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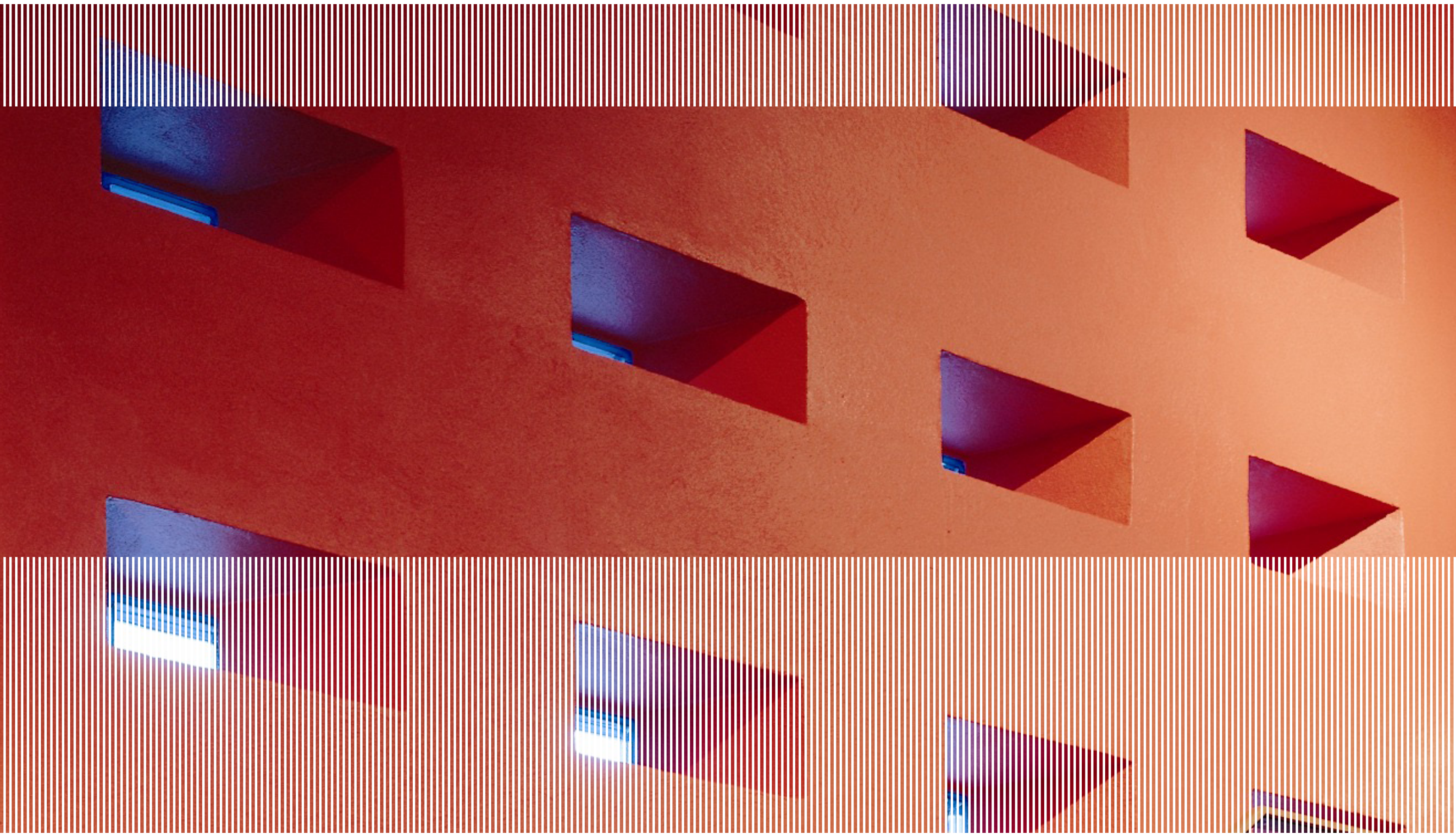
COIN Project report 10 - 2009



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COIN P5 Energy efficiency comfort

Sub P5.2 Comfort

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Preface

This study has been carried out within COIN - Concrete Innovation Centre - one of presently 14 Centres for Research based Innovation (CRI), which is an initiative by the Research Council of Norway. The main objective for the CRIs is to enhance the capability of the business sector to innovate by focusing on long-term research based on forging close alliances between research-intensive enterprises and prominent research groups.

The vision of COIN is creation of more attractive concrete buildings and constructions. Attractiveness implies aesthetics, functionality, sustainability, energy efficiency, indoor climate, industrialized construction, improved work environment, and cost efficiency during the whole service life. The primary goal is to fulfil this vision by bringing the development a major leap forward by more fundamental understanding of the mechanisms in order to develop advanced materials, efficient construction techniques and new design concepts combined with more environmentally friendly material production.

The corporate partners are leading multinational companies in the cement and building industry and the aim of COIN is to increase their value creation and strengthen their research activities in Norway. Our over-all ambition is to establish COIN as the display window for concrete innovation in Europe.

About 25 researchers from SINTEF (host), the Norwegian University of Science and Technology - NTNU (research partner) and industry partners, 15 - 20 PhD-students, 5 - 10 MSc-students every year and a number of international guest researchers, work on presently 5 projects:

- Advanced cementing materials and admixtures
- Improved construction techniques
- Innovative construction concepts
- Operational service life design
- Energy efficiency and comfort of concrete structures

COIN has presently a budget of NOK 200 mill over 8 years (from 2007), and is financed by the Research Council of Norway (approx. 40 %), industrial partners (approx 45 %) and by SINTEF Building and Infrastructure and NTNU (in all approx 15 %).

For more information, see www.coinweb.no

Tor Arne Hammer
Centre Manager

Summary

The case analysed represents a standard size landscape office room in Oslo climate. The office envelope and system properties are in compliance with the TEK (2007). The study concentrates on the thermal comfort conditions in summer and two cooling systems are considered, one representing an idealised room cooling system and one representing a typical ventilation cooling system. Three acoustic solutions are considered: conventional horizontal panels, micro perforated resonant box, and vertical baffles.

With the idealised room cooling system, micro perforated resonant absorbers covering the entire ceiling requires the same room cooling energy as the conventional insulating panels covering 30% of the ceiling. Vertical baffles with a sound absorbing area equivalent to fully covering the ceiling area are expected to have the same effect on room cooling energy as horizontally mounted panels covering 40% of the ceiling area. However, when accounting for room cooling energy and fans operation at day and night the differences between the acoustic solutions are levelled off, and the overall effect is small compared to the effect of using or not using the night flushing.

With the ventilation cooling system, night flushing proved sufficient to fulfil the legal comfort requirements on operative temperature, without use of direct room cooling. Concerning both the energy demand and the thermal comfort conditions, there is no significant difference in the performance of the various acoustic solutions.

A critical look at the results lead to the conclusion that the framework for calculations (NS3031, 2007) was suboptimal for estimating the effect of concrete's thermal mass. A more appropriate approach would require a finer description of the ventilation system and its control strategy both at day and night, as well as of the internal load and solar shading profiles.

Table of contents

1 Background

This subproject 5.1 fits into the wider context of the COIN project 5, which consists of three subprojects:

- *5.1 Energy efficient buildings and construction*
  Main objective: to find the optimal design of office buildings that realize the potential for energy savings by utilizing thermal mass (TM).
- *5.2 Comfortable buildings and construction*
  Main objective: to develop acoustic solutions that also realize the potential for energy savings by utilizing thermal mass.
- *5.3 Case studies*
  Main objective: to develop and document energy efficient building solutions and indoor climate where passive solar design and concrete as activated thermal mass is a major contribution to the solution.

Subproject 5.1 focuses on the amount, placement and design of exposed concrete surface area, as described in the CTR_SP5 (2008):

"Perform calculations and analyses to assess different office room designs with respect to:

- TM's potential to reduce overheating and undesired temperature swings in a typical office
- TM's potential to reduce energy demand in a typical office building
- Architectural possibilities using exposed TM in an office space
- Acoustical consequences and solutions in offices with high share of exposed TM

Special emphasis is put on analysing the need for exposed concrete surfaces for energy savings and the trade-offs with respect to acoustical and architectural considerations. Different acoustic element designs will be analysed, in cooperation with sub-project 5.2. The work will also be related to the case studies of sub-project 5.3. This activity should prepare the ground for making specific guidelines for office building design and to reveal the need for new product development for improved passive thermal mass utilization."

2 Introduction

The state of the art report on thermal mass concepts (Haasse and Andresen, 2007) makes a thorough review of available literature on the subject. The work performed for the SP 5.1 in this phase is based upon acknowledgement of those concepts but no direct reference is made to literature. What this work adds new to the existing body of knowledge is the analysis of a building built according to the most recent construction standard in Norway (TEK, 2007) and the comparison of conventional and non conventional acoustic solutions in terms of their consequences on thermal comfort and energy demand.

In buildings without active local room cooling, the supply of fresh air from the ventilation system (i.e. at temperatures of 17-18 °C) may prove sufficient to keep the room temperature within the desired range. This of course depends on the maximum flow rate allowed, which in turn is again dependent on comfort requirements. Too high flow rates may cause uncomfortable air movement in the rooms, perceived as chilly drafts, and/or may bring humidity out of the comfort range.

Night time ventilation, also named night flushing, is meant to cool down the thermal mass, therefore creating the preconditions for effective heat absorption by the thermal mass the following day. It can be classified as a form of *passive cooling* if the air transport is by means of natural ventilation. However, when the ventilation is provided mechanically there is some electricity consumption for operating the fans. The matter is whether or not the energy savings induced by night ventilation is worth the energy spent to operate the fans.

This is not just a matter of energy balance, but rather a matter strongly related to thermal comfort. If the indoor temperature swings are restricted to a small range (i.e. below 22 °C) by some form of active cooling, then the thermal mass would not be activated, or "charged", and the building would behave like a thermally light building. Thermal mass in this case is not used and therefore cannot give energy savings. If the indoor temperature is allowed to swing more (i.e. below 26 °C) then the effect is increased. Night-time cooled thermal mass is "discharged" in the morning and so it is ready to effectively absorb the heat generated internally and the heat wave coming from outside during daytime. The effect is a lower and more stable indoor temperature during daytime when compared to a light building. If the thermal mass is not cooled enough, not fully "discharged", then its effect is reduced and the building behaves as if it is thermally lighter. Hence, the extra electricity consumption caused by night cooling should not be regarded as unnecessary energy expenditure but as an indispensable part of the cooling strategy without use of active room cooling.

Similarly, there are contrasting needs for acoustic and thermal comfort on covering the ceiling surface. On one hand, the concrete ceiling should be exposed to the room air to allow the thermal mass absorbing excess heat at daytime and reduce temperature swing and cooling demand. On the other hand, there is the need to install sound absorbing elements on the ceiling to satisfy acoustic comfort requirements.

The purpose of this work is to estimate and compare the effect of night flushing and different acoustic solutions on the thermal comfort and the cooling energy demand of a landscape office in summer.

3 Method

The calculations are performed using "EnergyPlus", a software tool for dynamic analysis of whole building energy and indoor climate performance, EnergyPlus (2008). The calculations are performed with a time step of 15 minutes.

The case analysed represents a standard size landscape office room. The office category was chosen because of its high internal gains that may give rise to overheating problems in summer time. The landscape typology was chosen because in such spaces there is the need to cover, all or in part, the ceiling with sound absorbing elements so diminishing the amount of directly exposed thermal mass.

The study concentrates on the thermal comfort conditions in summer, where three levels of thermal comfort requirement are considered. Two cooling systems are considered, one representing an idealised room cooling system and one representing a typical ventilation cooling system. In a first step it is estimated the effect of night flushing the thermal mass. In a second step it is estimated the effect of partially or totally covering the ceiling's thermal mass with three different acoustic elements.

3.1 Climate

The reference climate is that of Oslo. **Figure 3-1** shows the distribution of outdoor air dry-bulb temperature, May to September, for the reference climate of Oslo as from the IWEC (International Weather for Energy Calculations) format from ASHRAE. This is a format where twelve typical meteorological months have been extracted from the 18 years (1982-1999) of DATSAV3 hourly weather data originally archived at the National Climatic Data Center (IWEC 2009).

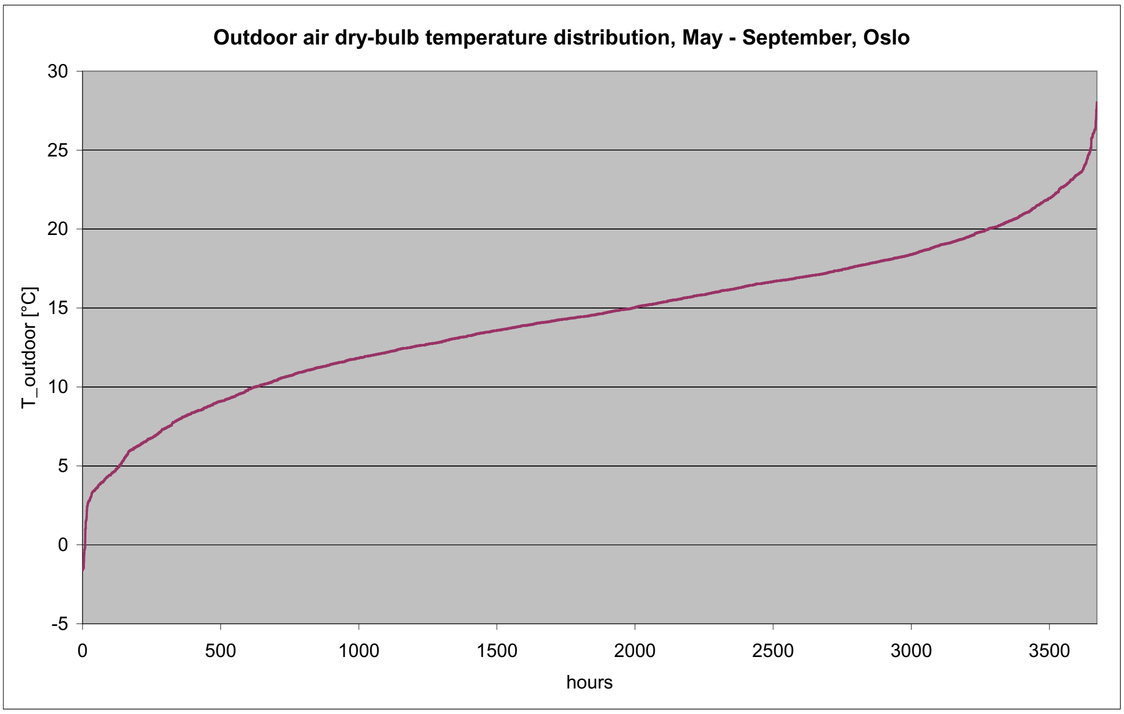

Figure 3-1: Outdoor air dry-bulb temperature distribution in summer for Oslo

3.2 The landscape office

The landscape office room has standard dimensions that can be considered representative of a typical case in Norway; it is rectangular of size 25x12 m with a floor to floor height of 3 m, see Figure 3-2. The room is in an intermediate storey and so it has little dispersions toward above or below; walls facing South and West are exposed to the exterior while the other two walls face internal zones with the same temperature regime and so are approximated as adiabatic. Each of the external walls has a window, and the orientation is such to have the long side exposed to South. A variable area of the ceiling, from 0-100%, is covered by acoustic elements, as shown in example at 50% in Figure 3-2.

Parameters like walls and windows U-value, glazed area, air tightness, internal loads, ventilation system are set according to TEK (2007). All relevant parameters are reported in Table 3–1 It shall be noticed that TEK (2007) also prescribes that all passive measures, including solar shading and night ventilation, should be adopted in order to avoid room cooling. The desired thermal comfort in summer should be achieved simply by means of the ventilation cooling effect.

The simulation period goes from the 1st of May to the 30th of September, and so the heating demand is not considered and the heating system is normally supposed to be off.

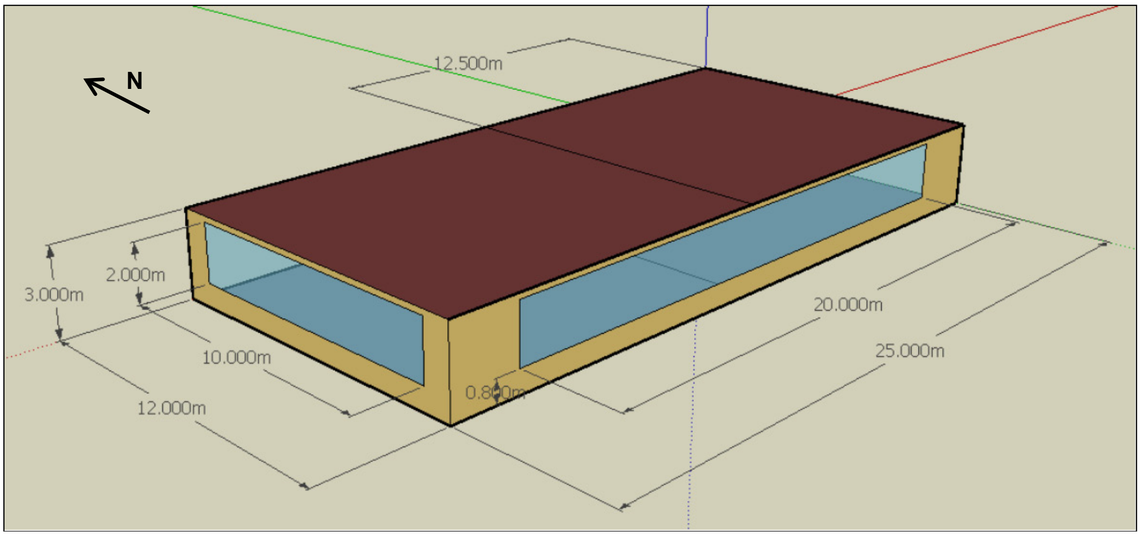

Figure 3-2: Layout of the landscape office room

Table 3-1: Main parameters used in the simulation

Parameter	Value	Notes
Envelope		
Floor area	300 m^2	
External walls	U-value = 0.18 W/m^2K	20cm concrete on the outside, inside 20cm insulation and gypsum board
Ceiling/floor	Concrete slab 20cm with linoleum	
Windows area	20% of floor area	
Windows	U-value = 1.2 W/m^2K g-value = 0.58	Overall, frame included
Solar shading	Automatic when solar radiation on window is > 200 W/m^2	
Thermal bridges	neglected	
Air tightness	n_{50} = 1.5 ach	Corresponds to an average infiltration rate of 0.1 ach
Internal Loads		
Occupancy	Mon-Fri 07:00-19:00	
People	10 m^2/pers, 40 W/pers	sensible heat only, NS3031 (2007)
Lighting	8 W/m^2	NS3031 (2007)
Electrical equipment	11 W/m^2	NS3031 (2007)
HVAC		
Cooling operating hours	Mon-Fri 07:00-19:00	Only for ideal room cooling system
Ventilation operating hours	Mon-Fri 07:00-19:00	
Ventilation supply air temperature	18°C May and September 17°C June to August	
SFP	Day: 2.0 kW/(m^3/s) Night: 1.0 kW/(m^3/s)	Calculated at nominal flow = 70% of maximum flow, see Eq. (3-1)

It is worth noticing a few things. The solar shading is optimal, so that a minimum amount of solar radiation is allowed into the office room. The heat contribution from people is taken from NS3031 (2007) and is somewhat lower than the reference value given in ASHRAE (1997) for a standard sedentary activity (with 1.0 met and 1.0 clo). The ASHRAE value is 108 W/person and is the sum of both sensible and latent heat; depending on actual air temperature the sensible part is somewhere between 60 and 70 W/pers (as from the Energy Plus Engineering Reference Manual), meaning around 6-7 W/m^2 in this case. The value from NS3031 (2007), hence, corresponds to the sensible heat value from AHSRAE (1997) when assuming an occupation rate of about 60%. Cooling system operation, when present, is specified in NS3031 (2007) to be active 24/7, while here is active only during occupation time. The ventilation system, according to NS 3031 (2007), should operate 12 h/day with daytime flow rate and at night and weekends with a reduced flow rate. Here, instead, the ventilation is completely shut off at night. Nevertheless, when necessary a night flushing is simulated as described later. At weekends the ventilation is always off. The specific fan power (SFP) at night is assumed to be half of the daytime value, even though the night time air flow may be as high or higher. The reason for this is the assumption that at night time a series of components can be bypassed in order to reduce the pressure drop, i.e. heat exchanger and heating and cooling batteries, and/or only the return ductwork need be used (possibly combined with a partial opening of the windows), and the supply fan can be switched off thus avoiding the fan heat gain.

3.3 Ventilation and cooling systems

Two cooling systems are considered, one representing an ideal room cooling system and one representing a typical ventilation cooling system. The analysis with the room cooling has the main purpose to estimate the effect of the different acoustic solutions on both thermal comfort and energy demand, when all other parameters are treated as optimal. The analysis with the ventilation cooling system has the main purpose to estimate the thermal comfort and energy demand achievable with a realistic system that has certain limitations, as on the maximum air flow available.

In both cases the landscape office is supplied of a minimal amount of fresh air that is calculated as follows. According to both KLIMA (2006) and TEK-VEIL (2007), the minimum requirements for supply of fresh air are: one part due to occupancy and one part due to material emissions, depending on the materials being considered as low- normal- or high-emissive. The two parts are specified as follows:

- Persons = 7.0 l/s/pers
- Materials
  - low-emitting = 0.7 l/s/m^2
  - normal-emitting = 1.0 l/s/m^2
  - high-emitting = 2.0 l/s/m^2

Special values apply to toilets, lift shafts and storage rooms but these are not considered here. The required flow rate will depend on the occupation density, which is here taken to be 10 m^2/pers on average, see Table 3-1. Concerning the materials, the building under examination is meant to have as much as possible exposed concrete structure in order to exploit its thermal mass properties. The TEK-VEIL 2007 (pp 90-91) states that concrete can be considered as a normal-emitting material, and so this should be the minimum reference value for the simulation. This would lead to:

- density 10 m^2/pers + normal-emitting materials = 6.1 m^3/h/m^2 (0.51 m^3/s)

This value would eventually reduce to 5.1 m^3/h/m^2 (0.43 m^3/s) if an average occupantion rate of 60% is assumed.
Nevertheless, NS3031 (2007) specifies in Table A.6 that the minimum value to be used for calculations is 7 m^3/h/m^2 (0.58 m^3/s). This value is finally chosen in order to perform calculations that are compliant with the standard NS3031.

Night flushing can be provided via natural, mechanical or hybrid ventilation, as described in each case.

3.3.1 Ideal room cooling system

In this case the temperature in the landscape office is maintained at the desired level by a room cooling system that simulate a perfect, or ideal, system. There is no specification on how such a system is made; simply, at each time step the simulation software calculates the zone (sensible) cooling load and artificially compensates for it in the heat balance equation, so to keep the setpoint temperature. The control is done directly on the operative temperature. Therefore, the value calculated by the software represents the minimum requirement on cooling energy. Simulating an ideal room cooling allows focusing the attention only on the effect induced by the different acoustic solutions described later.

In this case daytime ventilation is assumed to be constant at the minimum level that satisfies the requirements on fresh air for the occupants. In order to guarantee a constant flow rate mechanical fans should normally be used – one supply fan and one exhaust fan in a balanced ventilation system. Outdoor air increases its temperature when passing through the supply fan and the supply ducts, and this increase can easily be approx 0.74 °C for a balanced air handling unit with an SFP of 2.0 as required by the TEK:2007 for daytime operation of non-residential buildings. Nevertheless, this effect is neglected here, and so the calculations will express the minimum requirement on cooling energy.

In this case night flushing is considered in an idealised, yet simplified, way. The reason is, once again, to simulate an ideal situation in which the thermal mass effect can be fully exploited, and the differences between the acoustic solutions evaluated, without bothering about possible limitations of a real system. Night flushing is active when both the following conditions are satisfied:

- indoor air temperature is above 19°C
- indoor temperature is higher than outdoor temperature

The air flow rate is 4 ach, corresponding to 12 $m^3/h/m^2$ (or 1.0 m^3/s for the whole space), which is the maximum air flow that can be supplied through the ventilation system. Ideally, night flushing can be provided by natural ventilation and so have no impact on the energy consumption. However, if night flushing is provided through the ventilation system some restrictions apply. As shown in Eq. (3–1) fan power consumption grows with the cubic of flow rate, as this diverges from the nominal value. Hence, too high flow rates should not be considered suitable for night flushing because of their associated high fan power consumption.

3.3.2 Ventilation cooling system

In this case there is a ventilation system that supplies air at a constant temperature, 18°C in May and September and 17°C in June-August. There is Variable Air Volume ventilation system (VAV) that supplies the minimum flow rate required for fresh air (7 $m^3/h/m^2$), but it is also controlled by room air temperature. When the room air temperature tends to go above a given threshold – 22, 24 and 26 °C in the cases analysed – the flow rate is increased in order to provide more cooling effect. The maximum flow allowed is 12 $m^3/h/m^2$, which represents a typical practice in Norway and a safe value in order to avoid possible uncomfortable draughts in the conditioned space. The simulation software calculates at each time step the exact amount of air flow necessary to maintain the set-point temperature (limited to 12 $m^3/h/m^2$). Therefore, the value calculated by the software represents the minimum requirement on ventilation flow rate, and consequently on cooling energy, as given by a perfect control system. The temperature rise due to the fan, however, is considered in this case; temperature rise is about 0.6°c and 1.6°C when flow rate is at minimum and maximum, respectively.

The ventilation is equipped with a heat recovery unit and cooling battery; the heating battery is supposed deactivated in summer. The heat recovery unit has a temperature efficiency of 70%, in accordance to TEK (2007) requirements. When outdoor temperature is above 15°C the heat recovery unit is by-passed (free cooling mode). The cooling battery is supposed to have infinite capacity, and so is always able to guarantee the delivery of supply air at the desired temperature (17-18°C). Nevertheless, it shall be noticed that the cooling capacity of the ventilation system is limited because of its limited flow rate.

Night flushing is provided through the ventilation system, and therefore its maximum flow rate is limited to 12 $m^3/h/m^2$ (or 1.0 m^3/s for the whole space). Night flushing in this case has more restrictions and is active when all the following conditions are satisfied:

- indoor air temperature is above 19°C
- outdoor air temperature is above 12°C; to avoid condensation problems
- indoor temperature is at least 4°C higher than outdoor temperature; to operate the fans only when cooling can actually be effective

3.3.3 Fan power

The relation between fan power and flow rate can be assumed cubic, provided that flow is fully turbulent. Fan power consumption can be expressed as a function of the Specific Fan Power, SFP, and the nominal flow rate Q_{nom}, as shown in Equation (3-1) (from Høseggen, 2008):

$$P = Q_{nom} \cdot SFP \cdot \left(\frac{Q}{Q_{nom}} \right)^3 \qquad (3\text{-}1)$$

The cubic relation is ideal for a turbulent flow, while the relation proposed in NS3031, Appendix H, for the night ventilation is quadratic. Nevertheless, in that case it is assumed that airflow at night time is reduced to a minimum value. In this study, instead, night flushing will have high airflow – higher than the nominal flow rate – because it is used to cool the thermal mass. Therefore, it is safer to calculate the fan consumption at night with Eq. (3-1) because it approximates in excess.

VAV ventilation systems are usually dimensioned for an air flow rate of about 70% than the maximum allowed; in this case it means that Q_{nom} = 8.4 $m^3/h/m^2$, or 0.7 m^3/s. The SFP shown in Table 3-1 is assumed to be constant. The the flow rate data are obtained as an output from the simulation software while the fan power consumption is calculated in post-processing using Eq. (3-1).

As mentioned, the SFP at night is reduced, for example assuming that night flushing is done bypassing the air handling equipment and operated in combination with a partial opening of the windows, and therefore using only the return air path and so reducing the pressure drop. Hence, only the exhaust fan is supposed to be active and the temperature rise due to the fan is not considered at night.

3.4 Acoustic solutions

The acoustic solutions considered are taken from the report by Hveem (2009), and are here briefly introduced and shown in Figure 3-3. It is out of the scope of this work to discuss the acoustic properties of the different elements; for this purpose reference is made to Hveem (2009). In this paper only the effects on thermal comfort and energy demand are considered. Different acoustic solutions are compared at parity of sound absorbing area, but it shall be reminded that this, at any rate, does not imply that the acoustic performance will be the same for the different solutions.

The horizontal panels, Figure 3-3*a)*, are conventional sound absorbing elements made of porous materials that are both acoustically and thermally insulating. They can be mounted either in retracted parts of the concrete formwork or directly on the surface of a concrete slab. This acoustic element is simulated as a mineral wool panel of 5 cm thickness. The ceiling area covered by these elements varies between 0% (fully exposed concrete) and 100% (fully covered concrete), and the consequent effects are evaluated.

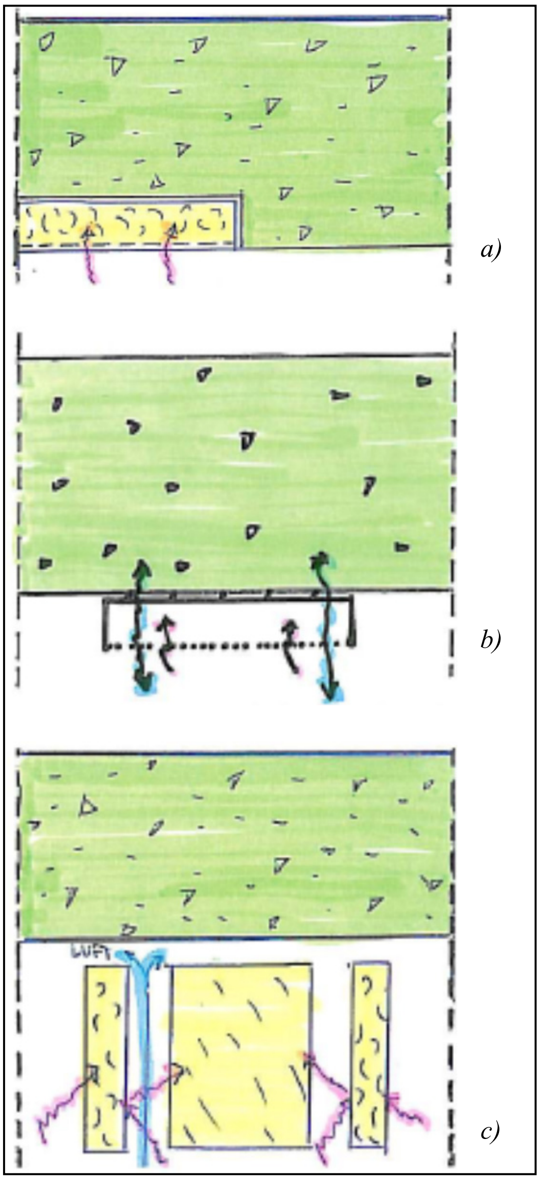

Figure 3-3: Acoustic solutions *a)* horizontal panels *b)* micro perforated resonant box *c)* vertical baffles

The micro perforated resonant box, Figure 3-3*b)*, is a special type of resonator sound absorber with extreme small hole or slot dimensions (some μm). A good absorption characteristic is depending of interaction with a cavity behind and often two panels and two cavities are used to get good properties in a broad frequency range. Resonant absorbers can achieve good results even though they behave poorly than porous absorbers at high frequencies. The absorbers may be produced of metal boards/panels or transparent polycarbonate. Sound absorbers with a good metallic contact with the concrete will transmit heat to the concrete, and so the thermal mass is not strongly de-coupled from the room air. Much depends, of course, on the goodness of the thermal contact between metal and concrete. Simulations have been done supposing no extra thermal resistance at the junctions, which is of course an ideal case. Two versions of this acoustic element are considered:

- profile with partial metallic contact with the concrete
- box with full metallic contact with the concrete

Because of the particular geometry of the cavities behind the micro perforated panel, it is not simple to simulate the thermal behaviour of such profiles or boxes. Wachenfeldt (2007) attempted a solution based on an estimation of the heat transferred by conduction to the back of the cavity, hence to the concrete, and simulated the cavity area as an additional thermal zone. In this work instead, the micro perforated panels are simulated by using their equivalent thermal resistance, as estimated by Wachenfeldt and Zhang (2007), see Figure 3–4. Their work is based on a Finite Element Method (FEM) analysis, and estimates the increased thermal resistance compared to the exposed ceiling being:

- For the profile with partial metallic contact: 0.054 m^2K/W
- For the box with full metallic contact: 0.031 m^2K/W

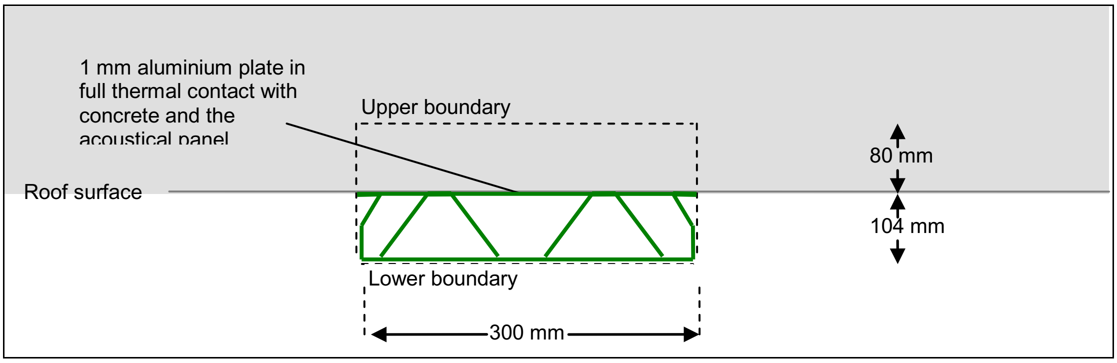

Figure 3-4: Sketch of calculation domain for the FEM analysis for the box with full metallic contact with the concrete slab, source: Wachenfeldt and Zhang (2007)

The vertical baffles, Figure 3-3*c)*, are- essentially the same conventional panels that can be mounted also horizontally. The vertical mounting allows passage of air to the ceiling. Just a small area of the absorbers will be horizontal, and the height, the thickness and the mounting closeness of the baffles will decide the sound absorption properties. A vertical baffle absorbs sound on both surfaces; so, in order to obtain a sound absorbing area equivalent to covering the entire ceiling, the baffles should be mounted at a distance of about 2xH (neglecting their thickness), given that H is the height of the baffle. Considering that the baffle's height may be in the range 30-50 cm, this means that the baffles should be mounted with an average step of 60-100 cm, which allows enough room for lighting ballasts and ventilation ducts. In addition, the baffles should be mounted at a certain distance from the ceiling for not affecting significantly the free motion of air. The room height – 2.80 m floor-to-ceiling – should allow mounting the baffles at a distance of 10-20 cm from the ceiling without particular problems. With this distance it is supposed that air motion, hence the convective heat coupling between room air and the ceiling, is not significantly affected. Nevertheless, the simulation of vertical baffles also considers a case of reduced convective heat transfer coefficient. Vertical baffles will affect also the heat exchange by radiation between the ceiling and the other surfaces. The simulation takes this into consideration by altering the surface view factors, as explained later. Nevertheless, it was not possible to modify how the software couples radiation heat exchange between the ceiling and the internal heat sources, i.e. people, lighting and electrical equipment.

3.5 Thermal comfort requirements

According to KLIMA (2006) and TEK-VEIL (2007) the operative temperature in a working place should not exceed 26ºC. As stated in the same documents, it is acceptable to allow the operative temperature to exceed 26ºC when the external temperature is higher than the one exceeded by 50 hours in a reference year. In more practical terms this is simply interpreted as a requirement for the operative temperature not to exceed 26°C for more than 50 hours of occupation time.

The indications from KLIMA (2006) and TEK-VEIL (2007) are in accordance with the calculation method in EN 15251 (2007). However, some study on the relation between temperature and productivity would suggest that stricter ranges of temperature fluctuation are to be preferred, as in Seppänen et al. (2006).

In consideration of this, the simulations are performed with three different temperature set-points: 26, 24 and 22°C. In the case of the ideal room cooling system the control is done directly on the operative temperature; while with the ventilation cooling the control is done on the air temperature, considering it a more realistic example. For all three set-points the energy demand and the actually achievable operative temperature are analysed.

4 Simulation results and discussion

This sections shows the results from the performed simulations grouped into two major groups: with the ideal room cooling system, and with the ventilation cooling system. In each of the two groups the focus is on the effect of night flushing and the adoption of different acoustic solutions that cover the ceiling (either fully or partially). Results show both thermal comfort parameters as the average operative temperature during occupation hours, its swing and the number of hours when the limit of 26°C is exceeded, and energy demand for cooling and fan operation. Results are shown for the three indoor temperature set-points 22, 24 and 26 °C, and also for a case of free floating temperature, in order to give an insight of what the starting point is in the absence of any cooling system.

4.1 Analysis with ideal room cooling

4.1.1 Night flushing

As a starting point it is worth seeing what would be the situation in case no cooling is applied to the landscape office with fully exposed concrete ceiling. Figure 4-1 gives an overview of the operative temperature profile over the entire summer season. In this case natural ventilation is limited to the minimum amount necessary to guarantee fresh air to the occupants[1]. The possibility to increase natural ventilation as a potential means of passive cooling is not considered. This stated, it appears clear that without night flushing temperatures would soon become highly uncomfortable in the office room, with the exception of early May and late September, even in a Nordic climate like Oslo. This is due to a combination of factors: in a landscape office the internal generation of heat is consistent, this is an intermediate storey, the exterior walls and windows have fairly good insulating values, and the room is exposed to South-West, which represents the worst condition in summer.

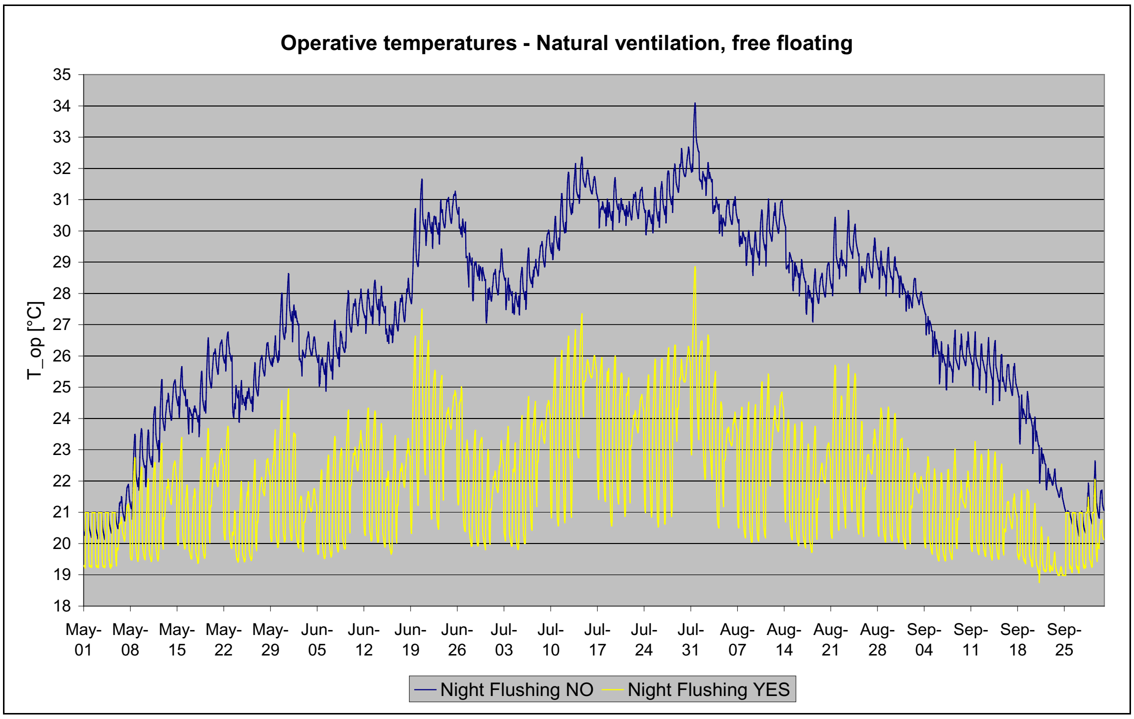

Figure 4-1: Overview of seasonal operative temperature profile for the case with natural ventilation and free floating temperature, with and without night flushing

The situation does improve significantly when introducing night flushing, even though that is not sufficient to meet the minimum requirements on thermal comfort, as shown in Table 4-1.

[1] During the first week of May and the last of September some heating is still needed to avoid low temperatures caused by the natural ventilation.

The average operative temperature is drastically reduced by about 4°C, from 27 to 23°C, and so does the maximum, from 34 to 29. As an effect of cooling the building at night time the temperature swing during occupation increases when adopting night flushing; however, it remains within the range of 4°C recommended by TEK.VEIL (2007). The number of hours when operative temperature is higher than 26°C is also reduced drastically from over 800 to 75, but is still higher than the prescribed limit of 50 hours.

Table 4-1: Summary of thermal comfort results for the case with natural ventilation and free floating temperature, with and without night flushing

Ventilation	Night Flushing	Operative Temperature [°C]		Temperature Swing [°C]		Occupation hours $T_{op} > 26°C$
		T_{op}_avg	T_{op}_max	ΔT_{op}_avg	ΔT_{op}_max	
Natural Ventilation	No	27.2	34.1	1.2	2.7	878
Natural Ventilation	Yes	23.1	28.9	2.3	3.7	75

When introducing a temperature set-point it is possible to see how much cooling energy is needed to maintain it (only sensible cooling load considered here), whether with or without night flushing. This result is shown in Figure 4-2, where it is also shown as a function of how much ceiling surface is covered by horizontally mounted sound absorbing panels, as those in Figure 3-3*a)*.

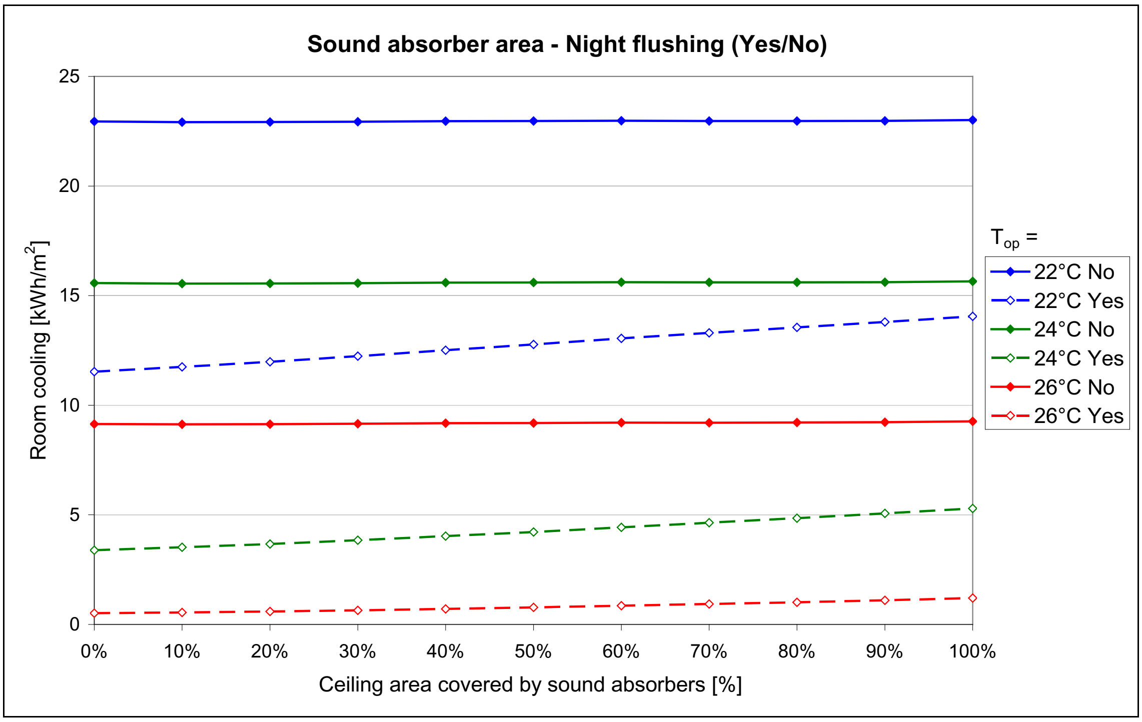

Figure 4-2: Cooling energy demand for various temperature setpoints and with or without night flushing, as function of the ceiling area covered by sound absorbers

Figure 4-2 already shows the most important results of this work. Cooling energy is significantly dependent on the temperature set-point, obviously; but it can be drastically reduced by using night flushing, at least in a thermally massive building like this one. Covering the ceiling with sound absorbers, partially or fully, does reduce the effect of night flushing because it somewhat decouples the ceiling thermal mass form the room air. However, the overall effect is small compared to the effect of using or not using the night flushing. In addition, it shall be noted that cooling demand increase linearly with the increase of covered ceiling area. This is biased by the fact that the software used treats heat conduction in solids as 1-dimensional. If the sound absorbers are mounted in stripes rather than as a compact body, there would be a bi-dimensional effect of heat conduction and storage from the exposed concrete to the concrete behind the panels. Had the software been able to capture this phenomenon the results would have not looked linear, as the cooling demand for intermediate ceiling covering would have been somewhat lower. However, the values corresponding to 0% and 100% would have remained the same.

Night flushing cannot always be done via natural ventilation, or not always natural ventilation can guarantee the desired air change volume, because it is strongly dependant on wind speed, temperature gradient between indoor and outdoor temperatures and so on. When night flushing is done via mechanical ventilation the energy used by the fans shall be taken into account. Furthermore, also the energy used by the fans at daytime shall be considered for completeness. Energy use from the fans, both at day and night time, is calculated as explained in 3.3.3 and assuming that the ventilation system is dimensioned as the one explained in 3.3.2. The resulting overall energy demand for room cooling, day and night ventilation is shown in Figure 4–3 for the case of fully exposed ceiling with temperature set-point of 26°C, with and without night flushing.

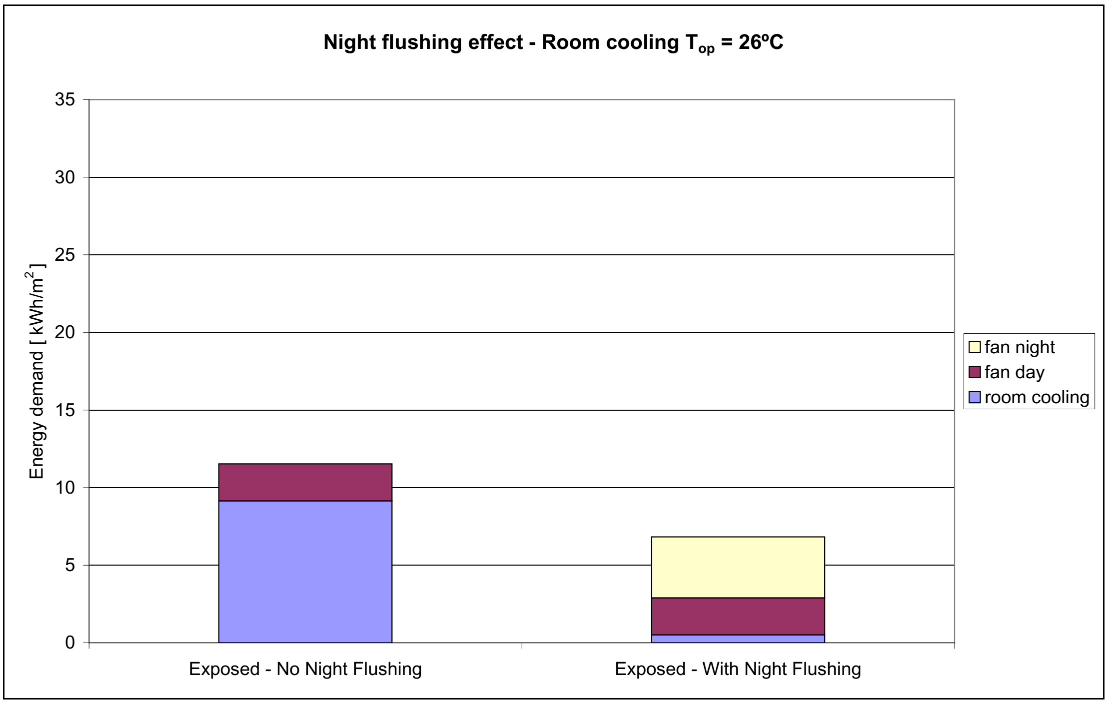

Figure 4-3: Effect of night flushing on the energy demand, whit exposed ceiling and room cooling system with set-point of 26°C

Energy use by the fans at daytime is by definition the same, while room cooling energy is drastically reduced when introducing night flushing, as already noticed. The price to achieve this, however, is a fan's consumption at night that is higher than at daytime, 3.9 and 2.4 kWh/m^2 respectively; this despite the fact that the SFP is assume 1.0 kW/(m^3/s) at night and 2.0 kW/(m^3/s) at day. The reason, of course, is that night flushing has a high air flow rate (12 m^3/h/m^2) and fan power increases with the cubic of the air flow, as this becomes higher than the nominal flow (Q_{nom} = 8.4 m^3/h/m^2). Nevertheless, the overall result is that adopting night flushing – with the given dimensioning of the ventilation system – not only improves significantly the indoor thermal comfort but also reduces the energy demand, from approx. 12 to 7 kWh/m^2.

4.1.2 Ceiling acoustic solutions

The different types of acoustic solutions introduced in 3.4 are evaluated for their effect on the energy demand. Because the temperature is maintained at the desired level by an ideal room cooling system there is no difference in the thermal comfort conditions between one solution and another. On the other hand, the different interaction between the ceiling thermal mass and the room air caused by the different acoustic solutions is captured in the amount of energy that the room cooling system must be able to provide in order to maintain the temperature under control. Hence, it is worth focusing on the case with temperature set-point of 22°C and use of night flushing, as this will show the most amplified differences between one acoustic solution and another. The results are shown in Figure 4-4.

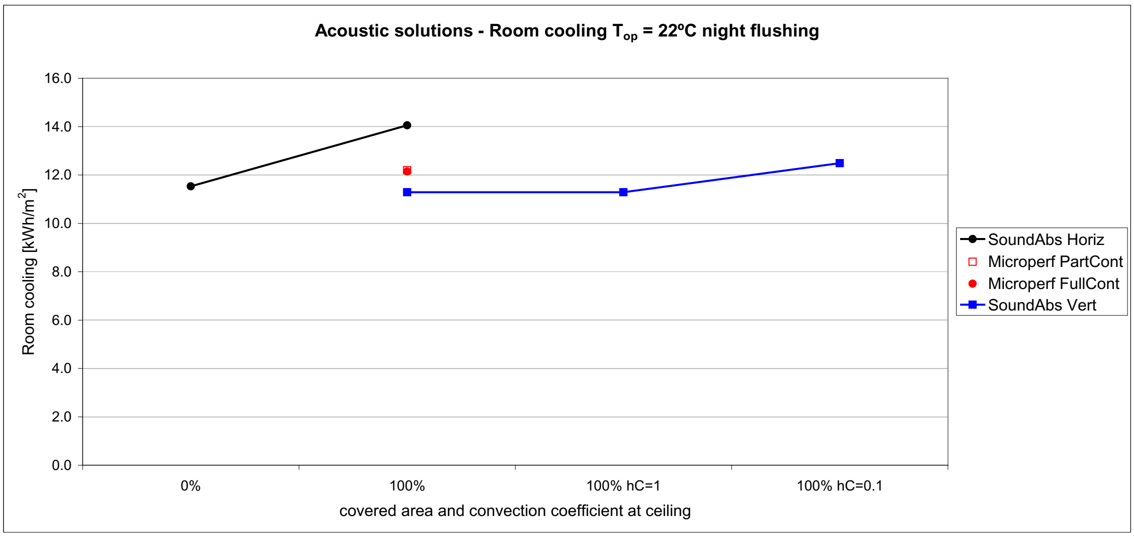

Figure 4-4: Cooling energy demand for various acoustic solutions, with set-point of 22°C and night flushing

The black line represents both the exposed ceiling and the ceiling fully covered with conventional sound absorbing panels mounted horizontally, see Figure 3-3*a)*; it is the same line shown in dashed blue in Figure 4-2. The two red spots represent the two versions of the micro perforated resonant box absorber, see Figure 3-3*b)*. When used to cover the entire ceiling their energy performance is approximately the same as that of the conventional insulating panels covering 30% of the ceiling; with the full contact box performing slightly better than the partial contact profile.

A little more complicated is the case with vertical baffles, see Figure 3-3*c)*, whose results are plotted in blue line. As mentioned, the surfaces view factors has been modified in order to attempt a simulation of the obstructed radiation exchange between the ceiling and the other surfaces in the presence of such baffles. This was done using dedicated pre-processing software available with EnergyPlus. The pre-processing software calculates the view factors for the given geometry and guarantees reciprocity and consistency of the results (energy conservation principle is satisfied). To simulate the vertical baffles' obstruction a fictitious layer was introduced so to modify the pattern of radiation heat exchange between the surfaces. The ceiling can see only the upper side of the fictitious layer, while the bottom side of the fictitious layer is in visual contact with the other surfaces. The view factors calculated were then inserted in the EnergyPlus input file in order to overwrite those otherwise automatically calculated. The view factors for the fictitious layers were associated to two internal mass objects of equivalent surface area and having thermal properties of the mineral wool. These two internal mass objects are present also in all other simulations; just, in the other cases their view factors are calculated automatically by EnergyPlus, as for the surfaces. So, theoretically the only difference between this case and the case with exposed ceiling should be in the altered pattern of radiation heat exchange between the room surfaces and the internal mass objects representing the vertical baffles. In practice, one problem arises from the EnergyPlus inner algorithm for the view factors calculation. Indeed, when no user defined view factors are used – the normal way of running the software – EnergyPlus does not calculate direct view factors on the basis of the given geometry. Rather, an area weighted approximation method is used (EnergyPlus, 2008). This algorithm-peculiar difference, in this specific case, results in the paradox that the case with vertical baffles appears to behave slightly better than the case with exposed ceiling. Of course such a result is artificial, but it still allows withdrawing a conclusion. Modifying the pattern of radiation heat exchange between the room surfaces gives only minor, or eventually negligible, differences in the final cooling energy demand.

Using vertical baffles may have another effect, other than altering the radiation heat exchange pattern. As mentioned, the air motion may be hampered and consequently the ceiling convective heat exchange coefficient reduced. This case is also shown in Figure 4-4. When the convection heat transfer coefficient (h_C) is calculated automatically by the software, its value varies at each time step[2], and in average it results that $h_C \Delta 1$ W/m^{2}K. Figure 4-4 shows that when h_C is imposed always equal to one (by a user overwrite value) the resulting cooling energy demand is indeed nearly the same. This expedient allows estimating what happens when the vertical baffles would actually compromise the ceiling convective heat exchange, i.e. because they are mounted too close to the ceiling. Figure 4-4 shows that when it would be $h_C = 0.1$ W/m^{2}K – meaning 10 times lower than with exposed ceiling – the resulting cooling energy demand increases, like if the ceiling was covered by 40% with horizontally mounted panels. Therefore, it can be said that vertical baffles with a sound absorbing area equivalent to fully covering the ceiling area are expected to have – at worst – the same effect on cooling energy demand as horizontally mounted panels that cover only 40% of the ceiling area. So, at parity of sound absorbing area the vertical baffler offer a better energy performance than the horizontal panels. Vice versa, at parity of energy performance the vertical baffles can offer a sound attenuation double than horizontal panels.

The convective heat transfer coefficient at ceiling may be increase by means of ceiling diffuser specifically designed for the purposes, i.e. by increasing the air velocity at ceiling. Figure 4-5 shows what the effect will be on reducing cooling demand when it would be possible to increase the ceiling convection coefficient h_C from about 1 to 3 W/m^{2}K, when considering all acoustic solutions with a sound absorbing area equivalent to 100% covering of the ceiling. The result is that for all acoustic solutions except the horizontal panels the energy demand for cooling would become lower than the case with fully exposed ceiling and normal convection coefficient ($h_C \Delta 1$ W/m^{2}K). This means that for such solutions increasing the convection coefficient at ceiling has the potential to compensate for the decrease in performance caused by covering the thermal mass with acoustic elements.

[2] depending on the temperature difference between the surface and the air or on the number of air changes per hour, depending on which algorithm is chosen (EnergyPlus, 2008)

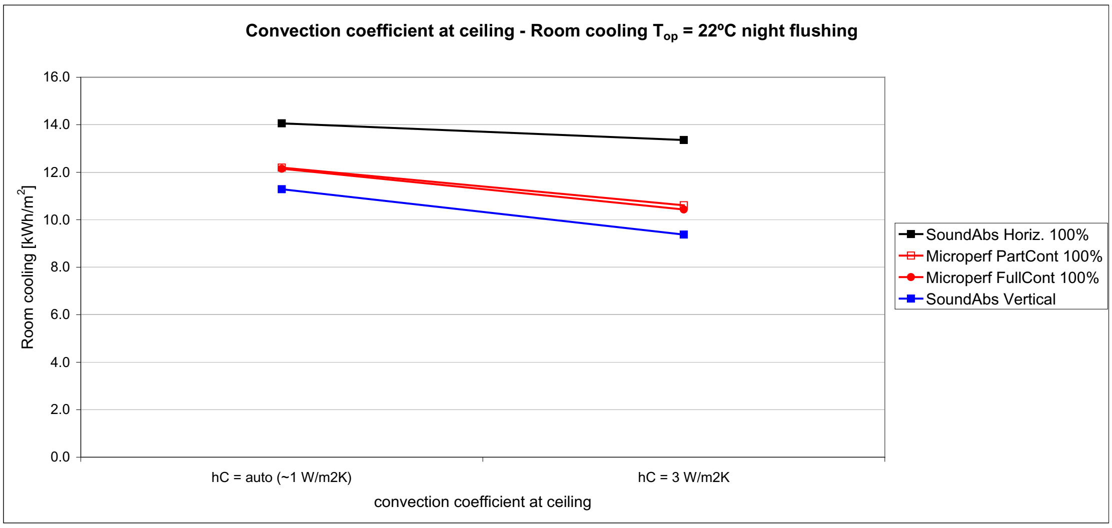

Figure 4-5: Effect of increased convection coefficient at ceiling on the cooling energy demand, with set-point 22°C and night flushing

Finally, it is worth looking at the total energy demand associated with cooling: room cooling and fans operation at day and night. The non-conditioned ventilation at day time is also considered here because it does contribute to keep the temperature lower, and for comparison with the cases with the ventilation cooling system. However, energy for fan operation at day is always the same because day ventilation is constant throughout all the cases. The following graphs from Figure 4-6 to Figure 4-8 show the energy demand for different acoustic solutions and temperature set-point of 26, 24 and 22°C, respectively. Because of their close similarity in performance, of the two micro perforated resonant absorbers only the one with full metallic contact to the concrete is shown. For the vertical baffles only the case with reduced convection coefficient is shown ($h_C = 0.1$ W/m^{2}K).

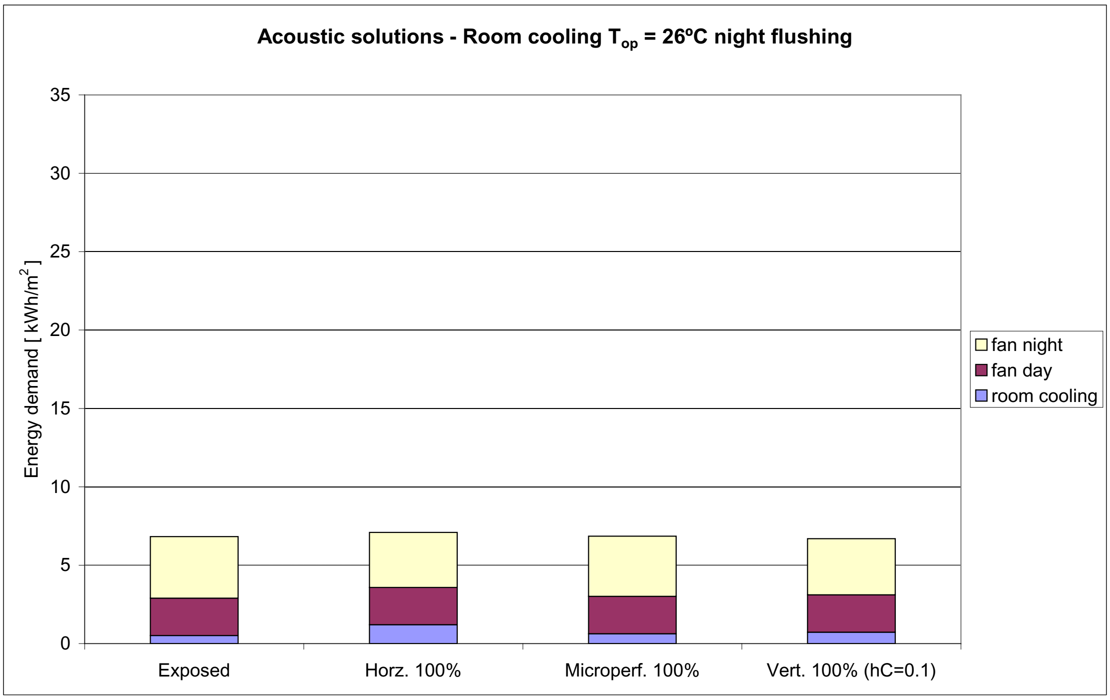

Figure 4-6: Energy demand with room cooling system for different acoustic solutions, with set-point 26°C

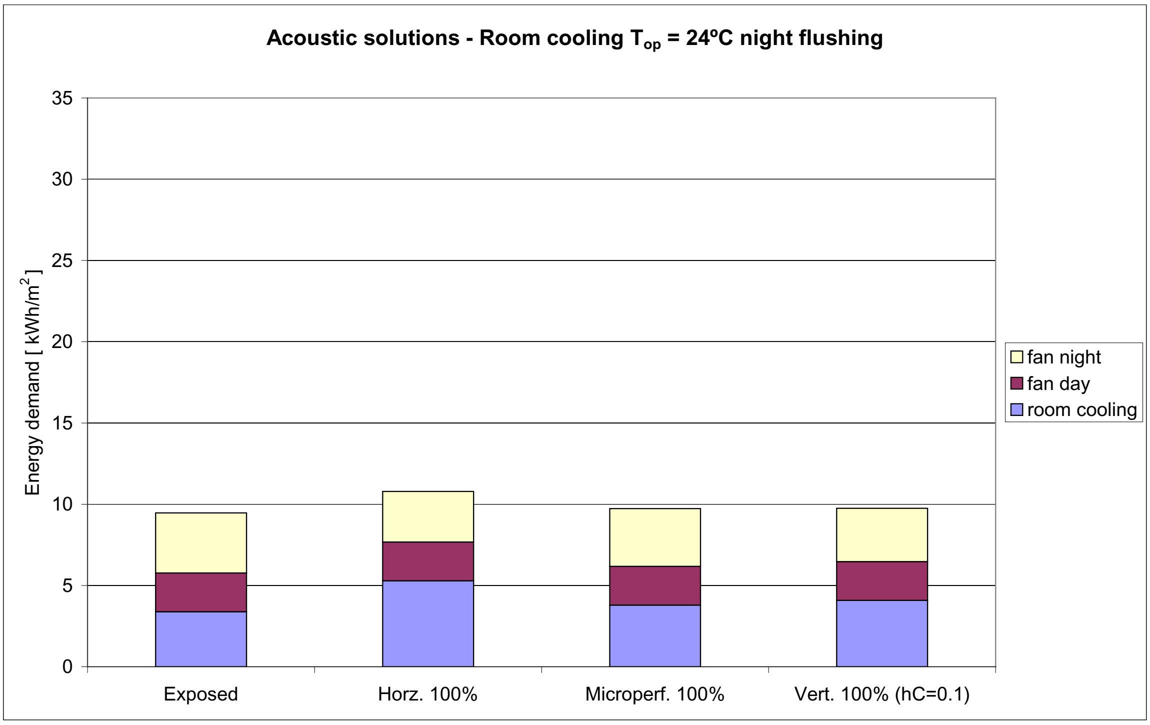

Figure 4-7: Energy demand with room cooling system for different acoustic solutions, with set-point 24°C

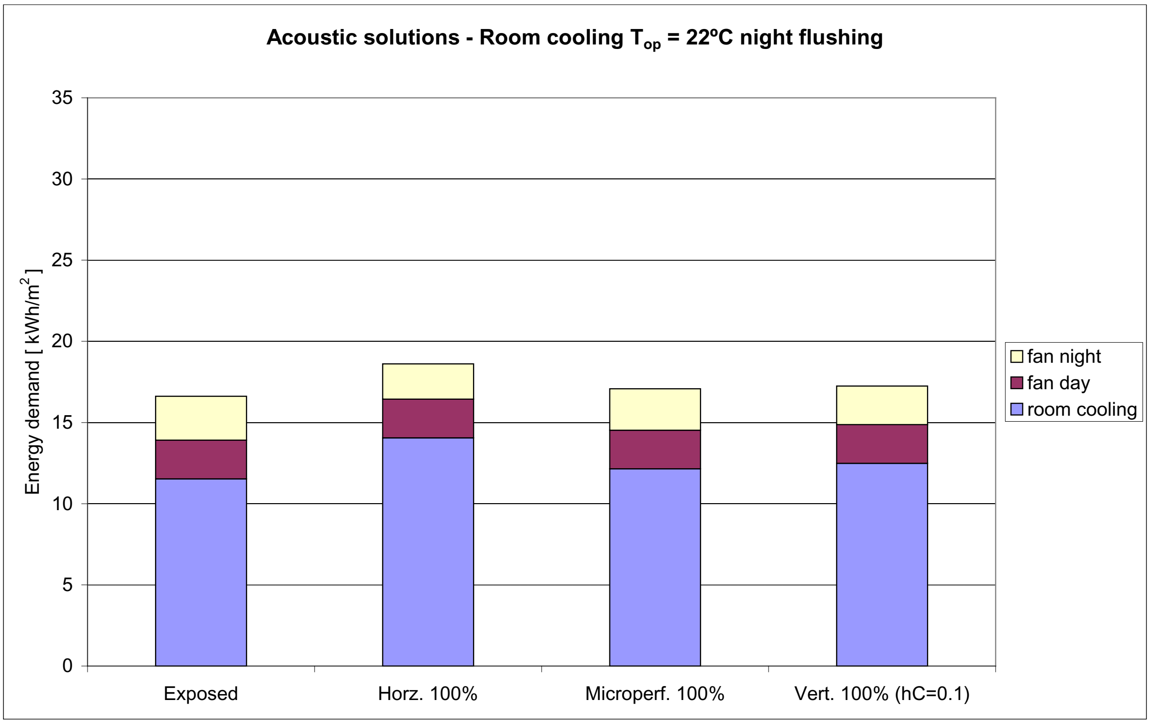

Figure 4-8: Energy demand with room cooling system for different acoustic solutions, with set-point 22°C

For all temperature set-points the results show no significant difference in energy demand for the different acoustic solutions, the horizontal panels always performing somehow the worst. With a set-point of 26°C all solutions demand approx. 7 kWh/m^2. Decreasing the temperature set-point from 26 to 24°C causes an increased energy demand of about 3 kWh/m^2, from approx. 7 to 10 kWh/m^2 for all cases – 11 kWh/m^2 for the horizontal panels.

A further decrease from 24 to 22°C comport twice as much increase in energy demand, about 7 kWh/m^2, from approx. 10 to 17 kWh/m^2 for all cases – 19 kWh/m^2 for the horizontal panels. It is worth noticing that fan consumption at night is higher than at daytime; the reason for it being the higher flow rate at night as already discussed previously.

4.2 Analysis with ventilation cooling

4.2.1 Night flushing

Using a mechanical ventilation system there is a certain cooling effect because the outside air is preconditioned and supplied at a temperature of 17-18°C. Figure 4-9 shows the resulting operative temperature when a Constant Air Volume (CAV) system is used to supply the minimum air flow for occupant needs, with and without night flushing. With respect to the temperature displayed in Figure 4-1 the situation is significantly improved, with lower average temperature and swing, but still above comfort condition. Introducing night flushing is beneficial in reducing both the operative temperature – now below 26°C in average, see Table 4-2 – and the number of hours it exceeds 26°C, from above 1000 down to 6, finally respecting the limit of 50 hours. This is possible because the minimum airflow value imposed by the NS3031 is relatively high (7 $m^3/h/m^2$) compared to the minimum requirement calculated according to the TEK-VEIL (2007), as discussed in §3.3. Already using airflow of 6 $m^3/h/m^2$ would make the CAV system unable to meet the 50 hours requirement.

It is necessary to introduce a Variable Air Volume (VAV) ventilation system and use night flushing to bring the comfort conditions safely within the desired range, as shown in Figure 4-10 and in Table 4-2. Without night flushing the ventilation system runs nearly always at the minimum flow rate at daytime, because that is enough to keep the room air temperature at 26°C. Nevertheless, because the thermal mass is never cooled down, the surface temperatures – and so the mean radiant temperature – become higher than the air temperature, and therefore the operative temperature exceeds 26°C. One solution could be to reduce the air temperature set-point until the operative temperature falls within the limits. However, this would increase the energy used by the cooling battery while making no good use of the thermal mass properties.

Alternatively, introducing night flushing brings the average operative temperature at 24°C, with a maximum of just 25.8°C and a swing that during occupation hours is in the order of 2°C. The number of occupation hours with operative temperature above 26°C is reduce drastically from about 900 down to zero. This happens because night flushing cools down the thermal mass, therefore sinking the surface temperatures and resulting in a mean radiant temperature that is always lower than the air temperature (opposite case then without night flushing). Then, controlling the air temperature becomes automatically a guarantee that also the operative temperature is under control.

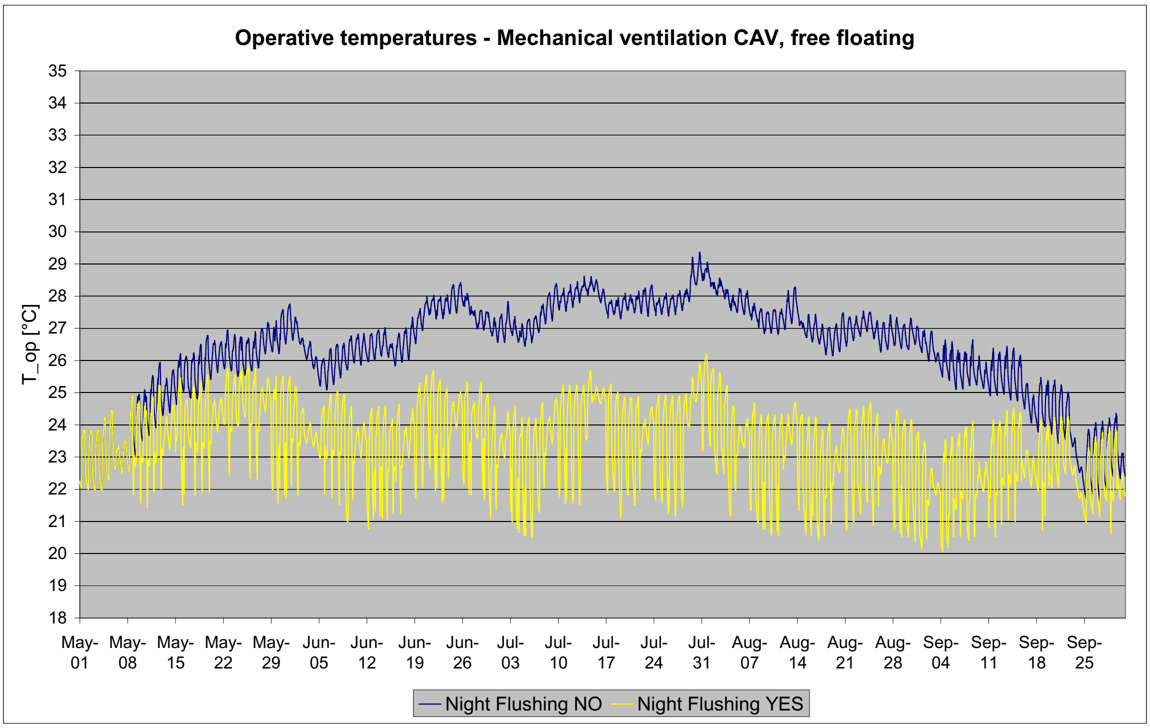

Figure 4-9: Overview of seasonal operative temperature profile for the case with mechanical CAV ventilation and free floating temperature, with and without night flushing

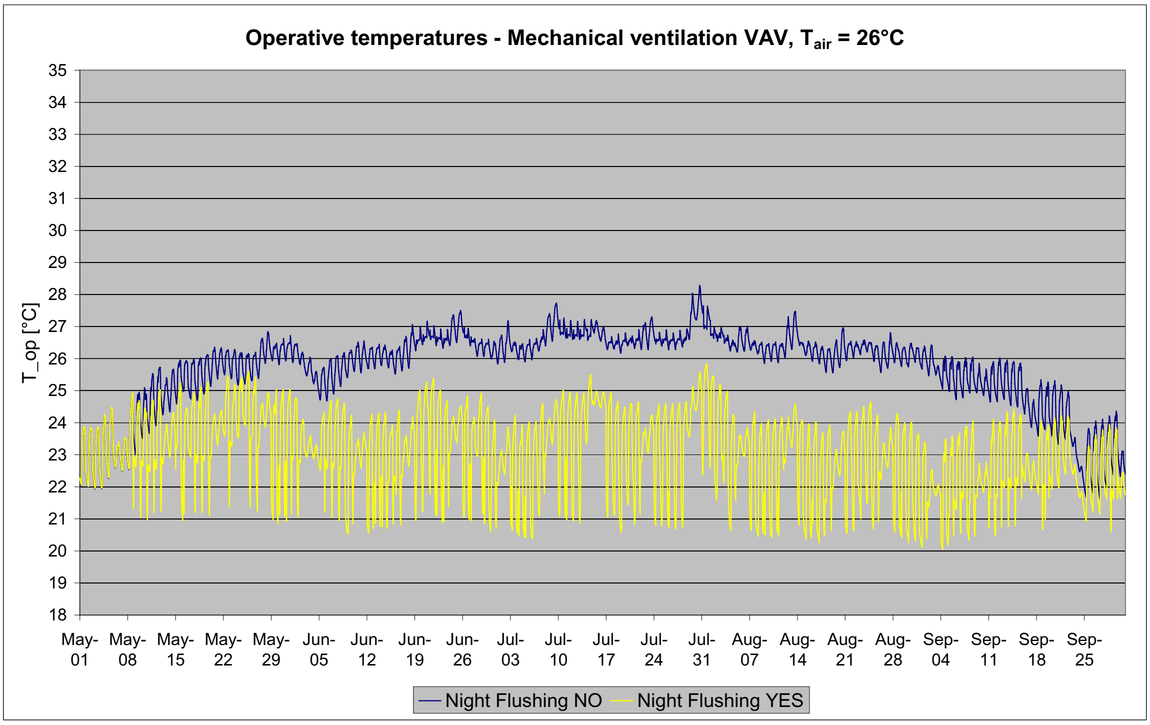

Figure 4-10: Overview of seasonal operative temperature profile for the case with mechanical VAV ventilation and temperature set-point of 26°C, with and without night flushing

Table 4-2: Summary of thermal comfort results for the cases with mechanical ventilation, with and without night flushing

Ventilation	Night Flushing	Operative Temperature [°C]		Temperature Swing [°C]		Occupation hours > 26°C
		T_{op}_avg	T_{op}_max	ΔT_{op}_avg	ΔT_{op}_max	
Mechanical CAV	No	26.6	28.9	26.6	28.9	993
Mechanical CAV	Yes	24.3	26.2	24.3	26.2	6
Mechanical VAV, T_{air} = 26°C	No	25.9	27.0	0.4	1.6	866
Mechanical VAV, T_{air} = 26°C	Yes	24.0	25.8	1.5	2.2	0

Figure 4-11 show that when looking at the overall energy demand – cooling battery in the ventilation system and the operation of the fans at day and night – introducing night flushing has an overall marginal effect. The difference between the cases with and without night flushing is < 1 kWh/m^2, from 10.2 to 9.4 kWh/m^2; while night flushing allows achieving the comfort requirements.

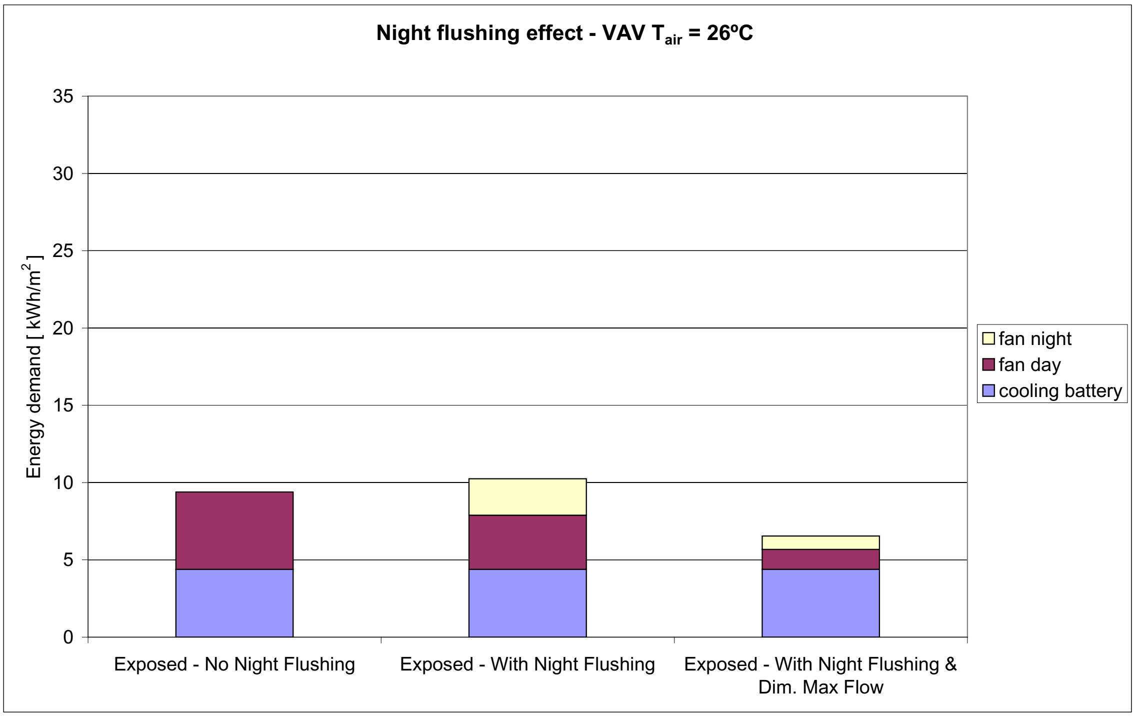

Figure 4-11: Effect of night flushing on the energy demand, whit exposed ceiling and ventilation cooling system with set-point of 26°C

Reminding that the fan power consumption is dependent on the nominal flow, see Eq. (3-1), it is possible to reduce the energy used by the fans dimensioning the ventilation system for the maximum flow $Q_{nom} = Q_{max}$ = 12 m^3/h/m^2. This, of course, implies the use of larger ducts that is not always possible because of the extra space they would require. However, would it be possible to dimension the ventilation system for the maximum flow, instead of the 70% of it as it is usual practice, the energy demand would drop down to approx. 6.5 kWh/m^2.

4.2.2 Ceiling acoustic solutions

The different types of acoustic solutions introduced in 3.4 are evaluated for their effect on thermal comfort and energy demand. Because of their close similarity in performance, of the two micro perforated resonant absorbers only the one with full metallic contact to the concrete is considered; and for the vertical baffles only the case with reduced convection coefficient is considered (h_C = 0.1 W/m^{2}K). Table 4–3 reports the thermal comfort results for the different acoustic solutions, and for different daytime temperature set-points, when night flushing is used. Figures from Figure 4–12 through Figure 4–4 show the energy demand for different acoustic solutions and temperature set-point of 26, 24 and 22°C respectively.

Table 4-3: Summary of thermal comfort results for the cases with mechanical VAV ventilation and night flushing for different acoustic solutions

Ceiling Acoustic Solution	**Room Air Temperature Set-point [°C]**	**Operative Temperature [°C]**		**Temperature Swing [°C]**		**Occupation Hours > 26°C**
		T_{op}_avg	T_{op}_max	ΔT_avg	ΔT_max	
Exposed	26	24.0	25.8	1.5	2.2	0
Horizontal 100%	26	24.4	26.0	1.9	3.0	2
Micro perforated 100%	26	24.2	25.9	1.3	2.0	0
Vertical 100% (h_C= 0.1)	26	24.1	25.8	1.6	2.4	0
Exposed	24	23.5	24.4	1.0	1.8	0
Horizontal 100%	24	23.7	24.5	1.2	2.3	0
Micro perforated 100%	24	23.6	24.5	0.8	1.5	0
Vertical 100% (h_C= 0.1)	24	23.5	24.4	1.1	1.8	0
Exposed	22	22.5	24.0	0.8	1.3	0
Horizontal 100%	22	22.8	24.2	1.0	1.9	0
Micro perforated 100%	22	22.7	24.1	0.7	1.2	0
Vertical 100% (h_C= 0.1)	22	22.5	23.9	0.8	1.4	0

Concerning the average and maximum operative temperature and the swing, there are no significant differences in the performance of the various acoustic solutions, irrespective of set-point temperature. With the temperature set-point at 26°C all acoustic solutions are within the limit of 50 hours. No relevant differences are seen with set-point at 24°C; while it is interesting to notice that when the set-point is on 22°C the ventilation system is not able to satisfy it. Hence, the ventilation system at daytime works always at maximum flow rate, trying to cope with the set-point of 22°C, but it only achieves an average operative temperature of about 22.5-23.0°C.

Concerning energy demand, once again, in all groups of temperature set-point there are no significant differences in the performance of the various acoustic solutions. With set-point at 26°C the overall energy demand is approx. 10 kWh/m^2 for all solutions. Decreasing the temperature set-point from 26 to 24°C causes an increased energy demand of about 5 kWh/m^2, from approx. 10 to 15 kWh/m^2 for all cases – 16 kWh/m^2 for the horizontal panels. A further decrease from 24 to 22°C comport an increase in energy demand of about 15 kWh/m^2, doubling the total to approx. 30 kWh/m^2 for all cases. It is worth noticing that decreasing the temperature set-point, the ventilation system will work at daytime with higher and higher flow rates in order to provide more cooling effect. Hence, the fan energy increases significantly because of the cubic relation between flow rate and power, see Eq. (3-1).

Concerning night flushing, it shall be noticed that as the daytime set-point is reduced the effect of activating the thermal mass is less and less. Consequently there is less and less need to cool it at night, and so the energy for night flushing get reduced.

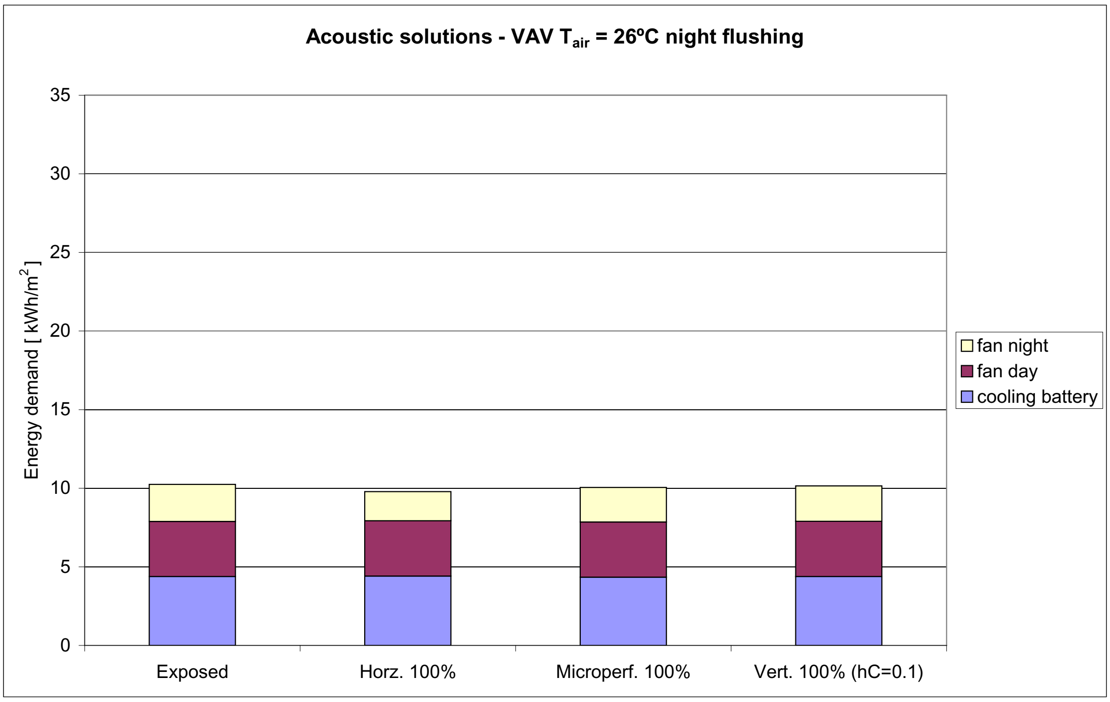

Figure 4-12: Energy demand with ventilation cooling system for different acoustic solutions, with set-point 26°C

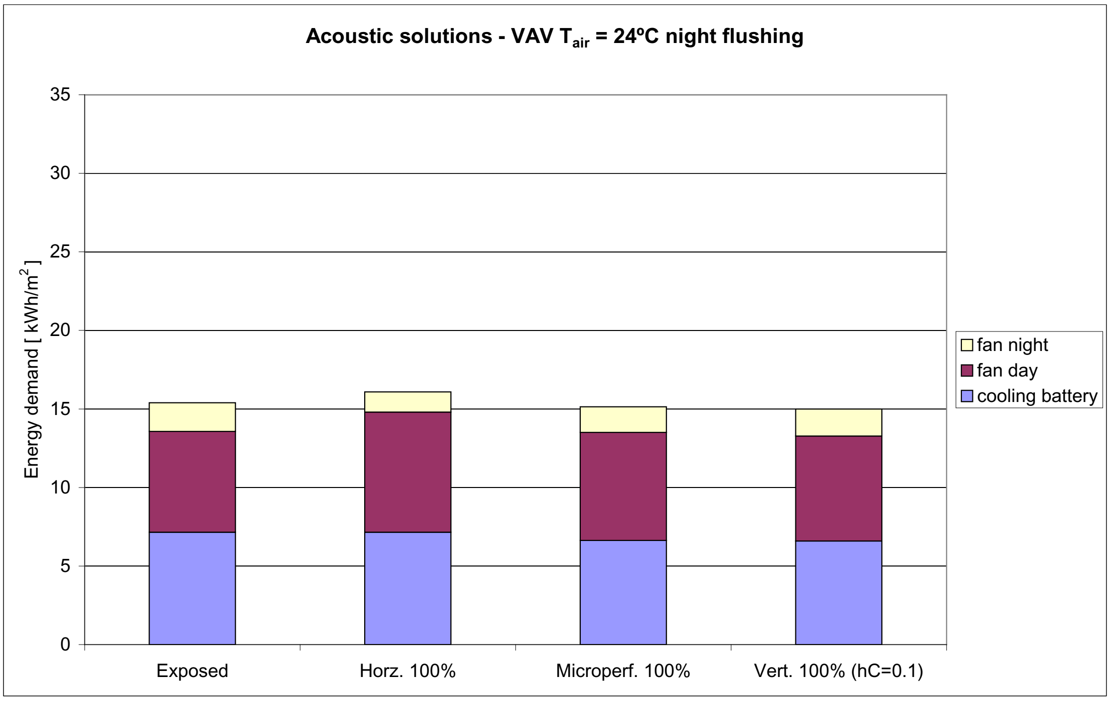

Figure 4-13: Energy demand with ventilation cooling system for different acoustic solutions, with set-point 24°C

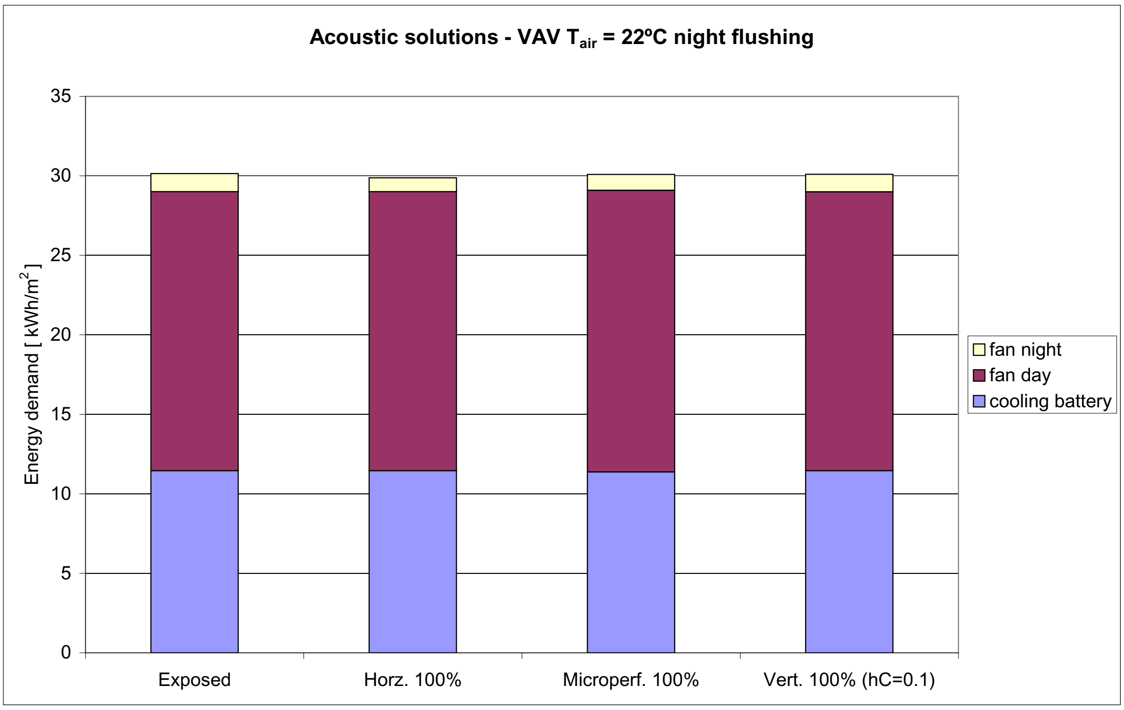

Figure 4-14: Energy demand with ventilation cooling system for different acoustic solutions, with set-point 22°C

In comparison with the ideal room cooling system, the overall energy demand with set-point at 26°C is higher. With the ventilation cooling the cooling battery will consume more than the room cooling system for two reasons. In the case of an idealised room cooling system it was assumed that outdoor air is used for ventilation. This means that no temperature rise by the fans is considered, and also implies that when the outdoor air is cool enough it will provide some free cooling effect. Hence, the room cooling system only has to add the minimum indispensable cooling effect to reach the balance at 26°C for the operative temperature. With the ventilation system the control is made on the room air temperature, which is always higher then the operative temperature when using night flushing that cools down the surfaces (operative temperature is the average between air and mean radiant temperatures). In addition, the fan's temperature rise has to be compensated for. Furthermore, all the air supplied to the room is always conditioned, even when this would not be necessary. Indeed, the resulting average operative temperature is 24°C, see Table 4-3, meaning that the ventilation system is also "paying for" some better comfort than the equivalent cases with room cooling. On the other hand, fan energy at night is reduced partly because the higher cooling effect at daytime reduces the need for night cooling and partly because of the extra restrictions applied, see §3.3.1 and §3.3.2.

With temperature set-point at 24°C the ventilation cooling system cases consume more than the corresponding cases with room cooling system for all of the above mentioned, and especially because of the strong increase in the fan consumption at daytime. With temperature set-point at 22°C the ventilation system works at maximum flow nearly all the time and still is not able to provide enough cooling. This implies that daytime fan consumption is at its maximum, while energy for the cooling battery is lower than the energy for room cooling in the corresponding case because the target of 22°C is not achieved, and the resulting average operative temperature is about 23 °C.

5 Conclusions and discussion

The analysis with the idealised room cooling system allowed estimations of the effect on energy demand of night flushing and different acoustic solutions. With the given dimensioning of the ventilation system, night flushing proved able to reduce overall energy demand, when accounting for cooling energy and fans operation at day and night. Covering the ceiling with sound absorbers, partially or fully, does reduce the effect of night flushing because it somewhat decouples the ceiling thermal mass form the room air. However, the overall effect is small compared to the effect of using or not using the night flushing. For all temperature set-points considered, the results show no significant difference in energy demand for the different acoustic solutions; the horizontal panels performing worst. Micro perforated resonant absorbers covering the entire ceiling induce approximately the same energy demand as the conventional insulating panels covering 30% of the ceiling. Vertical baffles with a sound absorbing area equivalent to fully covering the ceiling area may have – at worst – the same effect on cooling energy demand as horizontally mounted panels that cover only 40% of the ceiling area. Increasing the convection coefficient at ceiling has the potential to compensate for the decrease in performance caused by covering the thermal mass with acoustic elements.

The analysis with the ventilation cooling system allowed estimations of both energy demand and effects on thermal comfort conditions. Using a VAV ventilation system with night flushing proved sufficient to fulfil the legal requirements on operative temperature, without use of direct room cooling. Night flushing has an overall marginal effect on the energy demand, when accounting for cooling battery energy and fans operation at day and night. It is possible to further reduce the energy used by the fans dimensioning the ventilation system for the maximum flow; this, however, implies the use of larger ducts that is not always possible because of the extra space they would require. Concerning both the energy demand and the thermal comfort conditions, with all temperature set-points there is no significant difference in the performance of the various acoustic solutions. However, as the daytime set-point is reduced the effect of night flushing is reduced.

5.1 Critical conclusions

By looking at the results uncritically one may come to the conclusion that the effect of thermal mass is negligible because either leaving the ceiling exposed or covering it, no matter with what type of sound absorbers, does not make any significant difference on either the comfort or the energy demand. But this is, indeed, an uncritical conclusion. Looking closer at the cause-effect relations one realizes that the situation is another one.

First of all, it shall not be forgotten that the floor is covered just with linoleum finishing, so that even when the ceiling is completely covered the room still have half of its thermal mass fully coupled with the room air. Secondly, the results for different acoustic solutions refer to cases with the use of night flushing. Hence, the thermal mass is used and so its effect is implicitly included in the results. In facts, it was shown that the most effective measure is to introduce night flushing (a measure that would not achieve any result if there was no thermal mass to be cooled).

However, the main point is that the boundary conditions are not appropriate for studying the thermal mass effect. There was a quest to understand if and how the use of concrete, hence of thermal mass, could help satisfying the temperature requirements in new office buildings without using local room cooling (TEK 2007) and at the same time help reducing energy demand, as it is indicated by the literature on the subject (see state of the art from Haase and Andresen, 2007). The input values were chosen in compliance with the NS3031 (2007) because that is the reference norm for energy calculations and is commonly used by building industry practitioners to certify the energy performance of buildings. However, this does not mean that it can be used as a basis for answering questions of design and dimensioning nature. The NS3031 method, simply, is meant to be used for estimating annual energy demand and so is based on average values and approximated values that fall in the safe side, but are not suitable for more detailed analysis, i.e. of a VAV ventilation system. The minimum ventilation flow rate (7 $m^3/h/m^2$) is higher than the actual requirement – while the suggested one would be even higher (10 $m^3/h/m^2$, NS3031 Table B.1) – and this has a strong influence in the results. On the other hand, people occupancy and internal gains are average values and do not consider possible peaks. In addition, minimal contribution of solar gains was assumed in order to "isolate" the effect of the different acoustic solutions. In other words, with low solar gains, low internal gains and high airflow the result is that surely it is possible to satisfy the temperature requirements, no matter whether the ceiling is covered or not. However, this does not mean that thermal mass has no effect. The most appropriate conclusion is rather that the method chosen for calculation is suboptimal.

An adequate approach that aims at maximizing thermal comfort while minimizing energy demand would require a finer description of the ventilation system and its control strategy both at day and night, as well as of the internal load and solar shading profiles. Only in this way it is possible to appreciate the real effect of thermal mass.

As a consequence, a series of details about the ventilation system and its control strategy should be carefully analysed. For example, a VAV system is best operated with occupancy sensors or CO_2 sensors that can modulate the flow rate according to the actual needs in the ventilated space and not only according to the room temperature; in this way the energy consumption for the fans would be minimised. Night flushing control strategy can also be optimised, especially in combination with natural ventilation in order to reduce night fan consumption that can be the real bottle-neck (this is a task better suited for a real case study though, rather than for a generalised case). Again, as it was shown, the convective heat transfer coefficient at the ceiling can play a significant role, and different solutions to improve such coefficient could be studied. Further work is needed to find an optimal control strategy for the ventilation system.

It can be summarised that answering the question "if and how the use of concrete, hence of thermal mass, could help satisfying the temperature requirements in new office buildings without using local room cooling" is just not about changing an element with another. It is necessary to have a holistic approach that considers concrete, acoustic elements, operating conditions, ventilation system and control strategies all at the same time.

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