, extrapolating intuitively this conclusion from what happens in the cold climates.

Without excluding any solution, before adopting a ventilation strategy, we answered to 3 questions:

1. what are the wishes of the users and how do they feel in regards to the control of their environment;
2. what is the real impact of different ventilation strategies on heating, cooling and electricity demand;
3. what is the real risk of wrong use and bad ventilation control by the users ?

Question 2 and 3 may have a different answer according to climatic conditions, building function, physical characteristics of construction elements.

Users' wishes and feelings in regard to ventilation systems

The objective was not only a high-energy performance and a comfortable building. A municipal building is a professional tool for public service. Well-being of the users is a key factor on productivity and service quality. Before the building design, the great majority of the municipality personnel answered to a questionnaire about their current indoor environment quality and their expectations from their new place of work. The personnel showed a high degree of environmental consciousness with low CO2 emissions being their second concern. 63 people imagine an exemplary building of natural comfort and only 23 an exemplary fully air-conditioned building. 30% of the people consider mechanical ventilation as problematic. Less than 5% considered natural ventilation from the window as problematic. These results confirm the results of European research, showing higher acceptance and lower building sick syndrome index in naturally ventilated buildings.

Impact of ventilation strategy and heat recovery on energy demand

In order to quantify the impact of different ventilation strategies, we have simulated a typical section of the building with two offices of 4 m large, and 10 m deep representing the total depth of the building from north to south. One office faces south and the other north with an interior buffer corridor zone in the middle. The thermal model considers the whole space as a single zone with dimensions, building thermal characteristics and shading as explained in the previous paragraphs. It considers also standard use conditions and occupation schedules according to the Swiss regulations SIA 2024. DIAL+ software simulates dynamically the solar gains, internal temperature, and natural ventilation airflow, cooling or heating load and energy consumption.

For the electricity consumption we used an optimistic hypothesis of a high efficiency fan, consuming 0.16 W/m^3.h for simple extraction and 0.32 W/m^3.h for a system with heat recovery. We used a high-COP energy system of 4 to calculate electricity consumption.

Column Title	Heating need kWh/m²	Cooling need kWh/m²	Total thermal demand kWh/m²	Fan electricity demand kWh/m²	Total electricity demand kWh/m²
1. Mechanical, 36 m^3/h.pers	6.7	35.5	42.2	1.5	12.1
2. Mechanical + heat recovery 80%	3.4	33.1	36.5	3	12.1
3. Natural with 10 cm tilted window	6.0	31.8	37.8	0	9.5
4. Natural with 10 cm standard window	9.6	35.2	44.8	0	11.2
5. Excessive ventilation - 50 m3/h.pers	7.5	36.2	43.7	0	10.9
6. Natural with night ventilation	9.6	16.5	26.1	0	6.5

Table 2: Heating and cooling and ventilation thermal and electricity demand

From this table we can learn several not very well known truths.

1. Heat recovery in south climates is not always energy effective.

As we can see from the results in the second line of table 2, heat recovery may reduce thermal demand by 13% (5.7 kWh/m²y), but it consumes 3 kWh/m²y of electricity instead of 1.5 of a simple extraction system and 0 of a natural ventilation system. If we measure electricity to produce 5.7 kWh of heat or coolness with a system of COP = 4, the heat recovery system does not recover even the energy that it needs to run. It could be energy effective for cases with direct heating or very low COP cooling, or for cases of non-insulated buildings with very high demand for heating as it was the case before the Energy Law.

2. Controlled natural ventilation is the most energy effective strategy for office buildings.

We can see this, if we analyse the results of the 3rd line. Controlled natural ventilation with a well-designed window, allowing small openings during hot or cold hours, does not spend more energy than mechanical ventilation. If we take into account saved electricity to run fans, the total balance is positive for natural ventilation, with 9 kWh/m²y of energy consumption instead of 12.1 of a mechanical one. Dynamic simulations of the airflow showed that a vent of 40 cm large by 140 cm high, in tilted position, provides almost always the necessary airflow for pollutant evacuation without excessive ventilation.

<u>3. Excess ventilation, if it is reasonable, does not destroy the energy balance of the building</u>

Minergie® standard counts for natural ventilation an overestimated 50 m^3/h airflow rate (simulation of line 5), but if we take a pessimistic hypothesis considering that the window is kept open all the time at 10 X 300 cm, independently of how cold or hot is the outside temperature, we can see that heat demand is not excessive. It creates extra 6% losses (44.8 kWh/m^2y of thermal demand instead of 42.2). This is because extreme temperatures (38 to 42°C in summer end -2 to 5°C in winter) take place very few hours during the office working hours. Most of the time, temperature difference is moderate, not creating excessive thermal losses.

<u>4. The most interesting energy saving potential lies in night ventilation free cooling.</u>

As we can see from the results of line 6, a night cooling strategy reduces cooling demand by 56% (16.5 kWh/m^2y instead of 35.5) and the total energy demand by 38%. **This is the key energy potential for south climates.** It is equivalent of the passive heating for cold climates.

As a result of these findings, we concentrated the efforts to design a smart, simple and user-convenient window. This window should provide easy and intuitive control for limited ventilation during office hours and high airflow rate ventilation during night in summer. In addition, users should not thing how to ventilate; they have just to open a window. The opening should be protected, in order to control the risk of intrusion by undesirable people, animals, insects rain and dust. People should feel safe to leave the vents open during night without any concern.

Natural ventilation design and ventilation strategies

Vents are vertical, opening on the whole room height, in order to maximise stack effect. With 5°C temperature difference between inside and outside, a 40 by 300 cm vertical vent creates a stack effect of 611 m^3/h, while the same vent in horizontal position 300 by 40 cm creates only 223 m^3/h. The right disposition of the vent opening may boost ventilation airflow by 275%! High airflow rates are necessary only during night. During the day only 36 m^3/h per person are necessary. An opening of 40 by 140 cm height may provide 75m^3/h at a ΔT = 5°C and 47m^3/h at ΔT = 2°C. These dimensioning calculations leaded us to divide the high vent in two parts and to make it open right or tilted. The user instructions become simple and easy to understand: "tilt the top vent during working hours winter or summer. During winter, you close it when leave the office and during summer, you open completely one or both vents, according to your cooling needs; you put it back to the tilted position in the morning.”

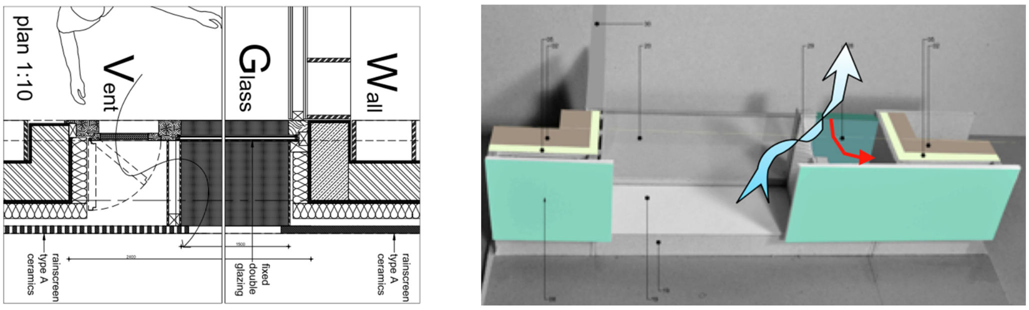


Figure 9: South façade vent. Air enters from the side of the glazing through a perforated protection sheet metal protection

As we can see from the photos of figure 10, lighting openings are dissociated from air vents. This makes it possible to treat correctly the glazed part, hiding frames or any obstacles and divide, protect or hide the vent part. In the south façade, air comes from the side after passing through a perforated sheet metal. On the north light and façade air comes directly after the protection.

Figure 10: Tilted north façade vent, north façade open right, south window from outside, south vent from inside

As we can see on the second picture of figure 10, all offices are equipped with a ceiling fan. This offers the possibility to the users not to use air conditioning up to 28-29°C of internal temperature and use the roof fan instead, reducing drastically the hours of use of air conditioning. It avoids also a wrong use of the window completely open when external temperature is around 30°C and users like wind breeze. If the user wishes an air movement, he may use the ceiling fan, avoiding excessive heat and dust entering in the office.

Ventilation and cooling strategies for the municipal Hall

The municipal hall is a light structure fully glazed building. According to what preceded in this article, bioclimatic architecture should exclude this design. However, the social needs constrained the design team to find special solutions for this building.

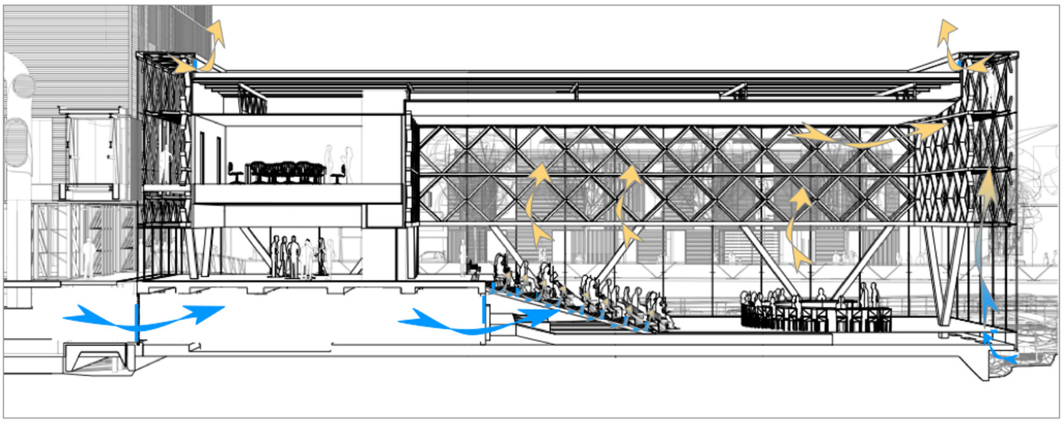
The constraints: light structure without thermal mass; high glazed-façades, difficult to equip with movable solar protection.

The advantages: rare and limited use during the day; a country yard in the south, possible to be shaded, creating a social place to meet and shading the exposed south façade; complete shading from the west building, a free massive technical space, under the bleachers, a high building offering high stack effect for ventilation and allowing a hot buffer zone outside the living zone.

The strategy: create a double skin façade using the acoustic element at the top part of the interior space; use a selective glazing with g value 0.4 to reduce sun thermal load; complete solar shading with an interior movable awing at the lower part of the façade and shade the south external yard; allow openings on to top and on the bottom of the façade canal, able to evacuate solar gains; allow a big opening behind the building for air inlet under the bleachers. We ventilate the building during night to cool the concrete slabs of the technical spaces under the building. Air may also come in the canal on the bottom of the double skin façade, when outside air is cooler than the interior air, in order to evacuate accumulated heat from solar gains locally very early in the morning in the north façade and during the morning in the east façade. When outside air becomes hotter than the interior, it comes in only from the back of the building. It is cooled by the coolness accumulated during night in the thermal mass. An automatic control system opens and closes the bottom and top openings of the space according to the desired strategy.

Simulations of interior temperature and stratification showed that with this strategy the interior climate is never worse than the exterior. The occupied space has the best climatic conditions, profiting from the precooled air under the building. By controlling stratification, we localise hot air in non-occupied spaces (within the double skin canal and on the top of the building behind the false ceiling).

Under these conditions, mechanical ventilation with cooled or heated air follows the same path with a part of natural ventilation. It comes to complete heating and cooling, when passive technics are not able to meet the demand. Mechanical ventilation recirculates air and recovers heat when this is beneficial (when stratification is low) but it may function as a hybrid system blowing only cooled air in cases where thermal gains are extreme and returning air is too hot.



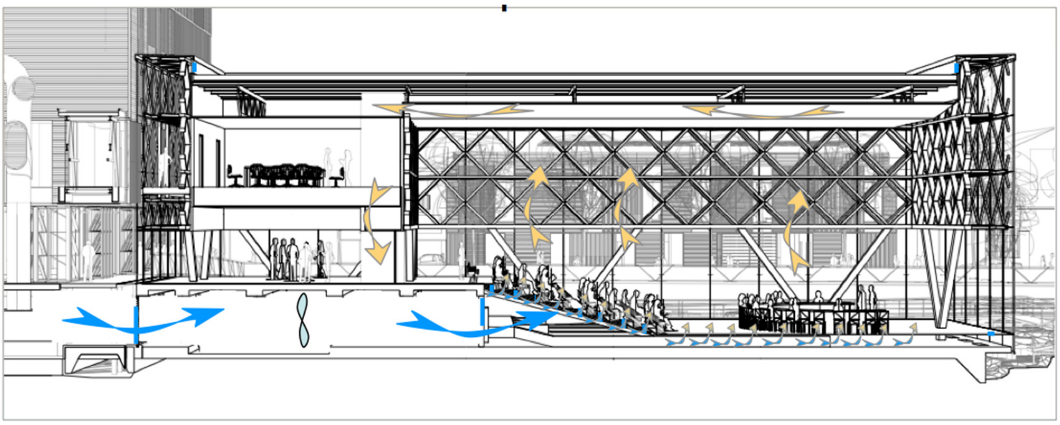


Figure 11: top natural ventilation patterns, bottom, mechanical ventilation patterns

By limiting mechanical cooling during some hours of the year, when the building is occupied and when bioclimatic technics cannot meet the demand, we limit energy consumption drastically and allow passive comfort without air conditioning for a large period of the year.

Conclusion

After thermal insulation and solar shading, free cooling is the key issue for low energy passive buildings in Mediterranean climate. Issues like high air-tightness and heat-recovery, which are important for North and Central Europe, have minor importance for southern Europe, simply because in the mild climates, most of the time the building should be open to benefit from the outside air, which is within or near the comfort zone.

Free cooling by natural night-ventilation is the simplest strategy, but it needs special design attention. Standard windows are not always the best way for natural ventilation. When ventilation strategy depends from the occupant's behaviour, simple smart windows with many opening possibilities, equipped with protections from insects, dust and vandalism, ensure the users and encourage a correct use. For common spaces and large halls, smart automation with the minimum number of openings and sensors is necessary, to achieve a sure result. Nicosia town hall is a good example illustrating both natural ventilation design principles on a very low energy consumption building.