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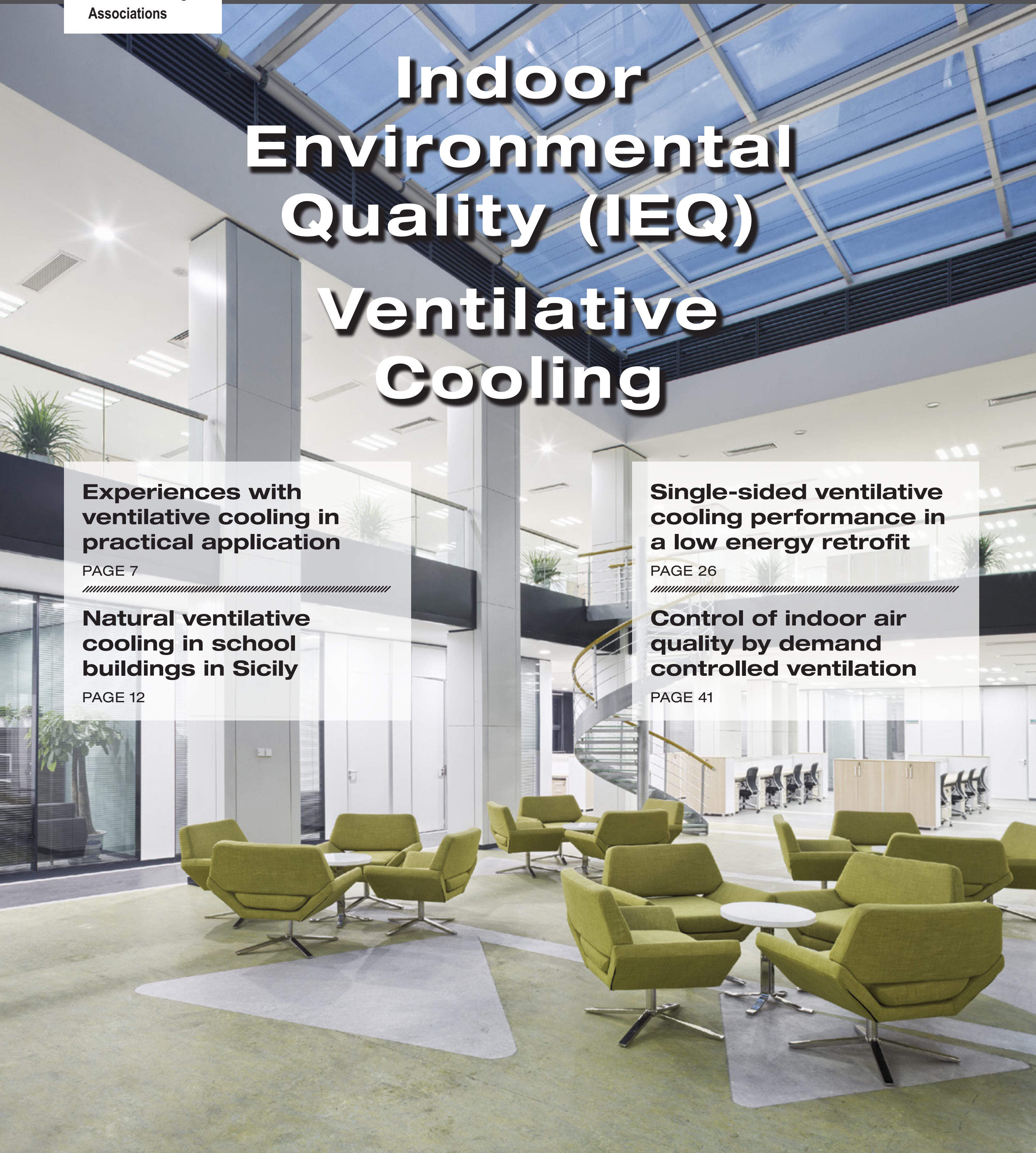

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Will the outcome of COP21 stimulate the EU member states to accelerate the building and HVAC sector towards a carbon-free build environment?

JAAP HOGELING
Editor-in-Chief

COP21 has crossed many thresholds needed for the planning and implementation of global climate change actions. One of them is that it gave a clear message that from the technology point of view the smooth transition to near-zero emissions in buildings is feasible. A key message of the COP21 side events was to feature available solutions enabling the achievement of the ambitious zero emission targets. Another novelty of COP21 was that for the first time ever the energy sector – including among others buildings energy efficiency and construction sector – was involved in the UN climate discourse (see article on page 72). Experts and stakeholders agreed that the only way to force the market to invest in decarbonisation of buildings are prescriptive codes and mandatory energy performance (and/or CO_2 emission) requirements that are to be enforced.

The EU Energy Performance Buildings Directive requires the EU-MS's to enforce this. By 2020 for new buildings nZEB requirements will lead to more energy efficiency. New nZEB buildings or complete new settlements where smart building design and smart grid solution will be integrated. It will be a challenge for our industry and the innovative building and system designers to bend the "nearly" in our nZEB definition to really "zero" towards positive energy buildings where even the plug-loads are covered. This to reach complete de-carbonisation of the build environment. Regulators are not expected to go that far; they are still too much focused on the short term "cost-optimality" approach which sadly hampers accelerating innovation. Our building and HVAC sector has to present the arguments to convince the long-term focussed investors and developers to take the lead.

HVAC, Energy and Building professionals have to be aware that these challenges require more training at all levels. Also a task for REHVA and the REHVA national member associations within Europe! Finally, we have to realise that the new building volume is just a very small fraction, in most EU counties less than 1%. Reaching a carbon-free build environment will require much more efforts towards deep-renovation. This to upgrade the existing building stock to the nZEB level. The resources needed for this could even exceed the efforts for new buildings. To quote Prof. Dr. Stefano P. Corgnati (REHVA President-elect) "Nevertheless, energy targets must be fixed with care. Energy consumption reduction from 250 kWh/m²y (typical of an existing building) to less than 30 kWh/m²y (towards nZEB), ensuring at the same time cost optimality, is a very ambitious goal in existing buildings. Moreover, it is not always feasible due to technological issues and real estate criticalities."

This January 2016 issue focuses on "ventilative" cooling as an important approach to limit and prevent mechanical cooling in buildings. This technology needs special attention not only for new buildings but moreover for the existing building stock. By renovating and improving the thermal performance of existing buildings ventilative cooling can be a good solution to prevent overheating.

A happy, healthy and successful year 2016 for all our REHVA Journal readers!

On behalf of REHVA board, staff and editorial board.

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Experiences with ventilative cooling in practical application

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Recently built Active Houses provide experience with ventilative cooling in practical application as a means to prevent overheating in energy efficient residential buildings. Detailed measurements of energy performance and indoor environment have been made in five single family houses, located in Austria, Denmark, England, France and Germany. The houses have generous daylight conditions, with a design target to reach a Daylight Factor of 5% in the main habitable rooms. This increases the risk of overheating, but the measurements show that the houses generally achieve thermal comfort category 1 of EN 15251 during summer, which means that overheating has not occurred. To succeed, natural ventilative cooling and dynamic solar shading was applied and controlled to avoid overheating (which is possible under European climate conditions, where humidity is not a main issue during summer). Some of the experiences from applying ventilative cooling can be generalised to other houses and are presented in the article.

Some barriers limit the use of ventilative cooling. Firstly, the current methods in standards and legislation that are used to determine the performance of ventilative cooling need to be further strengthened. Secondly, affordable, intuitive and simple control systems for residential hybrid ventilation and dynamic solar shading are needed to tap the full potential of ventilative cooling.

The Active House Specification is based on a holistic view on buildings including Comfort, Energy and Environment. It uses functional requirements to indoor air quality and thermal comfort.

Keywords: Active House, ventilative cooling, natural and hybrid ventilation, standards, controls systems.

Introduction

Overheating is an important issue for building designers. Even in Scandinavia, demonstration houses have frequently experienced problems with overheating, often due to insufficient solar shading and use of natural ventilation (Isaksson, 2006, Larsen, 2012, Rohdin 2013). Similar results were found in a review on the situation in UK, stating that in certain cases, dwellings that were recently built or refurbished to high efficiency standards have the potential to face a significant risk of summer overheating (AECOM, 2012). Porritt et al. (Porritt, 2011) found that living room temperatures could be maintained below the CIBSE overheating thresholds, as a result of a combination of intervention measures that include external wall insulation, external surface albedo reduction (e.g. solar reflective paint), shading (e.g. external shutters) and intelligent ventilation regimes. Orme et al. found that night ventilation is a particular important measure to prevent overheating (Orme et al., 2003), and also found that the risk of overheating will increase in the future due to climate change.

The Active House Specification (Eriksen et al., 2013) has requirements in three categories, and has a main ambition that the three categories should have an equally high focus. The three categories are:

- Comfort (incl. indoor environmental quality)
- Energy
- Environment

Four categories of maximum operative temperature are defined, setting requirements to air-conditioned and non-air-conditioned buildings, using the definitions of EN 15251. For non-air conditioned buildings, the adaptive approach is used:

1. $\Theta_{i,o} < 0.33 \cdot \Theta_{rm} + 20.8°C$, for Θ_{rm} of 12°C or more
2. $\Theta_{i,o} < 0.33 \cdot \Theta_{rm} + 21.8°C$, for Θ_{rm} of 12°C or more
3. $\Theta_{i,o} < 0.33 \cdot \Theta_{rm} + 22.8°C$, for Θ_{rm} of 12°C or more
4. $\Theta_{i,o} < 0.33 \cdot \Theta_{rm} + 23.8°C$, for Θ_{rm} of 12°C or more

Where Θ_{rm} expresses the average outdoor temperature (°C) weighted over time according to EN 15251 and $\Theta_{i,o}$ is the indoor operative temperature (°C).

Natural ventilation in combination with dynamic solar shading is a key instrument to avoid overheating with minimal use of energy, but there are no specific requirements in the Active House Specification to use the measures.

Daylight is important for humans, and the requirements are based on average daylight factors on a work plane in the main living room, which must be determined by a validated simulation tool. The criteria are:

DF > 5% on average
DF > 3% on average
DF > 2% on average
DF > 1% on average

Criteria for energy and environment are found in the Specification (Eriksen et al., 2013), which can be downloaded at no cost from the website of the Active House Alliance.

Experiences from completed active houses

Ventilation System Configurations

Many of the realized Active House have been built with demand-controlled, hybrid ventilation systems for optimal IAQ and energy performance.

An example is from the project Sunlighthouse in Austria. Natural ventilation is used during warm periods and mechanical ventilation with heat recovery is used during cold periods. The switch between mechanical and natural ventilation is controlled based on the outdoor temperature. The set point is 12.5°C with a 0.5°C hysteresis. Below the set point the ventilation is in mechanical mode, above the set point 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 IAQ, and a set point of 850 ppm CO_2 is used.

LichtAktiv Haus in Germany is an example of a house where natural ventilation is used as the only ventilation system.

Measured IAQ

Temperatures and CO_2-concentrations have been measured continuously for 1–2 years in several inhabited Active Houses, e.g. in LichtAktiv Haus (LAH), Germany. The measurements were supplemented with systematic qualitative feed-back from the inhabitants. LAH is designed with a demand controlled IAQ, with the aim to achieve category 1 (500 ppm above outdoor levels) or 2 (750 ppm above outdoor levels) (Feifer et al., 2013). The measured CO_2-concentration in the living/dining room is presented in **Figure 1**.

It is seen in **Figure 1** that category 1 or 2 is achieved for 60% to 70% of the time during winter, and approx. 100% of the time during summer. The CO_2-concentration is lowest during the summer period as natural ventilation is also used to prevent overheating in this part of the year. Good summertime IAQ is thus a positive side-effect of applying ventilative cooling to prevent overheating. CO_2 concentration above category 2 during winter is caused by user override of automated controls. These results are similar to those seen in Active Houses with mechanical/hybrid ventilation.

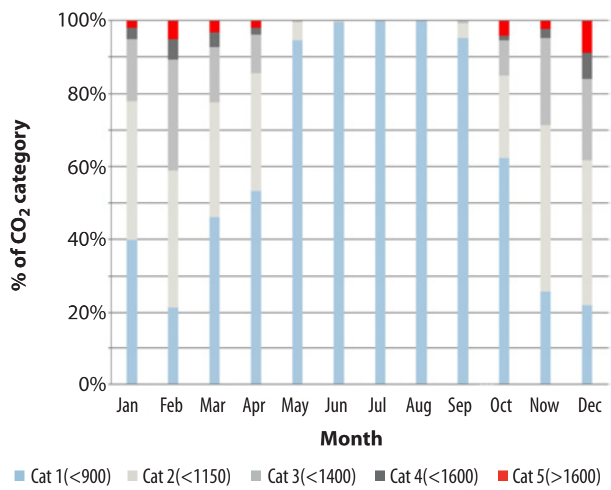

Figure 1. Measured CO_2 concentration in the kitchen/living room of LichtAktiv Haus, Germany. The data is categorized according to the Active House Specification. The outdoor CO_2 level is assumed to be 400 ppm.

It is the general experience that both natural, mechanical and hybrid ventilation systems are able to deliver the right ventilation rates and achieve the right IAQ. The key issue is that the systems must be designed, installed and maintained correctly, and most importantly, the controls must be transparent and intuitive for the occupants of the buildings.

Measured Thermal Comfort

Foldbjerg (Foldbjerg et al., 2013) reported on the thermal comfort in LAH and two other Active Houses. A typical characteristic of the realized Active Houses is that they have very generous daylight conditions. It is seen on **Figure 2** that the living-dining room in LAH achieve category 1 in most months, with the exception of a limited number of hours during the three summer months. Annually, the room achieves category 1. There are very few hours with temperatures below category 1. This means that there is no issues with overheating or low temperatures (undercooling).

Prevention of overheating is a key issue, as low energy buildings can easily overheat, as reported by Larsen (Larsen, 2012) and others. It is the general experience from the realized Active Houses those good thermal conditions with only insignificant periods with high or low temperatures can be achieved. The important elements to consider are natural ventilation and dynamic solar shading, as combined in ventilative cooling (venticool, 2014).

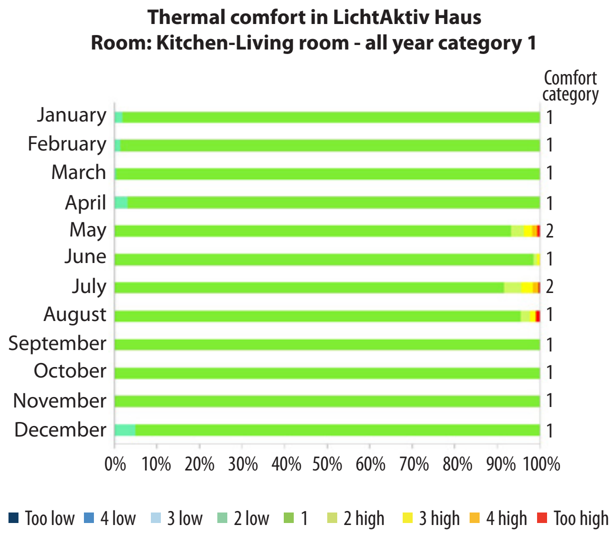

Figure 2. Measured indoor temperature in the kitchen/living room of LichtAktiv Haus, Germany. The data is categorized according to the Active House Specification. The number on the right side of the figure is the Active house category achieved for each month (max 5% of the time can exceed the category).

Ventilative Cooling in Standards

Peuportier (Peuportier et al, 2013) measured the air change rates achieved with natural ventilation as the means of ventilative cooling in the Active House called Maison Air et lumière near Paris, France. Air change rates in the range of 10 to 22 ACH were achieved. These results were confirmed by simulations in CONTAM. However, later calculations with the methods presented in EN 15242 show much lower results despite similar geometry and boundary conditions. This is to some extent explained by the fact that EN 15242 only includes single-sided ventilation. BS 5925:1991 presents a method that allows for a two-sided window configuration, still with very conservative results. In the on-going revision of EN 15242 it is being discussed if a more accurate and generally applicable method can be included. The work in IEA Annex 62 will further support this goal.

Ventilative Cooling in legislation

Ventilative cooling is only to a limited extend addressed in legislation through building codes and compliance tools. Recent years, several national building codes have included ventilative cooling with simplified calculation of ventilation flow rates that are not directly addressing the performance of the actual ventilation and building design. To correctly account for the effect of ventilative cooling, more accurate methods are needed. A method was recently implemented in the Danish Be10 compliance tool, and there is currently work on-going in France to improve the methodology for the calculation of ventilative cooling and to integrate summer comfort better in the French national code and compliance tool.

Also in France and Denmark, requirements to thermal comfort are likely to be more elaborated in coming revisions of the national building codes. This is a necessary step to prevent overheating, but requires that the underlying methodology adequately accounts for the actual performance.

Experience with Control Systems in Active Houses

An effective control of dynamic shadings and natural ventilation is important for achieving good summer comfort. Such control may be based on manual operations, knowledge and good habits but in the Active Houses described here, the full step towards fully automated control was taken. The automatic control was in general appreciated by the users, though the users needed some time for adjusting to the system. The user feed-back showed clearly that they appreciated the automatic control if override was possible. It is essential to offer intuitively manually operable devices such as windows, doors, and awning blinds allowing the users to override the automatic system.

There are few control systems currently available that deliver control of both mechanical and natural ventilation (as a hybrid solution), and which controls both ventilation, window openings and dynamic solar shading in a combined effort to maintain both good IAQ and good thermal comfort. Such systems should be cost-effective and are needed for the residential market to tap the full potential of dynamic building elements and to reach the ambitious nZEB targets of EU-28.

Conclusions

Good IAQ can be achieved with both natural, mechanical and hybrid ventilation systems. The important lesson is that they must be planned, installed and maintained right. This has been achieved in the investigated houses. By correct planning in the design process good IAQ can be reached with a minimum use of energy. Particular good IAQ during the summer period has been observed as a positive side-effect of applying ventilative cooling.

Whereas the above themes have been relatively unproblematic, some issues, mentioned below, have a greater need for increased focus regarding quality and compliance.

The realized houses are characterised by generous daylight conditions, which could potentially lead to overheating. This has not been the case. The houses show that good thermal comfort can be achieved in all seasons, regardless whether natural, hybrid or mechanical ventilation is used. But a strong relation between efficient natural ventilation in the summer (ventilative cooling) as well as dynamic solar shading has been a key element in achieving this, supported by windows being located towards more than one orientations in each room and not mainly towards the south as sometimes seen in low energy houses.

There is currently only limited support in standards and legislation to give a true and fair account of the performance of ventilative cooling and dynamic solar shading, and this needs to be improved.

There remains a need to identify and to discuss how ventilative cooling can become a standard solution in legislation and standards throughout Europe especially regarding renovation but also regarding Nearly Zero Energy Buildings.

Transparent and intuitive control systems scaled for residential buildings with regards to system architecture and price are needed. Such a control system should be able to control ventilation and dynamic solar shading to maintain both good IAQ as well as good thermal comfort. ■

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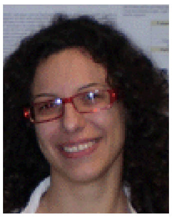
Natural ventilative cooling in school buildings in Sicily



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The paper shows analysis of the impact on thermal comfort of natural ventilation in a non-residential Mediterranean case-study.

Results are based on the EN 15251 approach on adaptive comfort. Different scenarios are proposed, granting an improvement of up to 10% in the number of summer comfort hours.

Keywords: Ventilative cooling, natural ventilation, TRNSYS, building simulation, thermal comfort.

In the context of a renewed interest of the European Commission (the EPBD-recast 2012/27/EU directive) regarding the energy efficiency of buildings [1], energy retrofit actions in the public buildings sector are gaining interest [2-4]. One of the sectors that may benefit more from energy efficiency actions is the education one: in Sicily (Italy), more than 70% of schools have been built before the 1980s and therefore they often have energy performances that do not comply with the most up to date energy regulations. Moreover, due to high internal loads and a hot climate from May to October, cooling may be needed during large parts of the year, but air conditioning systems are rarely available: a solution may lie in ventilative cooling techniques that aim to reduce indoor temperature by simply using outdoor air.

The case study

The study shows an analysis aimed to increase the indoor summer comfort levels by means of natural ventilative cooling techniques in a school building in Sicily. The analysis is based on the thermal simulation of a real building and the development of scenarios where different ventilative cooling techniques are used; comfort levels are quantified by means of the EN 15251 adaptive comfort models. The zones investigated in the analysis are only classrooms and some administrative offices. The school building (**Figure 1** and **Figure 2**) is located in southern Sicily, 20 km away from Agrigento, facing the Sicilian channel.

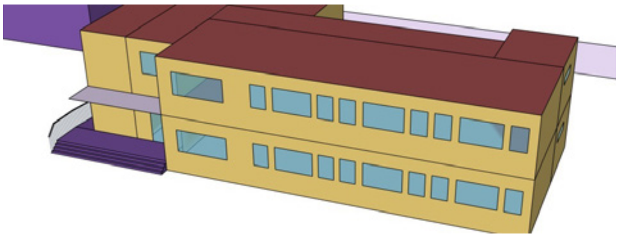

Figure 1. East Façade of the building, Google SketchUp.

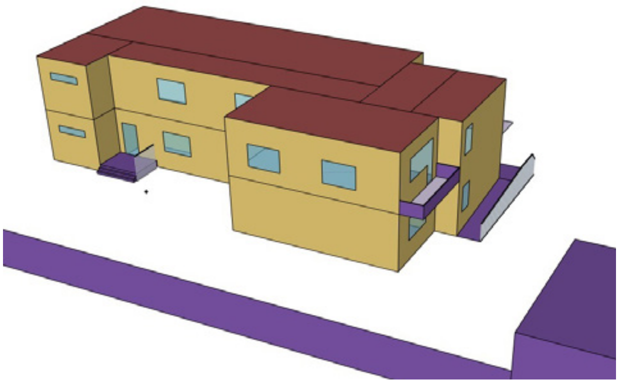

Figure 2. West Façade of the building, Google SketchUp.

The building has two levels, it has an overall 675 m^2 surface and, in the classrooms, the window to wall ratio is around 25%. The building develops mostly on the East-West direction; the overall U value of opaque external surfaces is around 1.2 W/(m^2 K) and it is around 3 W/(m^2 K) for the glazed surfaces (single) used.

Cooling systems are not available; no specific use of natural ventilation is reported during the day or the night.

The thermal building simulation is modelled in TRNSYS 17 environment, natural ventilation modelling involves the use of the empirical equations specified by ASHRAE in [5], where the ventilation air flow rate is function of wind speed, thermal stack effect, opening area of windows and their opening factors. Windows are considered close, if wind speed is higher than 3 m/s.

The thermal zoning performed in the analysis is represented in **Figure 3**. Zones Classrooms 1 and 2 have an overall area of around 165 m^2 while the zones Classrooms 3 and Administration office are nearly 66 m^2 large. The other zones including corridors, halls, technical locals, have been modeled but they are not analyzed as comfort zones.

The building use takes place mostly from 8:30 to 14:30 (all zones in **Figure 3**) in the morning but it is often occupied also during the afternoon from 16:30 due to some evening classes (only the classrooms 2 zone). This is particularly relevant since the whole comfort analysis is performed on the occupied hours only and will have quantitative impacts on the results as well.

As specified in the regulation, running mean temperatures (Θ_{rm}) are calculated as function of recurring average temperature values ($\Theta_{ed\text{-}i}$) calculated during previous week (*) while the comfort temperature is calculated through Eq.1.

$$\Theta_c = 0{,}33\,\Theta_{rm} + 18{,}8\ °C \qquad (1)$$

The comfort temperature ranges adopted for the study are respectively reported in Eq.2 and 3.

$$\Theta_c - 2 < \Theta_{indoor} < \Theta_c + 2\ °C \qquad (2)$$

$$\Theta_c - 3 < \Theta_{indoor} < \Theta_c + 3\ °C \qquad (3)$$

* $\Theta_{rm} = (1 - \alpha)\cdot\{\Theta_{ed\,-1} + \alpha\,\Theta_{ed\,-2} + \alpha^2\,\Theta_{ed\,-3} + \ldots\}$

Where: Θ_{rm} = External Running mean temperature for the considered day (°C).

$\Theta_{rm\text{-}1}$ = running mean external air temperature for previous day

α = constant between 0 and 1 (recommended value is 0.8)

$\Theta_{ed\text{-}i}$ = daily mean external air temperature for the *i*-the previous day

prEN 16798-1 (replacing the EN15215) states the following:

The following approximate equation shall be used where records of daily running mean external temperature are not available:

$\Theta_{rm} = (\Theta_{ed\,-1} + 0{,}8\,\Theta_{ed\,-2} + 0{,}6\,\Theta_{ed\,-3} + 0{,}5\,\Theta_{ed\,-4} + 0{,}4\,\Theta_{ed\,-5} + 0{,}3\,\Theta_{ed\,-6} + 0{,}2\,\Theta_{ed\,-7})\ /\ 3{,}8$

The case study model was run in non-steady state by using Meteonorm weather data, all the temperature data for each of the occupied thermal zones analyzed and compared to results of Eq.1 to verify whether each simulated hour for every thermal zone would fall inside the thermal comfort ranges.

The study has determined the percentage of comfort hours for the existing building including no natural ventilation strategies and in the case of daily ventilation, night ventilation and continuous natural ventilation during the whole 24 hours of the day in a period including May, June, early July, September, early October.

Results and discussion

The comfort hours percentages for each of the analyzed thermal zones are shown in **Figure 4**, calculated according to eq.2 and 3 temperature ranges for the existing building with no natural ventilation strategies applied.

Classrooms 3 and the administration locals show results very close to each other. Classrooms 1 and 2 instead, although being geometrically very similar, show results significantly different: this can be explained with the already described differences in occupation levels. Adding more occupation hours in the afternoon to a thermal zone (Class 2) that has undergone high internal gains and a high solar radiation during the morning, will surely cause an increase in thermal discomfort.

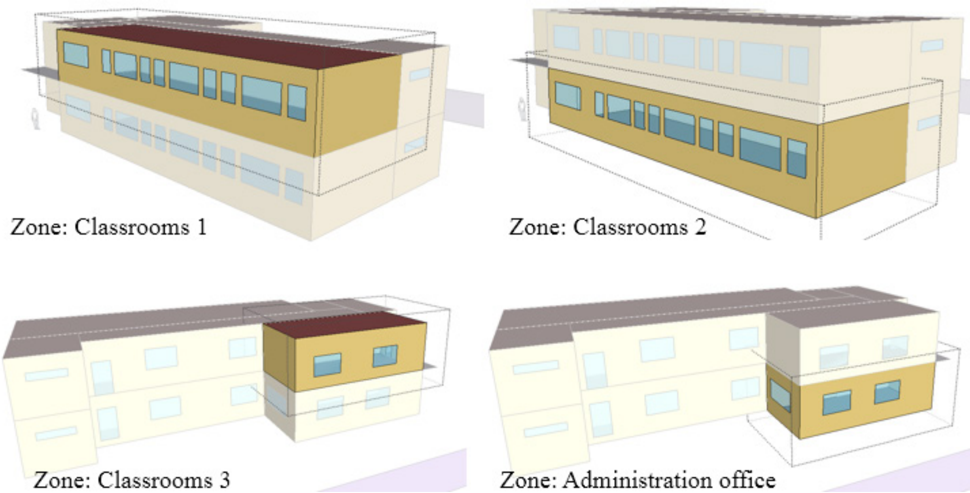

Figure 3. Thermal zones modeled and included in the study.

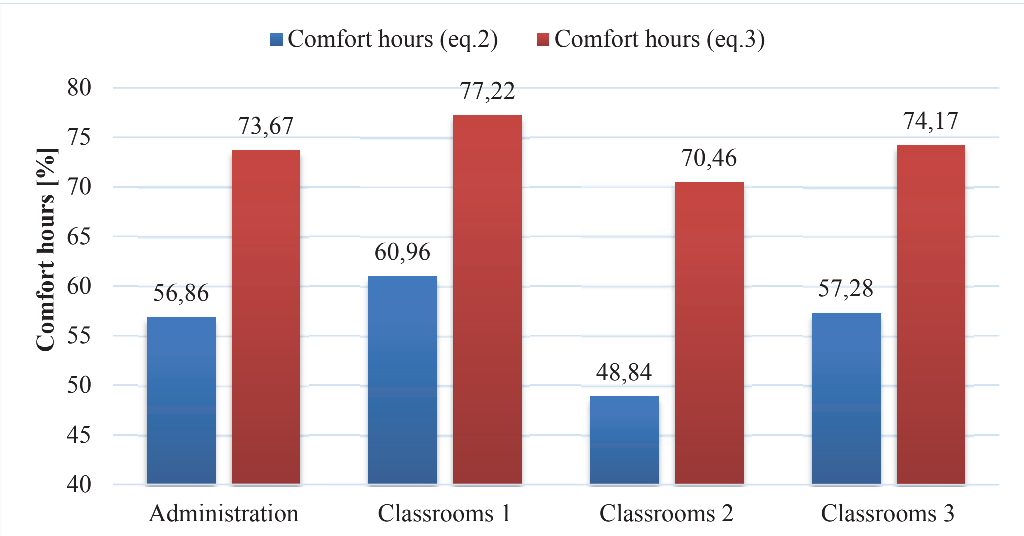

Figure 4. Comfort hours percentages.

The parametric analysis performed includes different scenarios that have been compared to assess the potential for comfort improvement through natural ventilation in the case-study:

- Scenario A, the state of the art of the building, with no natural ventilation strategies implemented,
- Scenario B1, daily ventilation (8:00, 20:00) with an opening area of the windows equal to 50% of the total windowed area,
- Scenario B2, daily ventilation (8:00, 20:00) with an opening area of the windows equal to 100% of the total windowed area,
- Scenario C1, night ventilation (20:00 – 8:00) with an opening area of the windows equal to 50% of the total windowed area,
- Scenario C2, night ventilation (20:00 – 8:00) with an opening area of the windows equal to 100% of the total windowed area,
- Scenario D1, whole-day ventilation with an opening area of the windows equal to 50% of the total windowed area,
- Scenario D2, night ventilation with an opening area of the windows equal to 100% of the total windowed area.

As example to the already discussed features of the case study and of the parametric analysis performed, **Figure 5** presents hourly results (3rd and the 4th of June, chosen as hot summer days in Sicily) for the transient model, representing air temperatures and air change per hour for the zone Classroom 2 for both the A and D2 scenarios.

The calculated comfort temperature is around 27.5°C and in **Figure 5** the eq.3 comfort zone is reported.

The air temperature in the classrooms in scenario A is never included in the comfort zone: it is always higher than 31°C and peaks in the morning and in the afternoon reach 35°C.

The internal loads drive the energy balance of the classrooms. Although external temperature reaches its peak between 14 and 16 during the day, indoor temperature drops by a couple of degrees before raising again in the afternoon following occupation: this clarifies the need for a careful ventilation to disperse excess heat from the rooms.

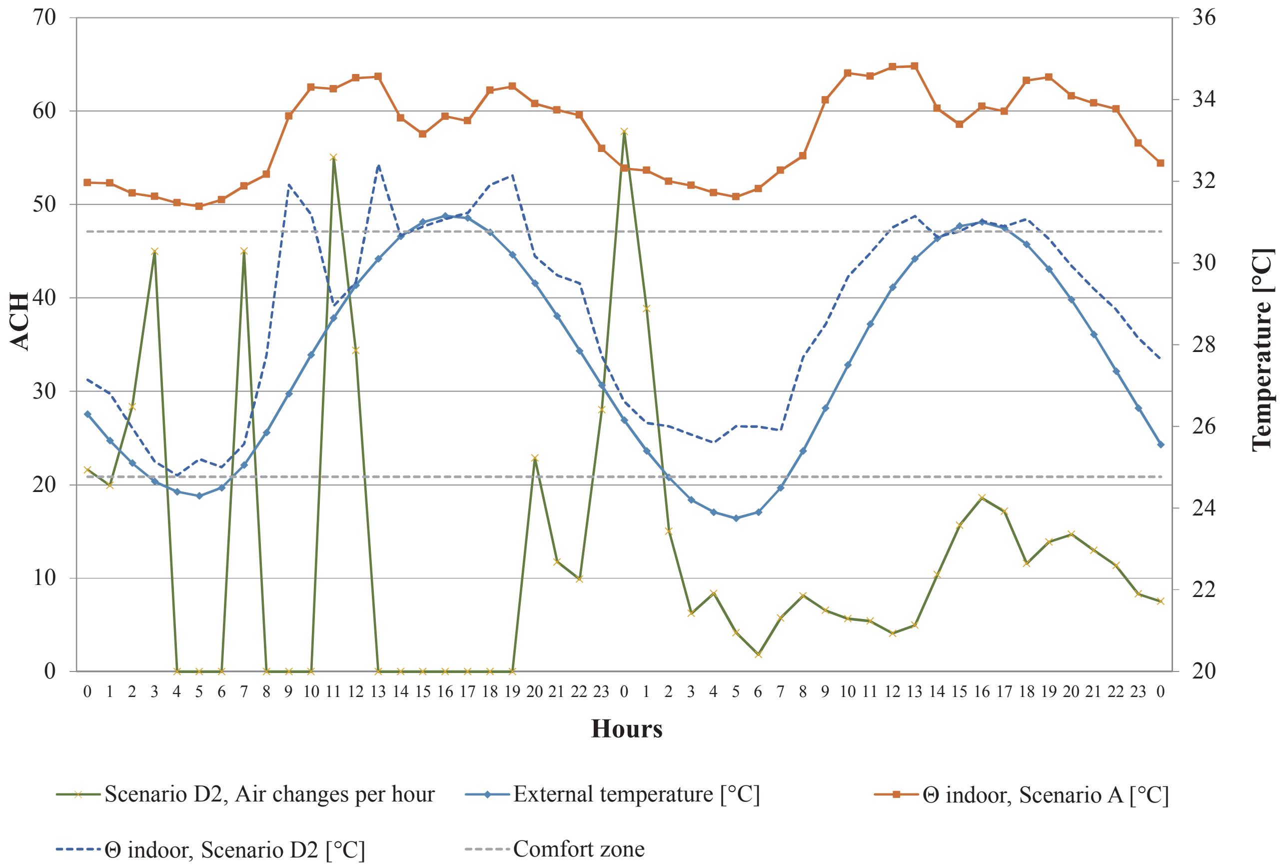

Figure 5. Hourly transient results for the thermal zone classrooms 2.

The D2 scenario proves suitable since during the selected days the external temperature is for most of the hours inside the comfort range.

It may be argued that during the daytime a finer control of the windows opening (e.g. closing windows if exterior temperature rises above 28°C) would slow the increase of indoor temperature (T indoor, Scenario D2) from hours 32 to 36 and thus night and early morning ventilation could be more appropriate.

It is easy instead to verify the positive impact of daytime ventilation even at higher temperatures: as soon as ventilation rates drop, indoor temperature rises immediately even by two degrees in the following hour due to high internal loads, if the zone is occupied (e.g. at 9 or at 13 hours).

Table 1. Results of the parametric analysis: percentage of comfort hours.

	±(2°C)	±(3°C)	±(2°C)	±(3°C)	±(2°C)	±(3°C)	±(2°C)	±(3°C)
Scenario	Administration		Classrooms1		Classrooms2		Classrooms 3	
A	56.86	73.67	60.96	77.22	48.84	70.46	57.28	74.17
B1	61.07	78.51	65.48	82.87	53.38	75.35	61.72	79.23
B2	62.43	80.29	66.64	83.98	54.96	76.92	61.98	79.82
C1	63.03	81.03	66.98	84.34	55.80	78.92	62.35	80.08
C2	65.08	82.08	67.59	85.03	56.75	80.25	62.97	80.60
D1	66.11	83.65	69.05	86.07	57.89	81.63	64.45	82.50
D2	66.38	84.03	69.37	86.46	59.18	81.91	65.04	83.20

Global results for all the scenarios discussed are following in **Table 1**.

Results identify improvements in all the classrooms of up to nearly 10% in the D2 comfort scenario that reports the highest increases.

Conclusions

The paper proposes the quantification of thermal comfort improvement obtainable in a school building in Sicily through the use of natural ventilation. The building is characterized by overheating from May to October due to high internal loads and solar gains: natural ventilative cooling is a potential solution to improve the thermal comfort of the occupants.

Natural ventilation scenarios include daytime ventilation, night-time ventilation as well as whole-day ventilation: all the solutions investigated allow for an increase in the percentage of hours of comfort on the total of occupied hours.

Of the scenarios proposed, the best proves to be the 24 hours natural ventilation strategies (D1 and D2) in all the thermal zones: although the features of the summer external temperature trend would probably suggest not to ventilate much during peak hours, the features of the spaces – characterized by high internal loads – require ventilation during most occupied hours. ■

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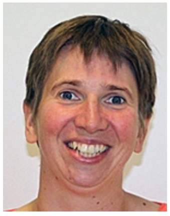
Design of a new nZEB test school building



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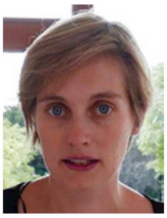

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A new nZEB school building is built at the Technology campus Ghent of KU Leuven (Belgium). The aim of this project is to realize a test facility with real users. The design process is documented with focus on the difficulties to reach a good thermal summer comfort.

Keywords: nZEB, school building, ventilative cooling, night ventilation, summer comfort, test facility.

Overview of requirements

A new nZEB school building is realised at the Technology campus Ghent of KU Leuven (Belgium) on top of an existing university building (see **Figure 1**). The building contains 4 zones (see **Figure 2**): 2 large lecture rooms (zone 1 and 2), a staircase (zone 3) and a technical room (zone 4). The lecture rooms have a floor area of 140 m², a volume of 380 m³ and a maximum capacity of 98 students each. Windows are provided on both sides of lecture rooms, the window-to-wall ratio is 27% (SW) and 26% (NE), the window-to-floor ratio is 14%.

The aim of this project is to realise a school building that is used as normal lecture rooms but at the same time is a test facility for research on building energy-efficiency strategies in a "real use" environment. Therefore, the 2 lecture rooms are thermally insulated from the outside, the neighbouring zones and each other. The lecture rooms are also designed as identical zones with a different thermal mass. The lower class room has a brick external wall with exterior insulation. The upper class room has a lightweight timber frame external wall with the same U-value. Both lecture rooms have a concrete slab floor.

Figure 1. nZEB school building at Technology campus Ghent of KU Leuven (Belgium).

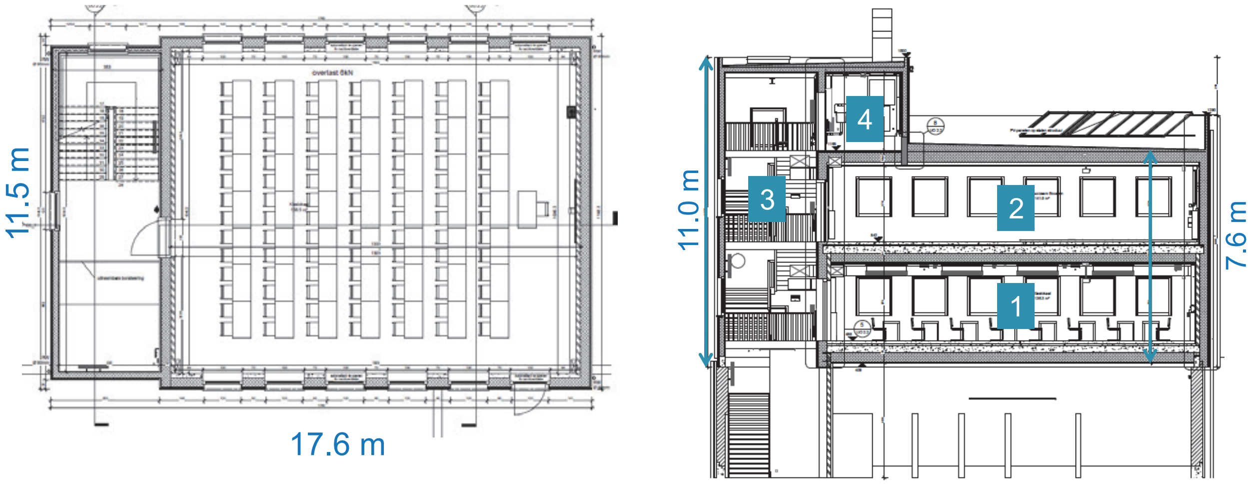

Figure 2. Floor plan of lecture room (left) and section of test school building (right).

Moreover, the building is designed according to the passive house standard. This requires a highly insulated and airtight building ($n_{50} < 0.6\ h^{-1}$) with a net heating and net cooling demand ≤ 15 kWh/m².a [1]. This is a challenge for a school building with (1) a dense occupancy (1.4 m²/student) and thus high internal heat gains, (2) high ventilation rates and thus ventilation losses (5.8 h^{-1} in the lecture rooms) and (3) intermittent use. In addition, an increased need for cooling and high temperatures in summer and shoulder season are expected in this highly insulated and airtight school building.

This paper discusses how to fulfil the requirements of a passive school building (focusing on heating) while guaranteeing a good thermal summer comfort. Design choices are considered and the final solution, i.e. the building with its HVAC systems and control, are presented.

Design choices

Net energy demand for heating

First, the net energy demand for heating of this school building design is evaluated. This demand is calculated by the quasi-steady state calculation method PHPP 2007 [2] adapted for school buildings, based on EN 13790.

The following assumptions are made. The occupant density during courses is 1.4 m²/pers. This value is comparable to the occupancy that DIN V 18599 assumes for auditoria rather than for classrooms. This density is increased to 200% during exams. An absenteeism of 10% is taken into account. The operational schedule: the academic year counts 124 days with courses and 63 days with exams (in January, June and August-September). Holiday periods are in April (2 weeks), July and the first half of August (6 weeks) and December-January (2 weeks). The lecture rooms are in use from Monday to Friday between 8h10 and 16h30. The internal heat gains amounts to 80 W/pers. due to occupants, 1 W/m² due to equipment and 6 W/m² due to lighting according to [3]. The ventilation rate is 29 m³/h.pers. [3] corresponding to IDA3 in EN 13799. The ventilation system consists of a balanced mechanical ventilation provided with an air-to-air heat exchanger with an efficiency of 75%. Building envelope has a U-value of 0.15 W/m²K and includes triple glazing windows (U = 0.85 W/m²K and g = 0.52). The average indoor temperature is 19.4°C [3]. Monthly average measured weather data (1971–2000) for the meteorological station of Uccle (Belgium) [3] is used.

This calculation results in a yearly net heating demand of 11 kWh/m².a. This means that this building design fulfils the requirements of Passive House standard concerning the net energy demand for heating in school buildings [1]. **Figure 3** shows the monthly heat losses, heat gains and net heating demand. A short heating season is noticed, from November till March-April. Moreover, large heat gains are noticed during the whole year, particularly caused by the high occupant density. This result shows the necessity to examine the thermal comfort, especially the overheating.

Thermal summer comfort

Thermal summer comfort and net cooling demand are evaluated using dynamic simulations in TRNSYS [4].

Following assumptions are made, in addition to the preceding assumptions for the calculation of the net heating demand. A hot weather data set for Uccle (Belgium) [5] with warm temperatures with an occurrence once every 10 year is used. The simulation time step is 10 min. Moveable external sunblinds are provided

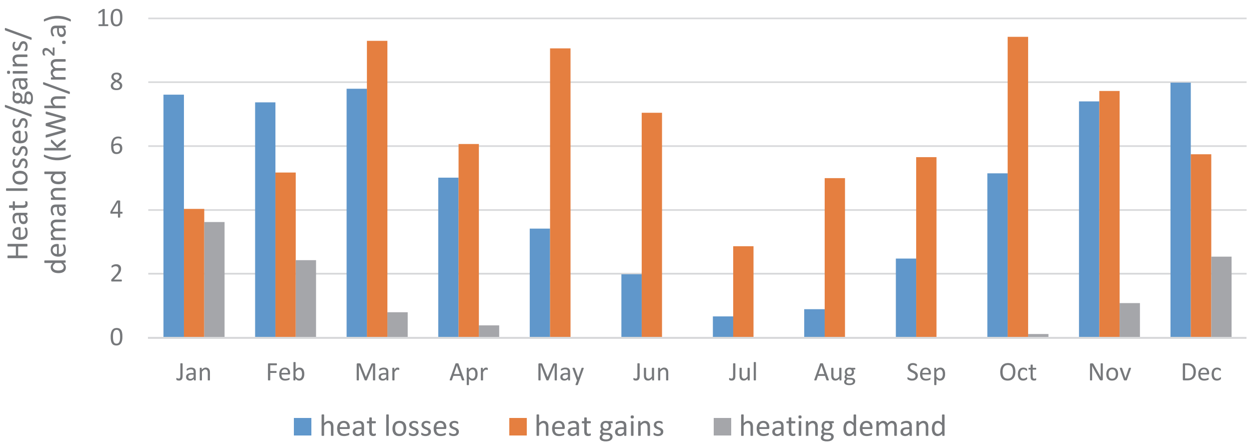

Figure 3. Monthly heat losses, heat gains and net heating demand (in kWh/m².a).

on both façades. The threshold of total solar radiation when blinds are closed or opened is 250 W/m². In case the blinds are closed, 75% on the shaded surface is blocked. The air-to-air heat exchanger is bypassed when the indoor operative temperature exceeds 22°C and the outdoor air is cooler than the indoor air. The minimum supply air temperature is 14°C.

The impact of night ventilation on thermal summer comfort is studied. Night ventilation is activated at night (between 0h and 6h) when the indoor air temperature exceeds the outdoor air temperature with at least 2°C and the surface temperature of a massive wall exceeds 21°C.

Figure 4 evaluates thermal summer comfort in both lecture rooms and shows the effect of night ventilation. The amount of hours exceeding 26°C (in % of the time in use) is shown. As expected, without night ventilation, none of the rooms meet the requirement of overheating hours less than 5% of the time in use according to EN 15251. A significant impact of night ventilation is depicted in **Figure 4**. However, the requirement for good thermal summer comfort is not fulfilled in the upper lecture room with timber frame external wall. In addition, **Figure 5** shows the indoor temperatures in both rooms during a week at the end of May. Thermal comfort is reasonable but not excellent. This means that additional (ventilative) cooling is needed to reach a good level of thermal comfort in summer, spring and autumn in both lecture rooms.

On the contrary, the net cooling demand fulfils the PassiveHouse requirement of 15 kWh/m².a, even without night ventilation when looking at the whole building (see **Figure 6**). When night ventilation is applied, both lecture rooms have a net cooling demand

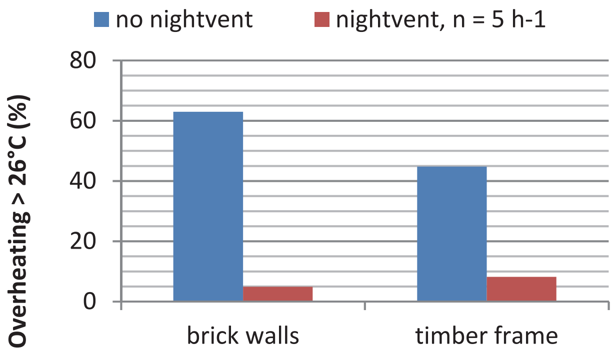

Figure 4. Evaluation of thermal summer comfort.

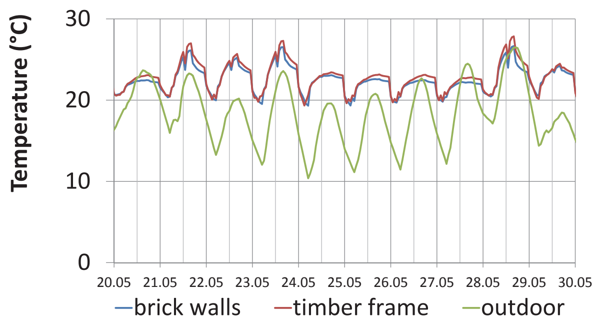

Figure 5. Indoor temperature in both lecture rooms (with night ventilation).

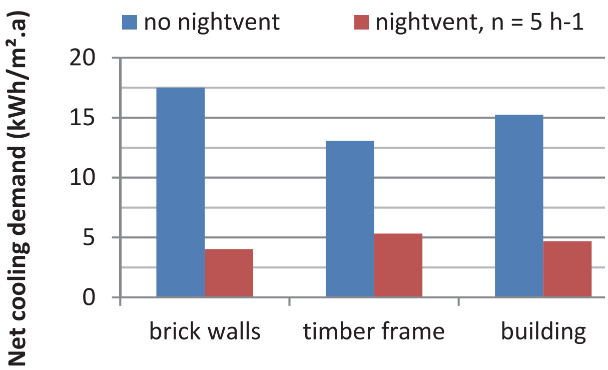

Figure 6. Evaluation of net cooling demand.

of less than 6 kWh/m².a. This means that assessment of the net cooling demand of a whole building does not guarantee a good thermal comfort in the separate zones.

Final solution

HVAC and control systems

The building is equipped with an all air system with balanced mechanical ventilation with a the total supply airflow of 4 400 m³/h (i.e. 22.4 m³/h.pers). **Figure 7** depicts this HVAC system with the AHU in the technical room and the distribution in the staircase and in the lecture rooms. Demand controlled ventilation with 4 VAV boxes control the airflow based on CO_2-concentrations in the rooms. For heating purposes, the air is preheated by air-to-air heat recovery, i.e. two cross flow plate heat exchangers connected in series, with an efficiency of 78%. Additionally, heating coils of 7.9 kW each are integrated in the supply ducts in each lecture room.

The building is cooled by three techniques of ventilative cooling: (1) a modular bypass in the AHU (2) hybrid night ventilation (natural supply by opening the windows and mechanical exhaust) and (3) indirect evaporative cooling (IEC) with a maximum capacity of 13.1 kW. In addition, external sunblinds (screens) are provided on the southwestern façade and automatically controlled.

Artificial lighting has a low normalized power density (1.49 W/m².100 lx) and is equipped with daylight control system, i.e. a closed loop system with centrally positioned daylight sensor.

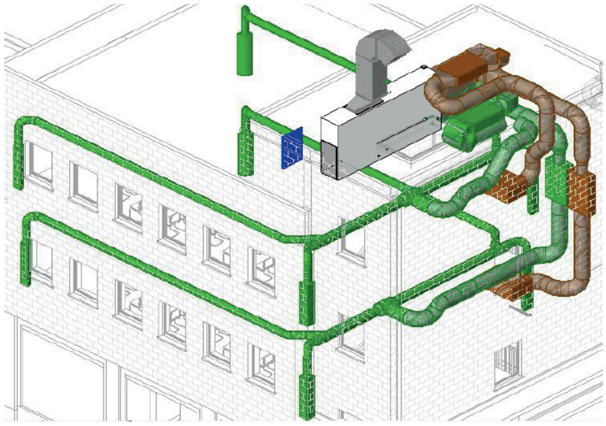

Figure 7. Ventilation system (green is supply, brown is extraction).

Monitoring set up and first results

A set of sensors has been installed to monitor indoor and outdoor conditions and is described in [6]. The building has its own weather station which monitors the main outdoor parameters: solar radiation, the outdoor temperature and humidity and the wind speed and direction. For the indoor conditions, the indoor temperature, the CO_2 concentration and the indoor humidity are continuously monitored. Occupancy level is measured by motion detection sensors and high definition camera with face recognition.

The first monitoring results of this building are available and discuss the operation and ventilation efficiency of demand controlled ventilation [7]. ■

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Ventilative cooling in shopping centers' retrofit

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Because of the customers' need of best possible comfort condition and satisfaction, shopping centers are conditioned by means of basic HVAC systems, often without considering the potential of natural ventilation to reduce energy consumption related to cooling and ventilation. Within the European project CommONEnergy, EURAC researchers are dealing with ventilative cooling strategies as retrofit solutions for shopping centers.

Keywords: Ventilative Cooling Potential, Ventilate Cooling Design Method, Shopping centers retrofit, Integrated Modelling Environment (IME), Cooling Energy Savings.

Introduction

Nearly all retail locations use full air HVAC systems to ensure adequate air exchange, primarily for hygienic reasons, and indoor comfort temperatures. Considering the trend towards longer opening hours and increased number of opening days, the electricity consumption due to ventilation and conditioning systems is expected to continue to rise across Europe.

Specific inefficiencies, related to cooling and ventilation topics, concern mainly energy losses in ventilation, absence of free cooling strategies and unmodulated airflow for different periods of the day.

Generally, mechanical ventilation systems are preferred to natural ventilation strategies because more controllable and reliable, since they are not affected by the uncertainty of natural forces. Thereby, within the design process the team never focused neither on opening sizing nor on control strategies definition for natural or hybrid ventilative cooling systems. So far, shopping centers design has included a small proportion of automated windows, sized for smoke ventilation only. Depending on the external climate conditions, acceptable levels of thermal comfort and IAQ can be reached without or with partial use of the mechanical systems, leading also to operational and maintenance cost savings.

According to the British Council of Shopping Centre (BCSC, 2012), in the UK climate annual electricity usage is known to be up to 50% less where natural ventilation is employed over mechanical ventilation depending on the mechanical systems. Furthermore, natural ventilation uses typically between 2–5% less plant space versus 5–8% used by HVAC, which can be utilized and improve net to gross ratios. Therefore, the CommONEnergy project (http://www.commonenergyproject.eu/) investigates ventilative cooling strategies among the energy efficient solutions for the retrofit of shopping centers' common areas (shop galleries and atria).

As case studies we referred to the reference buildings (**Table 1**) identified within the CommONEnergy project as representatives of the EU building stock, showing the typology they belong to (Bointer,R, 2014), and the climate classification according to (Cory S., 2011).

Technical components

Technical components for ventilative cooling are already available on the market and, according to the IEA Annex 62 state-of-the-art analysis, are structured in:

- Airflow Guiding Ventilation Components, such as windows, skylights, doors, dampers, flaps, louvres plus special effect ventilators;
- Airflow Forcing Ventilation Components, such as such as buoyancy chimneys, solar chimneys, atria, Venturi roofs, powerless roof ventilators, wind towers and wind scoops;
- Passive cooling elements, such as convective cooling components, adiabatic cooling components, phase change components

Some of the component and concept available on the market are shown in **Figure 1**.

Design concept

A ventilative cooling strategy involves the whole building envelope. Vents and openings can be located on both façade and roof to exploit buoyancy due to temperature difference between shops and central spaces and along the atrium height. Applying ventilative cooling is dependent on building design and indoor spaces layout. Where implementing ventilative cooling strategies, it is important to take into account the following features regarding shopping centers' internal layout:

- Interconnected galleries and atria;
- Building shape, number of levels and ceiling height;
- Location of parking areas, possibly avoiding the inlet of polluted air.

The assessment of the most suitable ventilative cooling strategy should also consider that, from an architectural point of view, most of the shopping centers are generally similar to atria with large open spaces between shops that are often heated, cooled and ventilated separately from the mall central space. The shops are connected with the common central areas by means of open doorways through which natural air exchange occurs bridging the two spaces.

For instance, common central areas can be seen as unconditioned buffer zones that temperate the outside and inside climate, resulting in more relaxed ranges of interior conditions respect to selling area. The exploitation of airflow driven both by thermal buoyancy and by wind pressure can prevent overheating within these buffer zones if solar radiation is properly controlled.

As last important consideration, typically the common areas are managed by a unique referent (e.g. owner, energy manager), which is also the one who makes the decisions during a retrofit easing the retrofit process. Furthermore, there is a higher degree of freedom in the common areas design compared to the "leasing" area,

Table 1. List of reference shopping centers' typology, location and climate (HD=heating dominated, CD= cooling dominated, H&CD= mixed dominated).

ID	Name of the shopping center	Shopping Center typology	Location	Country	Climate
CS	City Syd	Medium Shopping center	Trondheim	NO	HD
ME	Mercado del Val	Specialized and Others	Valladolid	ES	H&CD
GE	Genova Ex- Officine Guglielmetti	Specialized and Others	Genoa	IT	CD
KA	Centro Commerciale Katané	Medium Shopping center/ Hypermarket	Catania	IT	CD
MO	Modena Canaletto	Specialized and Others	Modena	IT	CD
DO	Donau Zentrum	Very Large Shopping center	Wien	AT	H&CD
BC	Brent Cross	Very Large Shopping center	London	UK	H&CD
ST	Studlendas	Small Shopping center	Klaipeda	LT	HD
GB	Grand Bazar	Small Shopping center	Sint-Niklaas	BE	H&CD
WA	Waasland Shopping Center	Large Shopping center	Antwerp	BE	H&CD
PA	Pamarys	Small Shopping center/ Hypermarket	Silute	LT	HD

where franchising companies characterized by their own standardized protocols, restraint the applicability of overall retrofit solutions.

Ventilative cooling potential

The potential of ventilative cooling strategies is evaluated for each reference building using the ventilative cooling potential tool (Belleri A., 2015). The tool, which is under development within the IEA EBC Annex 62 research project (IEA EBC Annex 62 - Ventilative cooling, 2014-2017), takes into account also building envelope thermal properties, internal gains and ventilation needs.

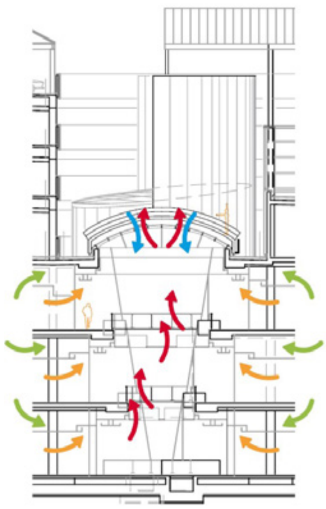
For each hour of the annual climatic record of the given locations, an algorithm splits the total number of hours when the building is occupied into the following groups:

- Ventilative Cooling mode [0]: no ventilative cooling is required because heating is needed;
- Ventilative Cooling mode [1]: natural ventilation can be exploited to meet the minimum ventilation rate required by the EN 15251 on indoor environmental quality;
- Ventilative Cooling mode [2]: ventilative cooling is needed and the ventilation rates needed to maintain indoor air conditions within the comfort ranges (calculated according to the adaptive comfort model – EN 15251:2007) are assessed according to the energy balance;
- Ventilative Cooling mode [3]: ventilative cooling is not useful and nighttime ventilation should be considered.

The graph in **Figure 2** reports the results of the ventilative cooling potential analysis for each reference building considering a specific lighting gain level of 30 W/m². The higher is the level of specific lighting gains the high is the cooling need and consequently the energy savings related to the use of ventilative cooling potential are higher. A parametric analysis for the definition of the ventilative cooling potential according to different levels of specific lighting gains is reported in (Avantaggiato M., 2015). Direct ventilative cooling strategy can potentially assure thermal comfort for the whole hours within a year in Trondheim and for more

(a)

(b)



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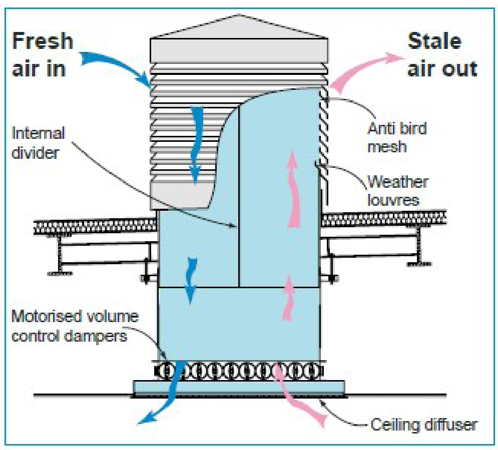

(d)

Figure 1. (a) Skylight window installation at Ernst-August Galerie in Hannover. Source: http://www.hs-montagen.de/archiv-montage-fassade-hannover, (b) Chain actuators for top hung windows. Source: http://www.topp.it, (c) Ventilation concept of the Ernst-August galerie in Hannover. Source: www.windowmaster.com, (d) Wind catcher application on Tesco supermarket. Source: Monodraught, 2015.

than 90% in seven climates over eleven. The cooling dominated climates of Genoa, Catania and Modena are the ones with the highest percentage of direct ventilative cooling with increased airflow rate potential use (ventilative cooling mode [2]).

Design method

Naturally ventilated buildings require a specific design service dealing with building shape, internal layout distribution and airflow paths along the building. Therefore, natural ventilation design shall be ideally part of an integrated design process since the early design stages.

Since shopping centers are mostly object of partial retrofitting actions, building shape cannot be modified and internal layout can be only partially modified. However, typical architectural archetypes of shopping centers such as atria and galleries revealed to be suitable for the integration of natural ventilation strategies.

Based on climate analysis previously described, the most suitable ventilation strategy can be assigned by identifying possible airflow paths and the air intake and exhaust locations. It is necessary to integrate the natural ventilation in the overall existing building design, especially in relation to area partitioning (shops, common areas, areas closed to visitors), air tightness, building geometry, HVAC system and envelope porosity.

Designers are faced with many and sometimes conflicting requirements by designing natural ventilation in shopping centers, related to urban regulation, indoor environment quality, aesthetic appearance, building standard and regulations (acoustic, fire, zoning..), safety, operative and maintenance costs and the need to maintain the shopping center open during the retrofitting works. Those constraints are related to the design complexity and can be easily identified through an integrated design process by discussing with design team, building owner, energy manager and other actors directly involved in the building design.

The scheme in **Figure 3** represents the design process adopted to define ventilative cooling solution.

Considering that an indoor space of a shopping center highly interacts among each other, a multizone based analysis of airflows is needed to evaluate the ventilative cooling strategy effectiveness and to assess potential energy savings.

The Integrated Modelling Environment under development within the CommONenergy project allows to predict airflows throughout a building by performing coupled thermal and airflow building dynamic simulations. Furthermore, by gathering in the same simulation model (i) building (ii) HVAC and refrigeration systems and components (iii) daylighting/shading/artificial lighting (iv) storage technologies (v) RES technologies (vi) natural ventilation and infiltration (vii) non-conventional envelope solutions (vegetation, multi-functional coating and materials, etc.), the Integrated

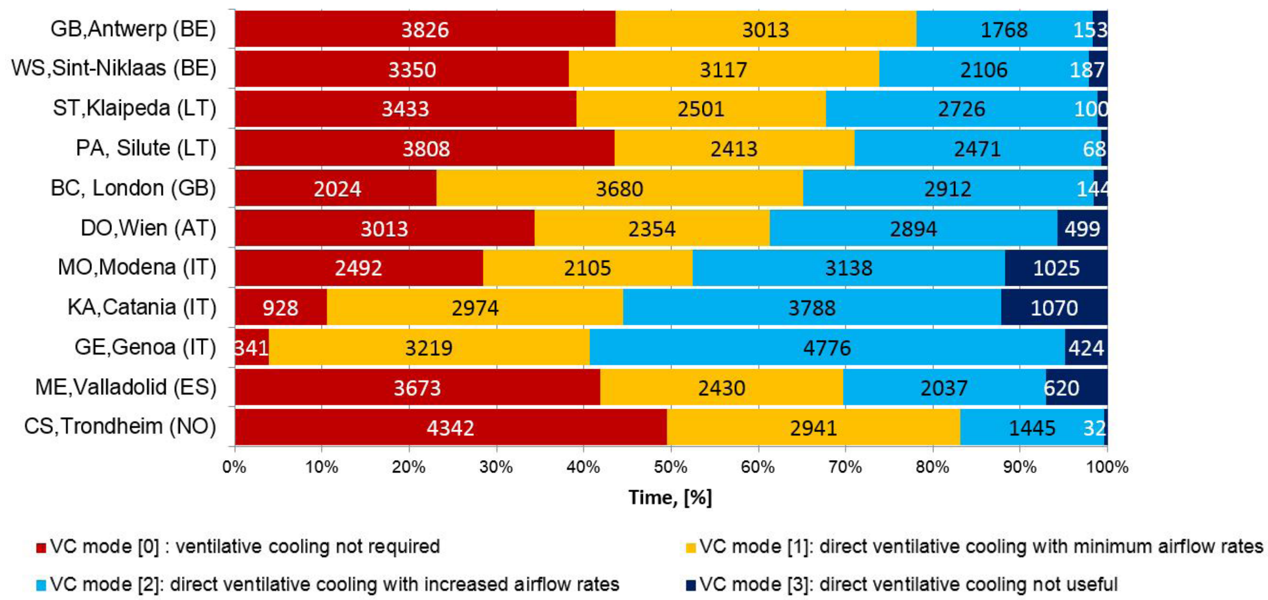

Figure 2. Percentage of hours within a year when direct ventilative cooling is required, useful or not useful in the eleven reference case climates when the specific lighting power is 30 W/m².

Modelling Environment allows to test the energy performance of ventilative cooling solutions in combination with other active or passive solution and to elaborate effective solution sets and control strategies.

Cost optimization can be performed by properly sizing each technical component (openings, actuators type, vents etc.). The modelling results are then used as basis for discussion with the building owner on the definition of the retrofit solution and its installation mode within the shopping center.

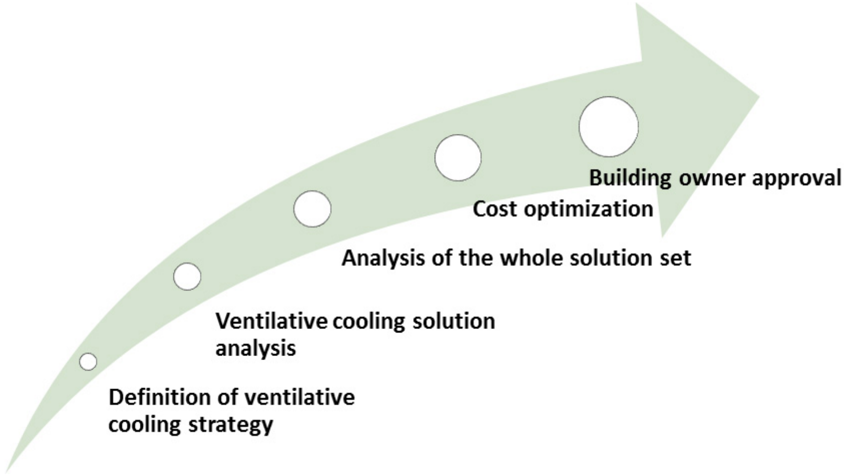

Figure 3. Ventilative cooling solution design process.

Conclusion

The CommONEnergy project investigates ventilative cooling strategies as energy efficient solutions for the retrofit of shopping centers. The paper investigates the retrofit opportunities to exploit ventilative cooling in retail buildings taking into account climate condition, architectural features and level of retrofit. As case studies we referred to the reference buildings identified within the CommONEnergy project as representatives of the EU building stock. Typical architectural archetypes of shopping centers such as atria and galleries revealed to be suitable for the integration of natural ventilation strategies. A climate analysis showed that the ventilative cooling potential is suitable for all the eleven climates analyzed with difference in the percentage of hours of utilization and in the airflows needed to offset the internal gains.

A ventilative cooling strategy involves the whole building envelope as vents and openings can be located on both façade and roof to exploit buoyancy due to temperature difference between shops and central spaces and along the atrium height. Technical components needed for ventilative cooling are already available on the market but the performance of a ventilative cooling strategy is strictly dependent on building design and indoor spaces layout.

Considering that an indoor space of a shopping center highly interacts among each other, a multizone based analysis of airflows is needed to evaluate the ventilative cooling strategy effectiveness and to assess potential energy savings. The Integrated Modelling Environment under development within the CommONenergy project allows predicting airflows throughout a building by performing coupled thermal and airflow building dynamic simulations. ■

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Single-sided ventilative cooling performance in a low energy retrofit

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Externally applied building envelope retrofit measures intent on upgrading thermophysical properties and air tightness performance can result in substantially modified cooling and ventilation requirements for a given internal space. Field studies presenting demonstrated performance of such situations are still relatively infrequent. The article presents findings from a case study retrofit project (zero2020 testbed) at Cork Institute of Technology in Ireland. Measured performance of an externally applied fenestration module incorporating purposed provided ventilation openings is presented and discussed. Findings show that, depending on configuration, acceptable ventilations rates and internal thermal environments are possible using a single sided ventilative cooling strategy.

Keywords: retrofit, single sided ventilation, purpose provided openings, thermal comfort, zero2020 testbed.

Introduction

Ventilative cooling coupled with exposed thermal mass is widely accepted as an important strategy for reducing summer overheating in non-domestic buildings. Extended monitoring has shown that naturally ventilated buildings typically use less than 50% of the corresponding energy consumption of air conditioned buildings and assessment of ventilative cooling techniques in Europe have shown they may contribute highly to reducing the cooling needs of buildings (Kolokotroni et al, 2008) and be an effective tool for tackling climate change adaptation in existing buildings. Furthermore, increased ventilation rates can also lead to improved work performance. Recently, focus for market activation in the construction sector has shifted towards dealing with the overhaul of the existing building stock. The Irish National Energy Efficiency Action Plan 2013–2020 report has identified refurbishment of existing public sector buildings as a key focus. The report states that there are over 10,000 existing public sector buildings in Ireland. In responding to these external drivers Cork Institute of Technology in Ireland (CIT) have recently completed a pilot project/research testbed, zero2020, for the low energy retrofit of their existing 29,000 m² teaching building constructed in 1974. The retrofit pilot project covered 1.5% of the total building floor area and is shown in **Figure 1**. At both concept and design stage there were no guidelines within the Irish context upon which to base performance targets to achieve a near zero energy building (NZEB) through retrofit.

The design proceeded along a simple strategy of firstly, ensuring compliance with the environmental specification for occupant comfort and secondly to achieve the best fabric and energy performance subject to constraints imposed by budgets and retrofit/structural limitations. The final solution consisted of design and installation of a structurally independent external envelope solution. This resulted in U-values for opaque element 3 times better than current regulations and glazing U-values 6 times better. Further details of the design and specification of the retrofit solution can be found in (O'Sullivan et al. 2013). The objective of this article is to present measured performance of the retrofitted single sided natural ventilation system that utilises a purpose provided slot louvered opening (see **Figure 2**). Ventilation rate performance along with objective and subjective thermal comfort performance of the retrofit building have been experimentally investigated.

Purpose provided ventilative cooling components

For most enclosed spaces in the existing building (left of **Figure 1**) the ventilation system is based on single sided top hung pivoting window sections. There is generally one opening window per structural grid. In the retrofit space fenestration system, the ventilation module uses a flush faced external louvre with individual air inlet sections with 2 ventilation sections per structural grid. On the internal side of the slot louvres there are automated high level insulated doors and manual low level insulated doors providing different control mechanisms. The installed slot louvre system has a net 50% free open area for airflow and overall structural opening dimensions are 0.30 m (w) x 1.60 m (h) with a net opening area of 0.102 m² (e.g. in a single cellular office space there are 2 openings at low level and 2 openings at high level in the test space). Each of the ventilation openings has 17 airflow slots across

Figure 1. External facades; existing 1974 building (left) and retrofitted zero2020 test-bed building (right) where red indicates the location of the thermal comfort study and yellow indicates the location of the ventilation rate performance experiments.

Figure 2. Ventilation system configurations, from left to right: existing building window (CS.01), Retrofit space no ventilation scenario (RS.01), bottom manual louvre only (RS.02), top automated louvre only (RS.03), both louvres opened (RS.04).

the louvre bank. The overall thermal transmittance performance of this unit including doors and linear transmittance is 0.84 W/m²k. The new fenestration module resulted in an overall opaque/transparent area ratio reduction of 20%. Unwanted ventilation through adventitious openings has also been greatly reduced. The retrofit envelope air permeability was tested in accordance with BS EN 13829:2001. The envelope achieved an air permeability of 1.76 (m³/hr)/m² at 50Pa building pressure. The existing structure was measured as 14.77 (m³/hr)/m². In order to quantify the actual range of ventilation rates achievable 38 tracer gas concentration decay tests were completed as part of an experimental field study during summer 2013. The results are summarised in the following section.

Ventilation & thermal performance in cooling mode

To investigate ventilation performance in cooling mode in the retrofitted space an isolated, 6.0 m² west facing, first floor cellular office (highlighted in yellow in **Figure 1**) employing single sided ventilation was used to measure ventilation rates under various boundary conditions. A similar space in the existing building was used for comparative purposes. The field tests were completed in accordance with the procedures set out in ASTM E741-11. Details can be found in (O'Sullivan et al. 2014). All ventilation rate values presented have been calculated using the decay regression technique. **Figure 3** presents boxplot distributions of results from all tracer gas tests completed under each of the different operating configurations in **Figure 2**.

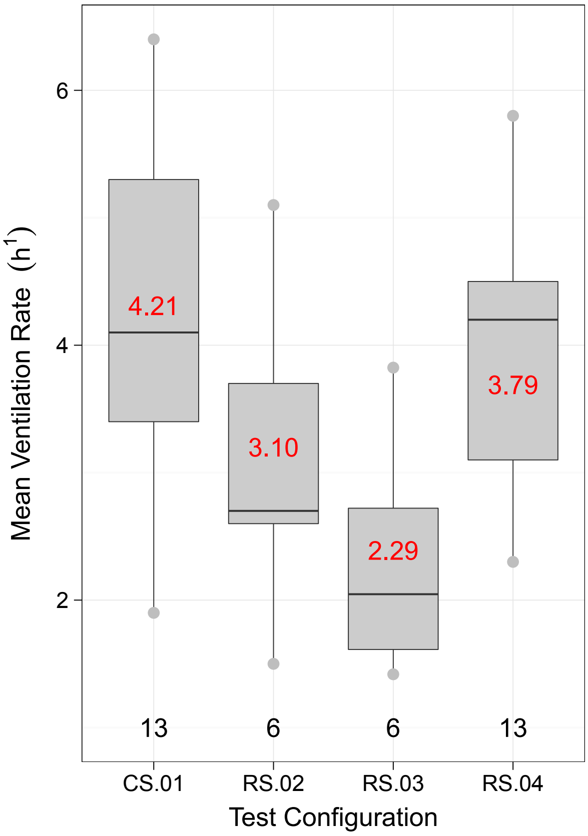

Figure 3. Boxplot distributions of measured mean ventilation rates grouped according to configurations in Figure 2 above (no of tests shown for each, mean ACH shown in red, median shown as bar).

The results show that for a similar spread of boundary conditions the existing control space (top hung window, CS.01) has consistently higher time-averaged ventilation rates with a mean value of 4.2 h^{-1} and standard deviation of 1.5 h^{-1}. In the retrofit space the full height configuration had the best performance profile with a mean value of 3.8 h^{-1} and standard deviation of 1.0 h^{-1} indicating a slightly more concentrated spread of results. Based on the guideline values for indoor air quality classification in BS EN 13779:2007 both the existing control space and retrofit space time averaged ventilation rates can be classified as IDA1 (High). Ventilation rate values were on average lower for the low level RS.02 & high level RS.03 configurations although there were individual instances of values above 5.0 h^{-1}.

As well as ventilation rate performance the potential risk to overheating was also evaluated for each ventilation configuration. Indoor air temperature is often used as an index for evaluating long term risk of overheating in buildings. The extent of exceedance of acceptable conditions is often based on the percentage of annual occupancy hours with indoor air temperatures above a reference exposure threshold value compared to a maximum acceptable percentage exceedance (i.e. 5% of hours above 25 °C for CIBSE, 1% above 28 °C for BRE). To investigate long term performance of the ventilative cooling system, indoor air temperatures were recorded in the single cell retrofit office (highlighted in yellow in **Figure 1**) during an extended period of warm summer conditions (May to October 2013). A range of different configurations were employed in the office during this time. **Figure 4** presents monthly binned percentage of hours' exposure for different reference threshold value. There was some overheating present during July, even with some night cooling in place but in general conditions were acceptable according to the criteria proposed. It should be noted that the percentage will reduce when the full annual hours are factored in.

Thermal comfort perception

In order to evaluate the thermal perception potential of the thermally decoupled low energy space and ventilative cooling system a subjective thermal comfort study was designed and carried out in May 2015 (O'Donovan et al, 2015). The study evaluated all four of the retrofitted ventilation configurations shown in **Figure 2**. The study gathered feedback from 35 participants (10 females, 25 males) as to thermal environment during controlled tests in the open plan seminar room of the building. This is a 42 m² first floor, north facing room employing single sided ventilation (see red in **Figure 1**). Participant feedback on perceived thermal state during the four tests was gathered using two standardised questionnaires based on ISO 10551.

Objective continuous environmental data was also gathered in order to calculate predicted response of participants for each test using the Predicted Mean Vote (PMV) index. **Table 1** indicates the ISO 7730 categories used in this comparison where the values presented are for cooling season performance only. The study was designed to evaluate the capabilities of all ventilation configurations to provide thermal comfort in a simulated overheating scenario, where before each test the seminar room was preheated to 26°C (±1°C).

Figure 5 presents predicted (objective) and subjective frequency of thermal vote for each configuration. Configurations RS.02 (high level only) and RS.03 (low level only) experienced the largest percentage of neutral responses (40%, 43%) with mean thermal sensation votes of –0.40 and –0.49 respectively (Category B). The full height opening configuration (RS.04) had a mean vote of -1.06 putting it outside all ISO7730 thermal environment categories. Subjectively no configuration tested achieved Category A (see **Table 1**).

Table 1. Categories of a thermal environment.

Category	PMV	t_o (°C)
A	-0.2 < PMV < +0.2	24.5 ± 1
B	-0.5 < PMV < +0.5	24.5 ± 1.5
C	-0.7 < PMV < +0.7	24.5 ± 2.5

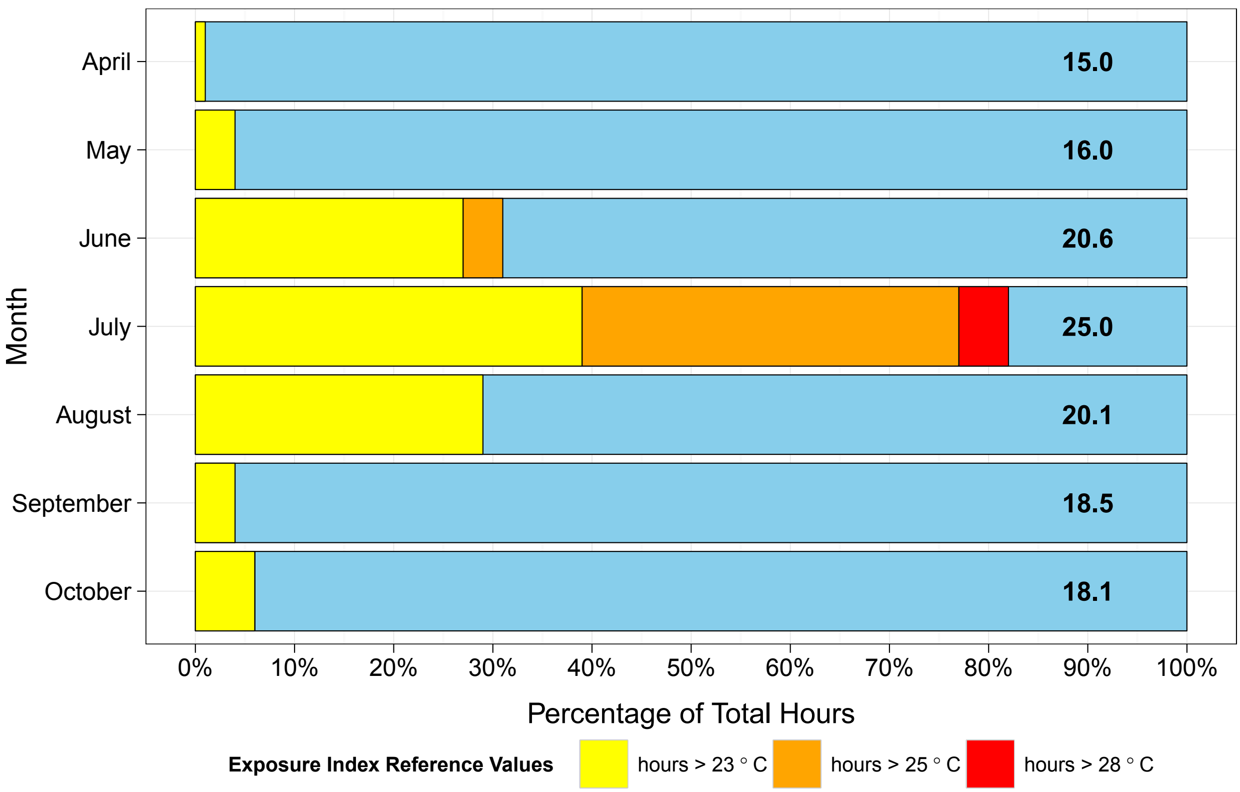

Figure 4. Percentage time exceedance of long term index reference values during extended cooling period in 2013 (Monthly 95th percentile T_{ex} values shown in each month).

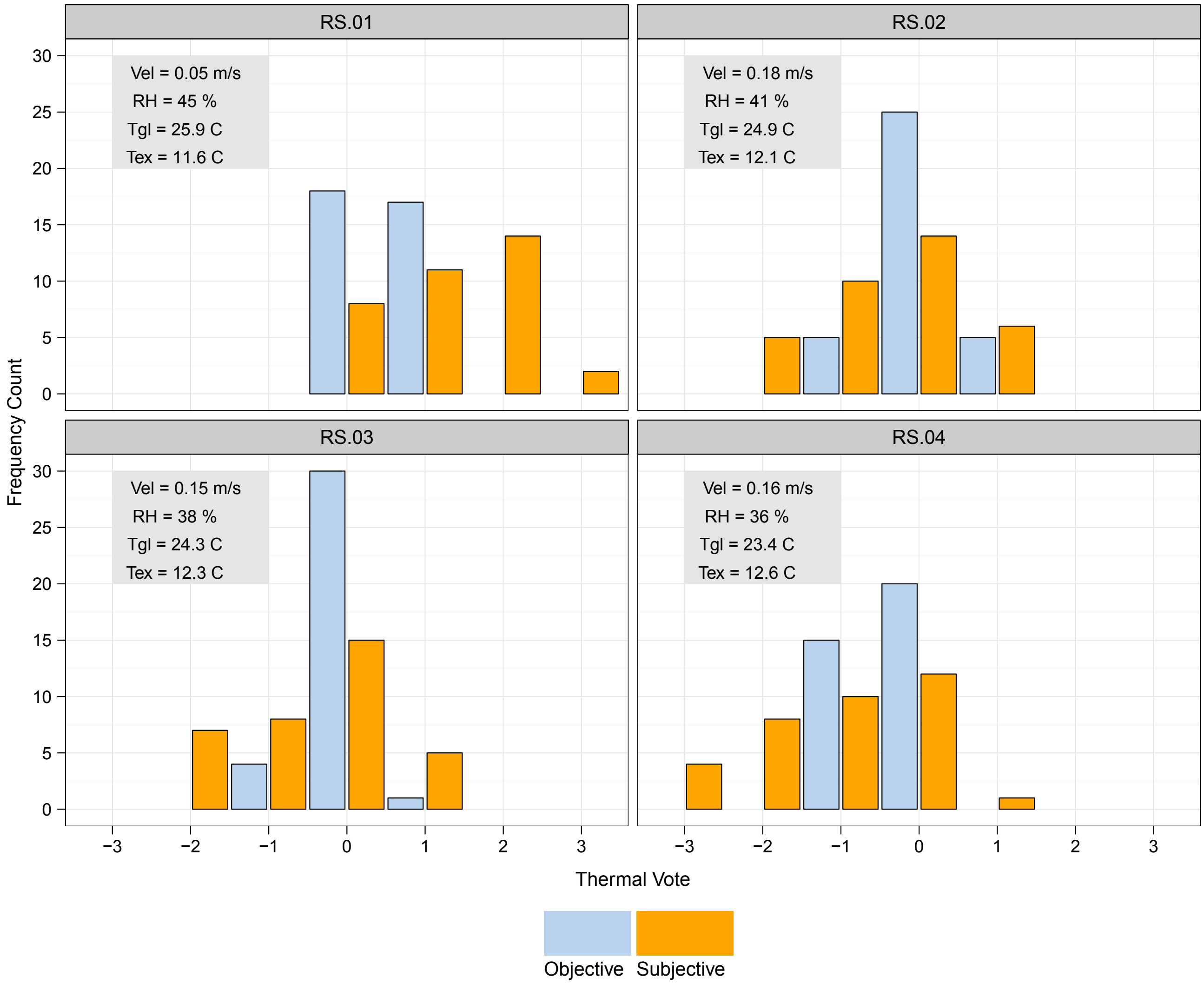

Figure 5. Results thermal comfort study showing objective and subjective thermal sensation votes for retrofitted space ventilation configurations shown in Figure 2 (Vel = Indoor air velocity, Tgl = Indoor Globe Temperature, Text = External Air Temperature, RH = Relative Humidity).

Conclusion

Overall good ventilation rates were achievable in the retrofit space with the new purpose provided slot louvred openings operated using a single sided ventilation strategy. Although there was a reduced envelope temperature difference due to the improved thermal performance of the building resulting in weaker buoyancy forces compared with the existing building the large opening height of the RS4.0 configuration seemed to ensure comparable performance. The thermal comfort study suggests that using high level only openings or low level only openings provided the most satisfactory thermal environments. It is therefore possible to achieve effective ventilative cooling using certain configurations but care must be taken when low external air temperatures are present using large openings that provide air flow directly to the occupied zone resulting in potential overcooling and local thermal discomfort. ■

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Overheating assessment of energy renovations

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In many post-occupancy studies of renovated houses elevated temperatures have been documented. This article presents in which situations overheating need to be addressed and which renovation measures are causing this need. The analysis contains representative houses from central and north Europe. Both dynamic and static overheating assessment criteria are used.

Keywords: renovation, overheating occurrence, single family house, nZEB, adaptive comfort.

Introduction

The greatest share of the European building stock (EU-27) is residential buildings and the majority of them are single-family houses [1]. The current target of the European Council is to have the majority of the existing stock be renovated until the middle of the century. More and deeper renovations (nZEB) are expected in the coming decades.

Energy renovations mainly concern the colder conditions in winter. However, in many comfort studies of energy renovated buildings and nZEB, elevated temperatures have been documented not only during the summer period but also during the transition months, even in Central and North Europe [2]. For the designers, builders and occupants of these areas overheating is an unknown challenge until recently. Moreover, cooling demand calculations are based on simplified monthly methods, averaging the need both in time and space. High temperatures for long periods cause serious impacts to the indoor environmental quality [3]. This article summarizes ongoing scientific work [4].

Case studies

This analysis involves investigation of four representative houses (1960's, 70's and 80's), of U.K. (London city); Denmark (Copenhagen); Austria (Vienna) and South France (Marseille, H3 climatic zone of France). The stock of these countries equates with one third of the European Union building stock [1]. Two out of the four case studies are real buildings extracted from the

TABULA project (Denmark, France, **Figure 1**) and the other two cases are the result of deep statistical analyses (energy certificates; [4]).

Table 1 presents the thermal and technical characteristics of the cases. The case studies represent typical heavy-weight constructions [4]. For the investigation of the overheating risk at room level two bedrooms (6.3 m² of net floor area) facing the south and west orientations (SW bedroom) and the north and east orientations (NE bedroom) were developed.

The applied renovation measures are divided into 2 categories:

- elements (improve of the efficiency) and systems (mechanical ventilation, shading systems)

and have been analyzed in three phases:

- base case; deep renovation (regulation) and nZEB renovation

The renovation in every phase is conducted in steps:

- replacement of windows; improvement of the ceiling; improvement of the external walls; improvement of the floor and improvement of the airtightness

The improvements of the efficiency of the elements is conducted and simulated externally (graphite EPS for the walls, mineral wool for the ceiling and high compressive strength XPS boards for the floor elements).

Figure 1. Examined case studies for Denmark (top) and South France (bottom).

System measures refer to new mechanical ventilation systems (higher capabilities) and shading systems as a package with the new windows.

Three different shading systems have been analyzed:
1. Internal venetian blinds with high reflectivity (0.8);
2. External slat blinds with high reflectivity (0.8) and
3. Fixed pergolas-awnings (0.5m projected)

Table 1. Technical and thermal characteristics of case studies and different renovation phases.

a/a	Period	U_{wall} (W/m²K)	$U_{ceiling}$ (W/m²K)	U_{floor} (W/m²K)	U_{window}, g (W/m²K), -	Infiltration n_{50} (ACH/h)	Storey	Net floor area (m²)
Austria (After 1960)								
Base Case		1.20	0.55	1.35	3.0, 0.67	3.0	2	144.4
Deep Renovation		0.27	0.15	0.30	1.2, 0.6	1.5		
nZEB Renovation		0.15	0.15	0.15	0.8, 0.5	0.6		
Denmark (1973–1978)								
Base Case		0.45	0.45	0.35	2.7, 0.76	5.0	1	116.2
Deep Renovation		0.20	0.15	0.12	1.65, 0.7	1.6		
nZEB Renovation		0.20	0.15	0.12	1.2, 0.6	0.8		
France (1982–1989)								
Base Case		1.00	0.60	1.00	4.6, 0.9	5.0	1	94.2
Deep Renovation		0.43	0.22	0.43	1.5, 0.7	1.4		
nZEB Renovation		0.15	0.15	0.15	0.8, 0.5	0.6		
U.K. (Before 1978)								
Base Case		2.25	0.85	1.35	3.2, 0.8	8.0	2 (semi-detached)	60.3
Deep Renovation		0.30	0.18	0.20	1.6, 0.7	4.0		
nZEB Renovation		0.15	0.15	0.15	0.8, 0.5	0.6		

The movable shadings systems are applied during the non-occupied hours. The mechanical ventilation system increase constantly the ventilation airflow rate from the basic value (0.5 ACH/hr for indoor air quality reasons) to 1.5 ACH/hr in two steps (all day application). The case studies were simulated without mechanical cooling systems. The profile and internal loads reflects a 5-member working family [4].

In the research two widely applied indices for the assessment of the overheating indoors is been used. The first index measures the percentage of the occupied hours with operative temperatures higher than the upper bound of the adaptive comfort temperature (Category II; [5]). The second index measures the percentage of occupied hours with operative temperatures above fixed thresholds, 26°C for bedrooms (also building) and 28°C for living room (static method; [6]).

Results and discussion

The way to building energy efficiency, through elements improvements and without passive or active cooling measures, leads to the increase of the overheating risk indoors. Both methods and all cases show similar peaks and valleys and critical renovation measures (**Figure 2**). The renovation measures for ceiling and external walls (variants 2, 3 and 7; French case) of both renovation phases slightly decrease or increase (neutral effect) the discomfort conditions indoors (**Figure 2**). The g value coefficient of solar gains seems to be the critical parameter, as far as the decreasing of these indices (variants 1 and 6). Additional floor insulation and improvements of the airtightness (variants 4, 5, 8 and 9) of both renovation phases increase overheating hours for both indices. Floor insulation seems to be the most critical renovation measure, in terms of overheating occurrence. Similar conclusion regard the overheating indoors for all the cases may be extruded. Elements' improvements increase also average and maximum indoor temperatures and extend the overheating period (**Figure 3**). For the Austrian and Danish houses the period with overheating incidents starts in May and finishes in October (nZEB renovation).

The static method always shows higher overheating values compared with the dynamic one for every renovation measure, room and case study (similar only in U.K. case). Moreover, rooms facing the northwest orientation overheated less compared with others (SE orientation) for both methods and renovation phases.

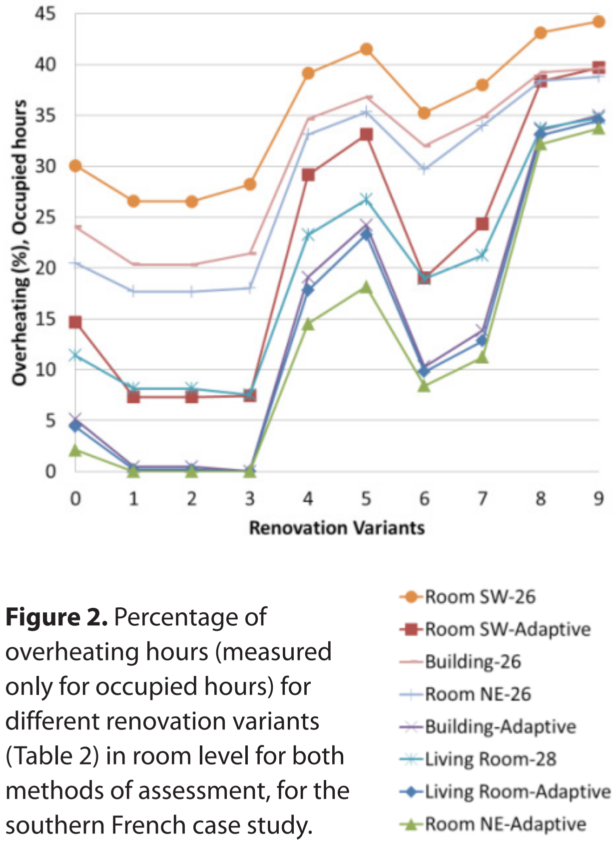

Figure 2. Percentage of overheating hours (measured only for occupied hours) for different renovation variants (Table 2) in room level for both methods of assessment, for the southern French case study.

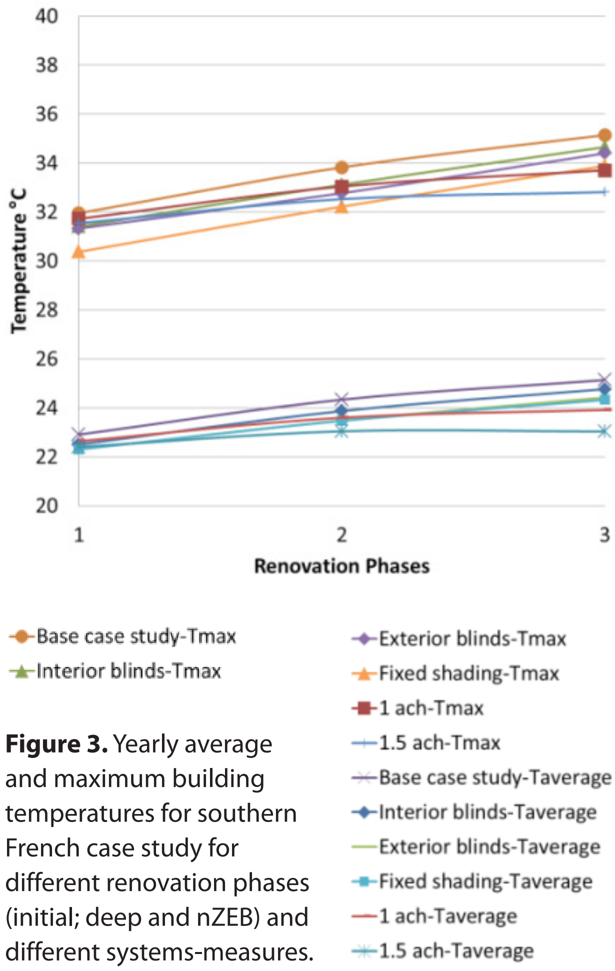

Figure 3. Yearly average and maximum building temperatures for southern French case study for different renovation phases (initial; deep and nZEB) and different systems-measures.

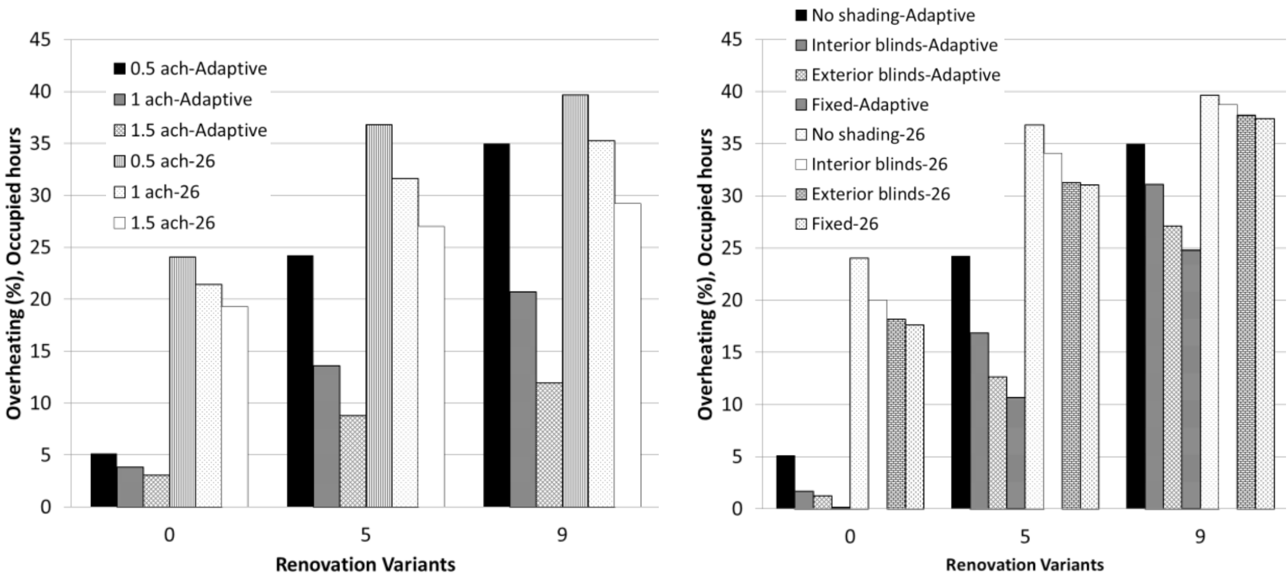

Figure 4. Percentage of overheating hours (measured only for occupied hours) for different renovation variants (Table 2), methods, ventilation rates and shading systems, for the southern French case study.

System improvements are actually antagonistic measures and decrease the overheating risk indoors. The most effective measure, compare to the others, is the increase of the ventilation rates (1.5 ACH/hr) for every case study and renovation phase (**Figure 4**). As far as the shading analysis goes, the shading systems are not very effective for the French case (**Figure 4**). The use of fixed systems or external movable blinds decreases the indices approximately 50% and the internal blinds approximately 25% (Danish case; not presented). **Figure 3** presents the yearly average and maximum building temperatures for both systems and in all renovation phases. Finally, there is an important decrease of the overheating period after the application of these improvements for every case study and renovation phase.

Conclusion

In terms of overheating, the most critical renovation measures are the insulation of the floor and the increase of the airtightness. The contribution of diminishing the g value of the window glazing is positive. Neutral contributions cause the energy improvements of the walls and/or ceiling. Total elements improvements result also an extension of the overheating period and higher average and maximum temperatures indoors. The increase of the ventilation rates of the mechanical systems, close to 1.5 ACH/h, may contribute significantly to the decreasing of the overheating discomfort. The external shading systems may decrease the discomfort effectively, especially to the northern countries. ■

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Ventilative cooling of a seminar room using active PCM thermal storage

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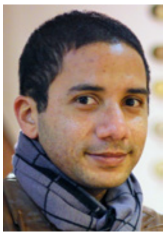

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One-year monitoring results of environmental conditions in a UK seminar room where the Cool-phase® ventilation and PCM battery system has been installed indicate thermal comfort and good indoor air quality throughout the year. CFD analysis indicates that air temperature and air distribution is uniform at occupants' level.

Keywords: ventilative cooling, seminar room, thermal comfort, thermal storage, phase change materials, operational data.

Thermal comfort and indoor air quality requirements for the case-study

Thermal comfort evaluation is usually based on current guidance on avoiding overheating in buildings. In the UK, current guidance for schools is provided by the Education Funding Agency [1]; it includes guidelines on ventilation, thermal comfort and indoor air quality, including the Services Output Specification [2], the Baseline Design Environmental Services and Ventilation Strategy [3] and the Building Bulletin 101 [4]. These documents are aligned with CIBSE's guidance on prevention of summertime overheating [5,6 ,7 ,8] which refer to calculations according to European Standard BS EN 15251 and UK Building Regulations Parts L (Conservation of Fuel and Power) and F (Ventilation) [9].

Until recently overheating criteria for schools were based on fixed air temperature (28°C which can be exceeded for 120 hrs and 32°C not to be exceeded) outside the heating season and during the occupied period from 1st May to 30th September.

Currently, the adaptive thermal comfort approach is used which follows the methodology and recommendations of European Standard EN 15251 to determine whether a building is overheated, or in the case of an existing building whether it can be classed as overheating. The new criteria are based on a variable (adaptive) temperature threshold that is generated from the outside running-mean dry-bulb temperature. There are three criteria, two of which must be met for compliance, as follows [3]:

(a) Hours of Exceedence: The number of hours operative temperature exceeds the maximum acceptable operative temperature (θmax) by 1K, must not exceed 3% of the total occupied hours or 40 hours, during the five summer months.

(b) Weighted Exceedance: The sum of the weighted exceedance for each degree K above θmax (1K, 2K and 3K) is ≤ 10.0.
(c) Threshold/Upper Limit Temperature (θupp): The measured/predicted operative temperature should not exceed the θmax by 4K or more at any time.

The case-study analysed in this paper was built to comply with the older requirements so operational data are analysed following both approaches.

In terms of IAQ based on CO_2 concentration, until recently the guidance was that when measured at seated head height, during the continuous period between the start and finish of teaching on any day, the average concentration of carbon dioxide should not exceed 1,500 parts per million (ppm). This criterion is changed to the following criteria [3]:

(a) Ventilation should be provided to limit the concentration of carbon dioxide measured at seated head height in all teaching and learning spaces.
(b) Where mechanical ventilation is used or when hybrid systems are operating in mechanical mode, i.e. the driving force is provided by a fan, sufficient fresh air should be provided to achieve a daily average concentration of carbon dioxide during the occupied period of less than 1,000 ppm and so that the maximum concentration does not exceed 1,500ppm for more than 20 consecutive minutes each day.

Description of the case-study computer seminar room

The case-study is a seminar room at a university campus in West England. Cool-Phase® systems have been installed in other spaces of the university but the seminar room was chosen because of its use (computer laboratory) with higher internal heat gains than other spaces. The room has a floor area of 117 m² and includes 26 desk top computers, peak occupancy of 26 students, and artificial lighting comprising of 24 luminaires each equipped with one 48 W lamp. The total internal heat gain in the room is 60 W/m². The room has one external wall facing west with U-value of 0.56 W/m² K while 23 % is glazing (U-value 1.82 W/m² K) with internal blinds. Ventilation and cooling is provided via a 8 kW Cool-Phase® unit. Heating is provided through perimeter hot water radiators and windows are operable. Climate is temperate maritime with 2,684 Heating Degree Days and 196 Cooling Degree Days; 20 year average, base 15.5°C, south west England [10].

Description of the ventilative cooling system

A Cool-Phase® system by Monodraught Ltd was installed in May 2013 to provide ventilation for indoor air quality and cool the air for thermal comfort. The Cool-Phase® system uses the concept of a thermal battery consisting of Phase Change Material (PCM) plates within the ventilation path to capture and store heat. Therefore, the thermal batteries use the latent heat property of materials to store energy, which is charged and discharged by passing air through a heat exchanger. A diagram of the system is shown in **Figure 1** where the principle of the PCM thermal battery function is shown. The system is concealed in the false ceiling and its appearance to the user is that of a conventional ventilation system with two air supply terminals and one air extract terminal. Air is drawn from outside or the room using a variable speed fan. During operational hours and depending on internal air quality (monitored through CO_2 sensors) the air is mixed with recirculated air from the room to conserve energy. The air is then directed through the PCM thermal battery to be cooled if necessary (determined by air temperature sensors and control rules) or by-passes it if cooling is not needed. Outside operational hours,

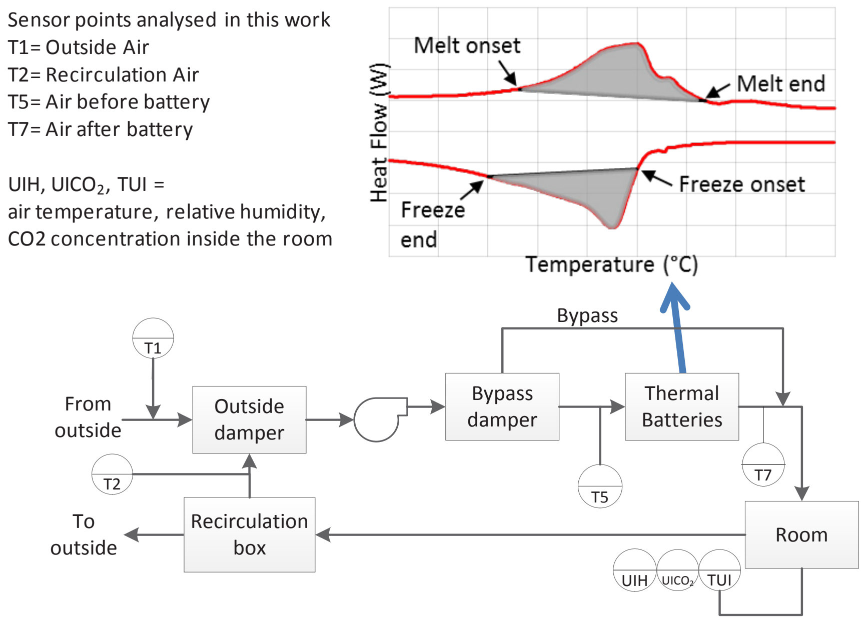

Figure 1. Schematic of Cool-Phase system with a graphical explanation of the PCM thermal battery principle of operation.

ambient air is used to recharge the PCM thermal battery the duration of which is determined by air temperature sensors and control rules according to the season.

Figure 2 shows how the system works based on monitored data during one day in August 2013. The system starts with a charging-purge mode between midnight and 1:00 and continues with charging mode from 1:00 to 7:00 am. Inlet and outlet temperatures through the PCM thermal battery are decreasing with a temperature difference between them indicating the battery is charging. The system is off between 7:00 and 8:00 am when the cooling mode is initiated and continues until 21:00. In the morning (8:00-~13:00) the temperature outside the intake damper is lower than the set-point for summer (22°C) so the PCM thermal battery is by-passed. At around 13:00, set-point temperature is exceeded and the inlet air is directed to the PCM thermal battery through recirculation. Inlet air is cooled to below room temperature until shortly before 21:00 when the system is off until midnight. Maximum temperature in the room is 24.5°C below max external temperature.

System performance

Figure 3 shows temperatures in the case-study room during operational hours in the summer of 2013 (May – September). According to adaptive thermal comfort criteria, it can be observed that the system has achieved internal temperatures within the upper and lower limits and therefore complies with all conditions. Also, air temperatures do not exceed 28 or 32°C and daily average inside/outside temperature difference is less than 5°C and therefore achieves comfort according to static thermal comfort criteria.

An analysis of monitored room CO_2 concentration was carried out for the whole year that data are available. **Table 1** presents the results. Daily average concentration during the occupied period is always less than 1.000 ppm and the 1.500 ppm limit was not exceeded with the exception of one occasion for 22 min when occupancy was higher than designed and there was a conflict between IAQ and thermal comfort.

The fan energy used by the system for the year was calculated to be 90 kWh. This equates to 0.77 kWh/m²/annum. Annual electricity energy use intensity for secondary schools has a median of 51 kWh/m² [8]. This increases by 5 kWh/m² when moving from 'heating and natural ventilation' to 'heating and mechanical ventilation' buildings. CIBSE TM57 [8] presents good case-studies with cooling energy intensity of 12.5 kWh and 3.5 kWh/m².

Room temperature and air velocity distribution using CFD

In the previous section average environmental conditions in the room were reported. However, the distribution is also important to examine whether there are areas within the room that deviate from thermal comfort requirements. This was investigated using a CDF model of the room. A 3D model of the room was constructed with summer boundary conditions; the hour in July with the highest internal temperature was selected as the worst case scenario and a steady state simulation was performed with full occupancy and internal heat gains. **Figure 4** shows the air temperature at 1.2 m height (student sitting plane) and velocity fields at the plane of one air inlet.

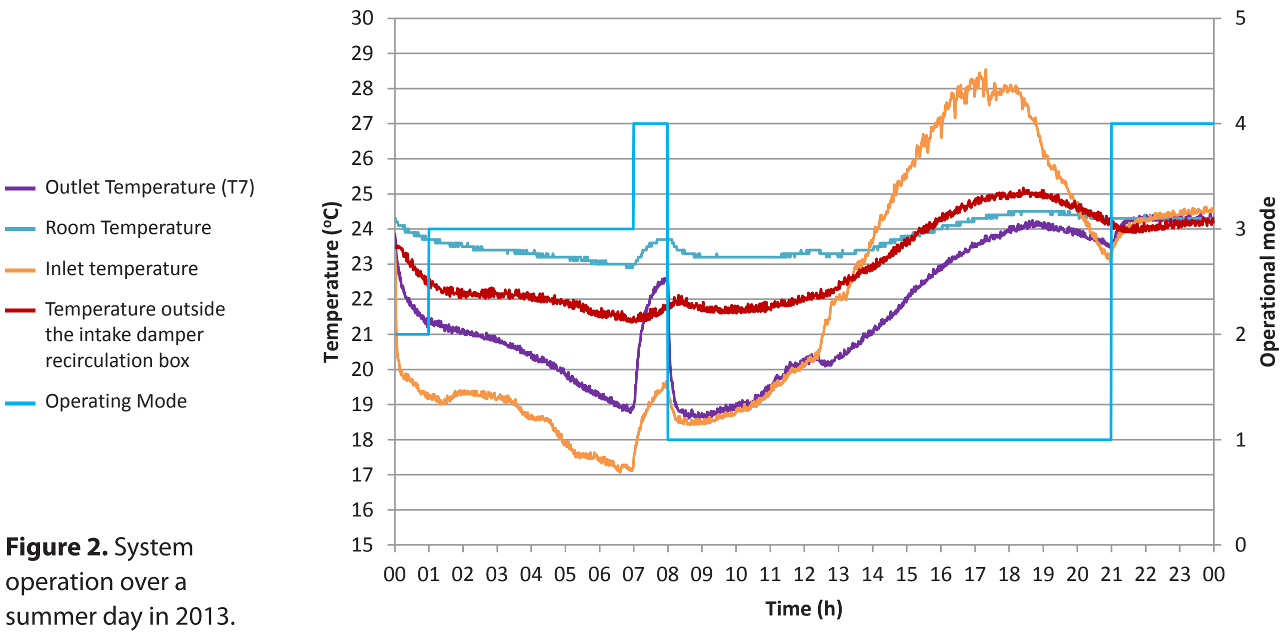

Figure 2. System operation over a summer day in 2013.

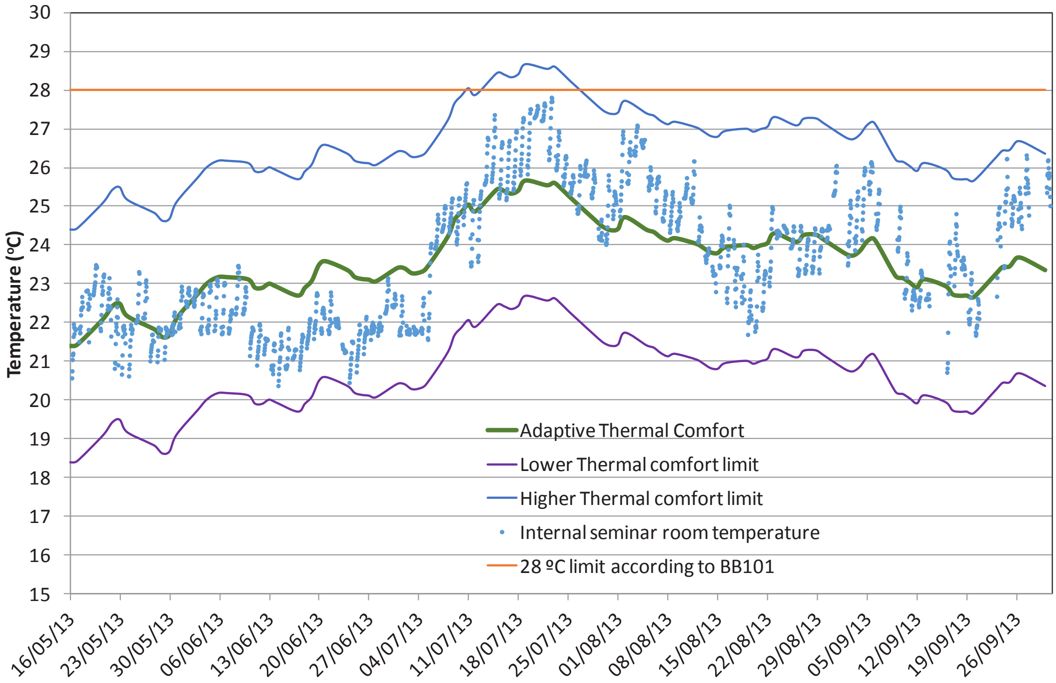

Figure 3. Thermal comfort performance over the summer months.

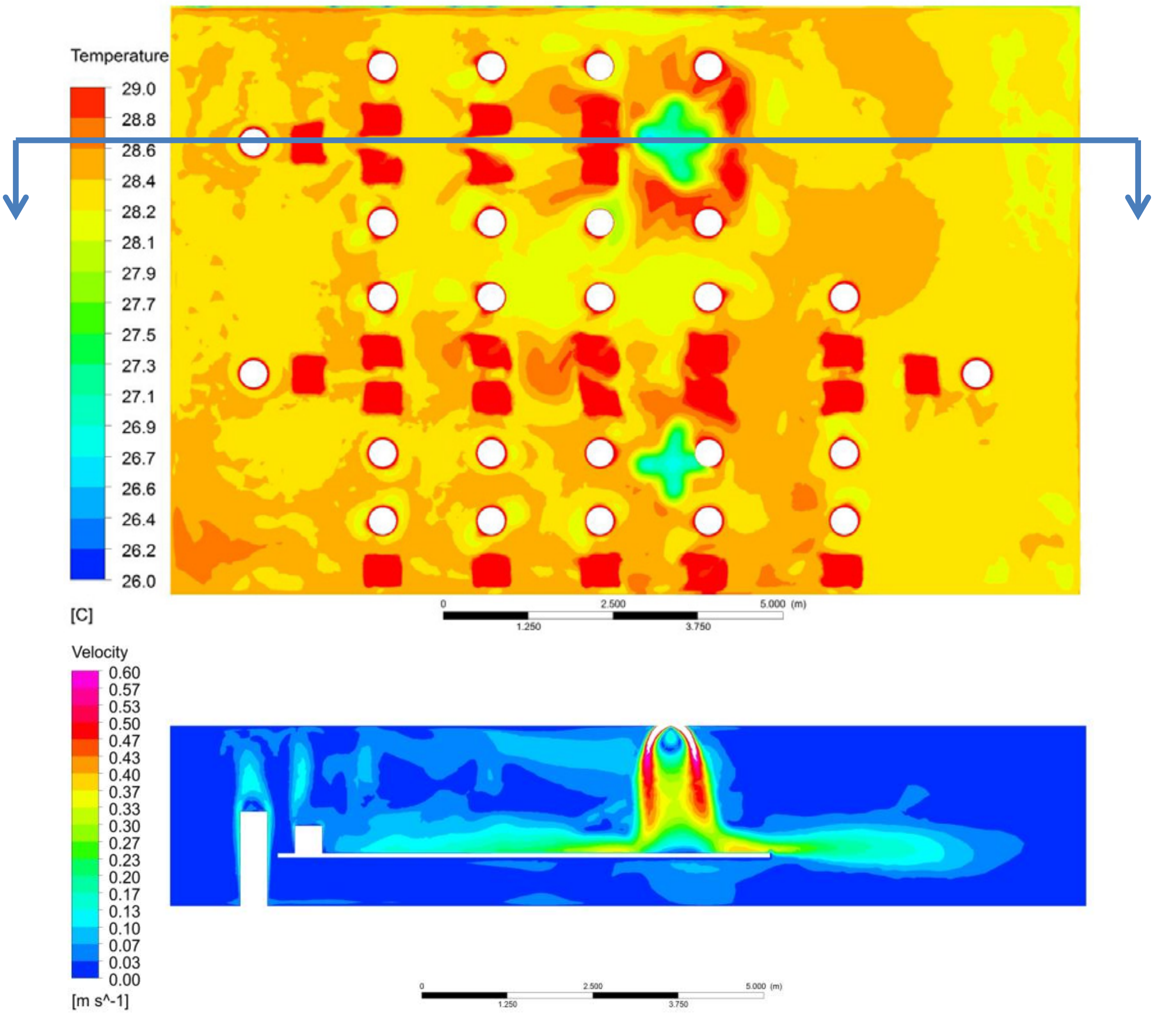

Figure 4. Air temperature and velocity in two sections of the seminar room during the hour with highest internal temperature (see **Figure 3**) and full occupancy and internal gains.

It can be observed that air temperature is uniform across the room and there are no areas with much higher air temperature which will cause discomfort. The air velocity contours indicate that at occupancy level underneath the air inlet velocity is in the range of 0.1–0.2 m/s with some small areas reaching 0.37 m/s Air velocity is lower in the rest of the room. Changing the direction of inlet louvres would reduce air velocities if this is required although higher velocities might aid thermal comfort.

Concluding remarks

Analysis of one-year operational environmental data for a seminar room equipped with a Cool-Phase® system to provide cooling indicate that the system performs well throughout the year in terms of IAQ and thermal comfort for an IT intensive seminar room. Further analysis of a second year of operational data plus additional monitoring to study the distribution of environmental conditions in the room and feedback by users is under progress and will be reported in a case-study being developed for EBC Annex 62. ■

Table 1. CO_2 concentration (ppm): daily average and exceeding 1500 ppm for more than 20 consecutive minutes.

Month	Average	> 1,500 ppm
May	502	0
June	423	0
July	413	0
August	416	0
September	500	0
October	595	0
November	741	0
December	566	Once*
January	601	0
February	719	0
March	695	0
April	579	0

* CO_2 concentration exceeded 1,500 ppm for 22 min on mid-morning on 6 Dec 2013 when occupancy in the room was more than its maximum and external air temperature at ~7°C. The control system restricted outside air to the room to less than maximum capacity to avoid thermal comfort issues.

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Control of indoor air quality by demand controlled ventilation

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This article discusses the advantages of using carbon dioxide concentration as a measure of indoor air quality and the possibility of improving it by choosing the optimal air distribution scheme and reducing energy consumption by indoor ventilation systems, which also reduces the emission of carbon dioxide into the atmosphere

Keywords: Air quality, health, carbon dioxide, ventilation, air exchange, energy efficiency, energy saving.

Environmentalists, physicians and diagnosticians as well as engineers and designers of ventilation and air conditioning systems all pay special attention to the influence of indoor air quality on human well-being. A person's physical condition depends on air quality; where it is unsatisfactory, people feel unwell, lose concentration, develop diseases, etc.

All kinds of pollutants may be released into indoor air and affect its quality (carbon dioxide released by humans; phenol/formaldehyde, acetone, ammonia and other components released by furniture and decoration materials). Both Russian and international experts have done a lot of studies [1, 2, 3, 4] that led to the adoption of carbon dioxide concentration as an indicator of indoor air pollution. In 2011, Russian standard GOST 30494 was amended to include this [5].

Highlights

- The human health depends on the indoor air quality.
- The carbon dioxide concentration is an indicator of the indoor air pollution.
- The indoor air quality was considered with different schemes of the air distribution.
- DCV systems – optimal indoor air quality and low power consumption.

Air quality is a key component of a healthy microclimate at the workplace.

The human breathing process under normal conditions mainly alters the concentration of two air components, oxygen and carbon dioxide. The metabolic processes in the human body reduce oxygen content in exhaled air from 20.9% to 16.3%, while increasing carbon dioxide concentration from 0.03% to 4% [6]. It should be noted that carbon dioxide concentration increases more than a hundred times. Both Russian and international experts have done a lot of studies [1, 2, 3, 4] that led to the adoption of carbon dioxide concentration as an indicator of indoor air pollution. Other hazardous gas emissions into the air of residential and public buildings (phenol/formaldehyde, acetone, ammonia and other components released from furniture and decoration materials) are converted into carbon dioxide equivalents [7].

General

GOST 30494-2011 'Residential and Public Buildings. Microclimate Parameters for Indoor Enclosures' [5], developed with the participation of the authors of this article, includes four indoor air quality classes depending on the concentration of carbon dioxide:

- Class 1 (optimal microclimate, high quality) – carbon dioxide level not higher than 400 ppm;
- Class 2 (optimal microclimate, medium quality) – carbon dioxide level between 401 and 600 ppm;
- Class 3 (acceptable microclimate, acceptable quality) – carbon dioxide level between 601 and 1000 ppm;
- Class 4 (unacceptably high carbon dioxide level, low air quality) – more than 1000 ppm.

The advantages of this approach to assessing air quality and the air exchange requirement over the traditional one (based on the relative blowing rate or air exchange rate) are as follows:

- air exchange calculations can take into account outdoor air pollution;
- higher ventilation efficiency is promoted: fresh air supply into the breathing area, no fresh air streams blowing across 'dirty' zones on the premises, etc.;
- the fresh air in the room can be taken into account before the room is filled by people;
- 'background' air exchange for removing hazardous emissions of furniture and decoration materials at non-working hours can be determined correctly;
- control of air quality becomes more adequate and accurate due to measuring carbon dioxide concentration directly in the room area serviced.

Information on carbon dioxide concentration in outdoor air is provided by weather observation stations. For reference: according to [5], approximate average annual values of carbon dioxide concentrations are:

- in the countryside, 350 ppm;
- in small towns, 375 ppm;
- the polluted center of a big city, 400 ppm.

The air exchange rate for the most widespread 'mixing' ventilation system is calculated from the formula:

$$L = 55 \cdot 10^4 \frac{G}{g_{out} - g_n} \quad \text{m}^3/\text{h} \tag{1}$$

where G is the amount of carbon dioxide entering the enclosure, g/h;
g_n and g_{out} are the normative and outdoor carbon dioxide concentrations, respectively, ppm.

Mixing ventilation is supposed to spread air evenly across the room, and the concentration of pollutants, including carbon dioxide, is expected to be the same everywhere (**Figure 1, A**). Mixing ventilation usually features a high air exchange rate, at least 3 1/h.

Mixing ventilation systems include air recycling systems and those combined with fan terminals of air conditioning systems (split systems and fancoils).

In many public and office buildings, false ceilings are used to house both air supply and exhaust devices. In traditional solutions, air exchange rates usually do not exceed 1 – 1.5 1/h. In some cases of isothermal ventilation or slightly overheated incoming air, a large share of fresh air is drawn into the exhaust grids, forming what is called 'short circuit' circulation (**Figure 1, B**). This is an example of inefficient organization of ventilation.

An example of efficient ventilation is 'displacement' ventilation [8, 9]. Fresh incoming area is supplied into the serviced area at a small velocity through air diffusers with a large surface area to effectively 'flood' it. Polluted air, lifted by convective flows from occupants and office and other equipment, will be displaced into the upper tier and then exhausted (**Figure 1, C**). In this case, concentration of carbon dioxide in the serviced area may be lower than in the air removed.

Formally, in all the three cases (**Figure 1**) the same air exchange rate may be adopted under the traditional design approach, but the resultant air quality will differ widely.

The air volume required for ventilating the premises should be calculated according to [5] taking into account the air distribution efficiency factor:

$$L = \eta \cdot L_b \quad \text{m}^3/\text{h} \tag{2}$$

where L_b is the base amount of external air according to the current Russian norms, m³/h.

The value of the air distribution efficiency factor is shown in **Table 1**.

Thus, if the statutory concentration of carbon dioxide is 800 ppm, and in the outdoor air its content is 400 ppm, for a workplace in an office building where a person exhales 45 g of carbon dioxide per hour (a quantity adopted according to [10] for adult brainworkers), the flow of external air in the ventilation system can be calculated from the formula (1):

$$L = 55 \cdot 10^4 \cdot \frac{45 \cdot 10^{-3}}{800 - 400} = 61.875 \text{ m}^3/\text{h} \approx 60 \text{ m}^3/\text{h}$$

This is the exact volume of air per workplace that the mixing ventilation system must supply to the premises. A 'short circuit' system will need more, 66 to 78 m³/h in the light of **Table 1**, while 'displacement' ventilation will permit a lower air exchange rate, 36 to 48 m³/h, and personal ventilation, 18 to 30 m³/h.

In other words, with air quality being the same, the air exchange rate and, consequently, energy consumption (on air transportation through ducts and heating/cooling) may differ 1.5 to 2 times.

The distribution of carbon dioxide concentration fields across the volume of premises can be calculated accurately enough. Still, in most cases the air and heat regime modeling effort is made for unique facilities only [11]. **Figure 2** shows approximate carbon dioxide distribution patterns for displacement ventilation (A) and in the vicinity of a fresh air stream (B) based on the calculation assumptions [17].

Table 1. Air Distribution Efficiency Factors.

S/N	Ventilation Systems	Air Distribution Efficiency Factor
1.	Mixing ventilation systems with air exchange rates higher than 2.5 1/h, including those using recycling, split systems and fancoils	1.0
2.	Isothermic ventilation systems or those combined with air heating that have a 'top to top' air distribution system and air exchange rates not exceeding 1.5 1/h	1.1 – 1.3
3.	Displacement ventilation systems	0.6 – 0.8
4.	Personal ventilation systems supplying fresh air into the breathing area	0.3 – 0.5

The efficiency of ventilation systems can also be characterized by the lifetime of fresh air – the time that air flowing from the air distributor takes to reach the breathing area. In personal ventilation system it takes less than a second; in 'displacement' systems, 20 to

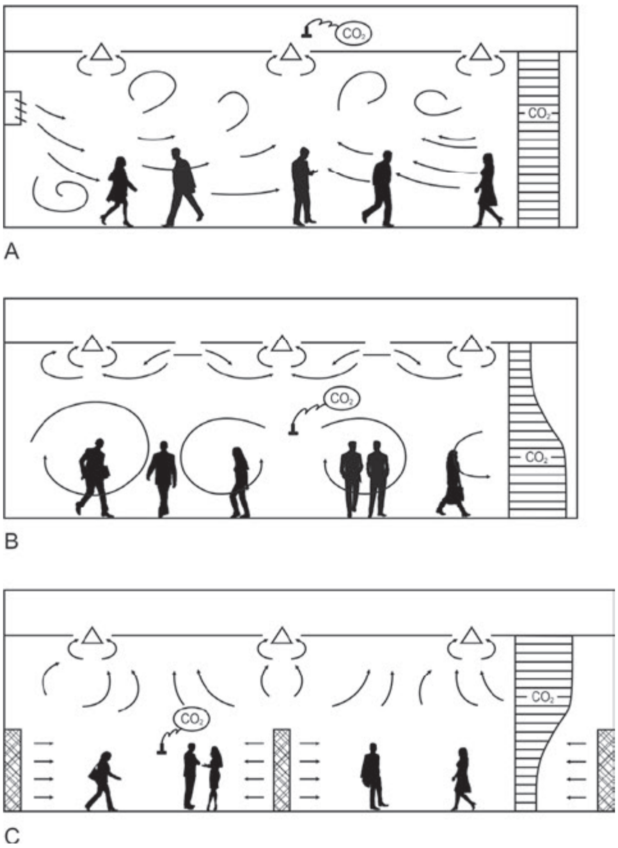

Figure 1. Carbon Dioxide Distribution Pattern with Mixing (A), Short-Circuit (B) and Displacement (C) Ventilation Installed.

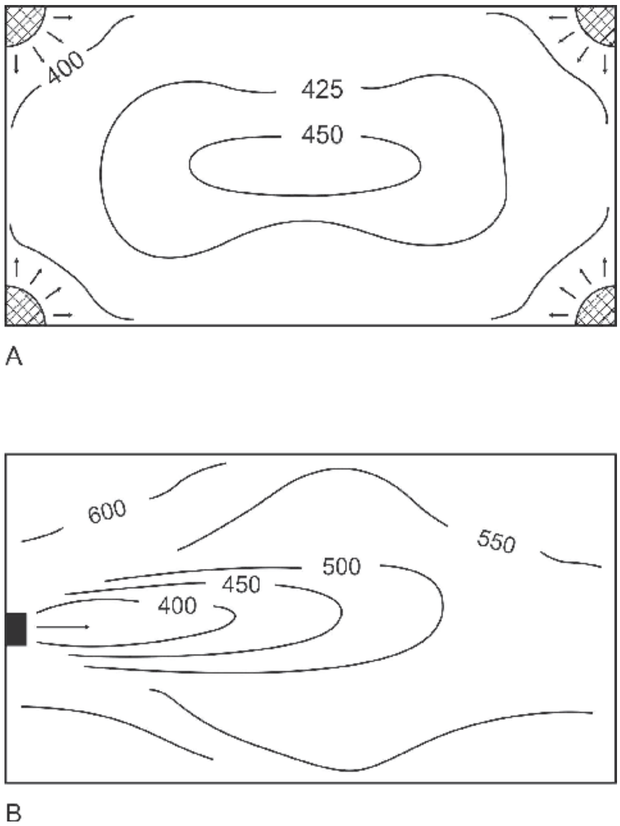

Figure 2. Lines of Equal Carbon Dioxide Concentrations on a Room Plan with Displacement Ventilation Installed (A) and in a Stream of Incoming Fresh Air (B).

30 seconds, and in 'short circuit' systems, up to ten minutes.

The efficiency retention of the air distribution system can thus be considered the criteria of the adaptability of ventilation systems (DCV systems). Demand Controlled Ventilation (DCV) stands for a special type of variable air velocity (VAV) ventilation systems that permit wide-range control of air exchange in individual areas and at different times depending on the actual occupancy of the premises [12, 13, 14, 15,16].

Another adaptability criterion should be the correspondence between the amount of the pollutants released (in this case, carbon dioxide) and the air exchange rate.

Traditional ventilation systems are designed for the rated occupancy of the premises and cannot adjust air exchange.

E.g., if the standard staff number in an office is 1,000 persons, the system will keep supplying and exhausting 60,000 m³ of air per hour. On the other hand, if holidays, sick leaves and business trips are taken into account, the actual number of personnel in the office will be just 70% of the rated figure, or fewer. Moreover, even if the business has fixed office hours, the first employees will come an hour or two earlier, and the last ones will leave three or four hours later than required to.

A traditional ventilation system will thus operate in its design mode since the first employees arrive and until the last leave.

Plotted in **Figure 3** are the working cycles of a traditional ventilation system with a constant air exchange rate and of demand-controlled ventilation depending on the number of personnel present at the office. The hatched area on the plot represents the power and air saving in the demand-controlled ventilation system than can reach 40 to 50%.

Air exchange control in a demand-controlled ventilation system can be governed by carbon dioxide levels measured by a special sensor. Following the sensor's signal, regulated gates will adjust the flow of air entering the premises. The signal is then forwarded to the

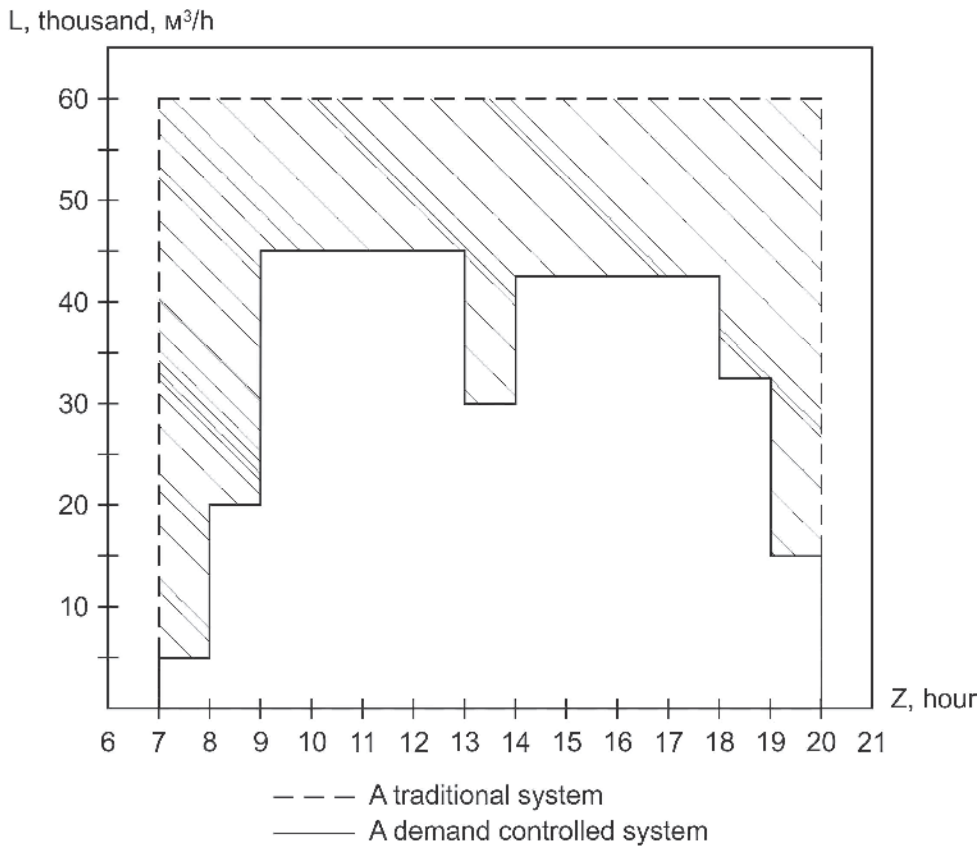

Figure 3. A Ventilation System Operation Schedule.

air-supply unit and the exhaust unit equipped with variable-frequency drives for adjusting fan delivery.

The place where the carbon dioxide level sensor is installed is important. In a mixing ventilation system, the sensor may be installed in the exhaust air manifold, and in other cases, in the serviced area or breathing area (**Figure 1**).

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Conclusions

1. The concentration of carbon dioxide can serve as an indicator of air quality in residential and public buildings.
2. The efficiency retention of air distribution is an important adaptability criterion to be used in selecting ventilation systems. The target should be for fresh air to reach the breathing area by a short trajectory, without crossing 'dirty' zones where hazardous substances are released.
3. It is important to make the fresh air inflow match the number of people on the premises. While a high air quality is maintained in buildings with variable numbers of personnel or visitors (such as railway stations, airports, trade centers, sports or recreational facilities, offices), demand controlled ventilation can save 40 to 50% of energy as compared to traditional ventilation systems. ■

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ISO 11855 - The international standard on the design, dimensioning, installation and control of embedded radiant heating and cooling systems

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This article introduces the structure and contents of ISO 11855 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems. This is the first international standards on radiant heating and cooling system, published in 2012. ISO 11855 includes the processes and conditions needed to determine the heating and cooling capacity of radiant heating and cooling systems in new construction and the retrofit of existing buildings. In addition, the standard stipulated the design conditions regarding components such as the heat supply, hydronic distribution systems, panels, and control systems of radiant heating and cooling systems.

Keywords: Radiant heating and cooling system for occupants' comfort, Determination of the design heating and cooling capacity, Design and dimensioning, Thermo Active Building Systems (TABS), Installation, Control.

Structure of ISO 11855

In recent years, radiant heating and cooling systems have seen considerable use not only in residential buildings but also in commercial buildings because of their many advantages including higher comfort level, lower noise level and greater possibility of integration to the architectural design in comparison with other HVAC systems. In particular, these systems have gained attention as heating and cooling systems with a high possibility of utilizing renewable energy sources because they can provide heating at a low temperature of hot water and cooling at a high temperature of cold water. Nowadays many countries in Europe, USA and Asia have developed and applied separate components such as pipes, prefabricated panels, manifolds and controller to new and existing building construction. Generally a few product standards had been adopted for the international trade of related products at that time. For that reason, CEN had been working on developing related standards for determining the heating and cooling capacity and finally developed EN 15377 (refer to **Figure 1**). However, there was no single and comprehensive international standard for the design of radiant heating and cooling system. Therefore it was impera-

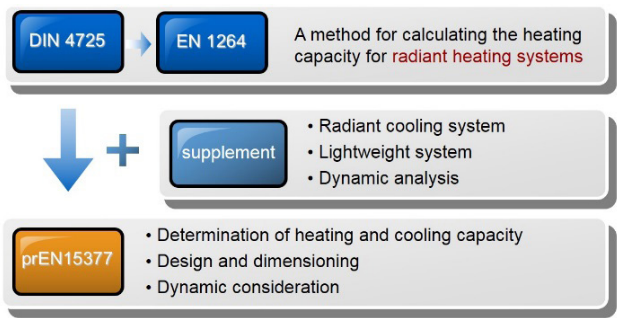


Figure 1. Structure of European standards about radiant heating and cooling system design.

tive that design standards that maximize the advantages of radiant heating and cooling systems should be developed. So the ISO Standards need to deal with the determination of the heating and cooling capacity, system construction, commissioning and operation all together. And this work could be done by supplementing above mentioned EN 15377. Furthermore, for the development of new ISO Standards, some regional characteristics needed to be considered. For example, some particular floor structures used in each country should be categorized to include the all types of radiant heating and cooling system in use. And different comfort criteria which has been preferred traditionally (e.g. maximum floor surface temperature) should be considered accordingly during design process.

The scope of ISO 11855 is the radiant heating and cooling systems that perform heating and cooling in new construction and the retrofit of existing buildings. Most of HVAC systems undergoes the life cycle as shown in **Figure 2**. To ensure the system performance, technical standards should be applied to each step of the life cycle.

Figure 2. Life cycle of general HVAC system design.

To establish the basic design principles, the design process and conditions were firstly addressed. So it was imperative that the established systems should be able to perform in accordance with occupants' comfort. And the process regarding the determination of the heating and cooling capacity and system construction (heat supply, hydronic distribution systems, panels and control systems) must be provided to make it possible to secure the same performance even when the designers are different. As for the performance of radiant heating and cooling systems, energy consumption throughout the building's life cycle must be taken into consideration. ISO Standards for determining the energy performance of radiant heating and cooling systems should be developed by providing methods for conducting dynamic analysis during design process. It is also imperative that standards be developed to make possible energy reduction, prolongation of the life cycle of heating and cooling equipment, and operation of radiant heating and cooling systems according to their design.

After the review of current EN Standards and discussions on references and the world-wide experts' opinions, 8 parts of ISO Standard structure was firstly proposed as shown in **Figure 3**. At the development stage, these standards were assigned to ISO 11855 series, and the structure of standards was changed into 6 parts accordingly. Experts agreed that ISO 11855 would deal with the embedded surface heating and cooling system that directly controls heat exchange within the space, and does not include the system equipment itself, such as heat source, distribution system and controller. The ISO 11855 series addresses an embedded system that is integrated with the building structure. Therefore, the panel system with open air gap, which is not integrated with the building structure, is not covered by this series.

The objective of the ISO 11855 series is to provide criteria to effectively design embedded systems. To do this, it presents comfort criteria for the space served by embedded systems, heat output calculation, dimensioning, dynamic analysis, installation, operation, and control method of embedded systems.

Part 1 of these standards specifies the comfort criteria which should be considered in designing embedded radiant heating and cooling systems, since the main objective of the radiant heating and cooling system is to satisfy thermal comfort of the occupants. Part 2 provides steady-state calculation methods for determination of the heating and cooling capacity. Part 3 specifies design and dimensioning methods of radiant heating and cooling systems to ensure the heating and cooling capacity. Part 4 provides dimensioning and calculation method to design TABS (Thermo Active Building Systems) for energy-saving purposes, since radiant heating and cooling systems can reduce energy consumption and heat source size by using renewable energy. Part 5 addresses the installation process for the system to operate as intended. Part 6 shows proper control methods of the radiant heating and cooling systems to ensure the maximum performance when the system is being actually operated in a building.

Radiant heating and cooling system for occupants' comfort

Occupant's thermal comfort would be the primary objective that any HVAC system pursues. Radiant heating and cooling systems can be used as primary or hybrid systems which are combined with an air system and provide unique and cost-effective approaches dealing with numerous conditions affecting human

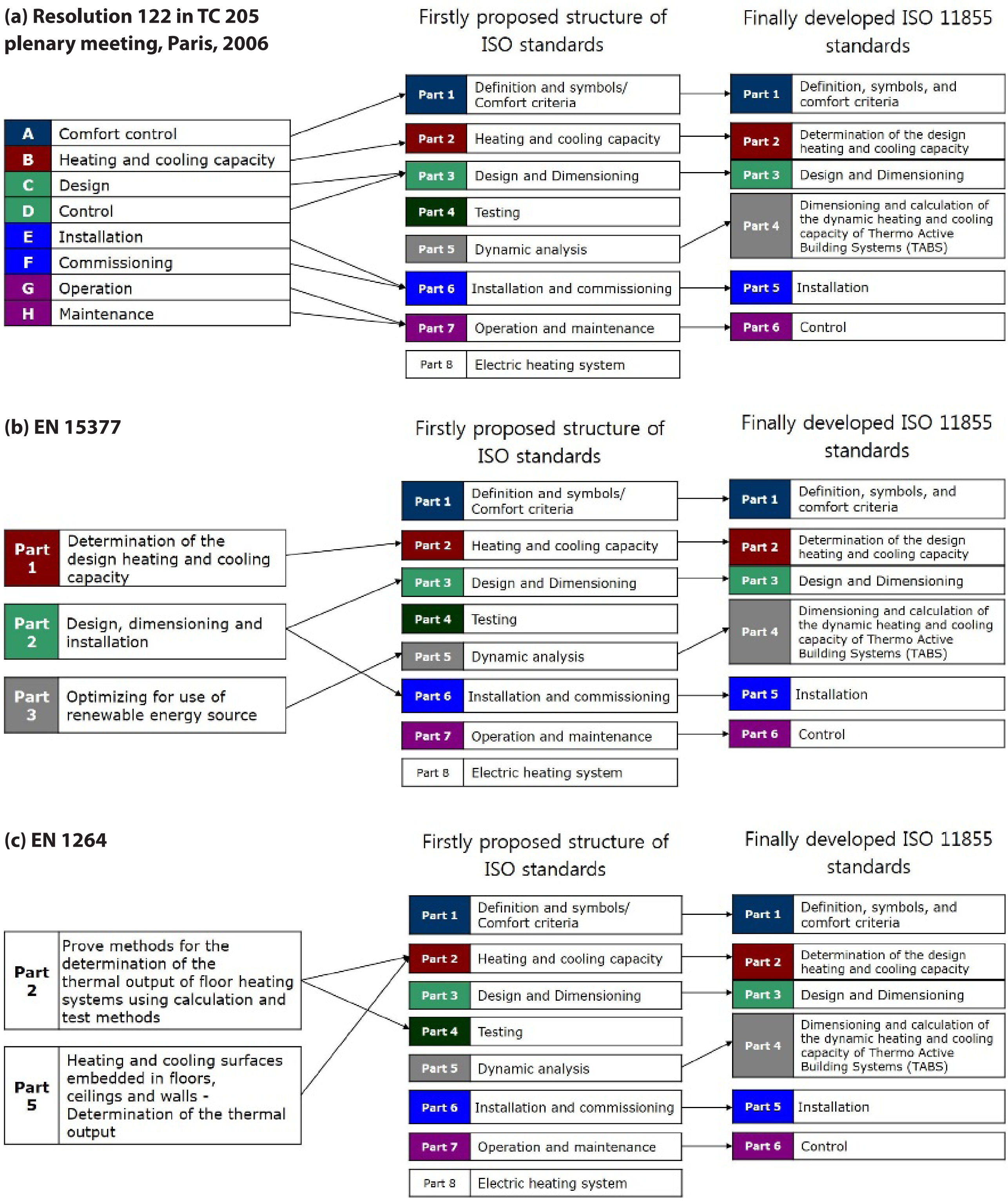
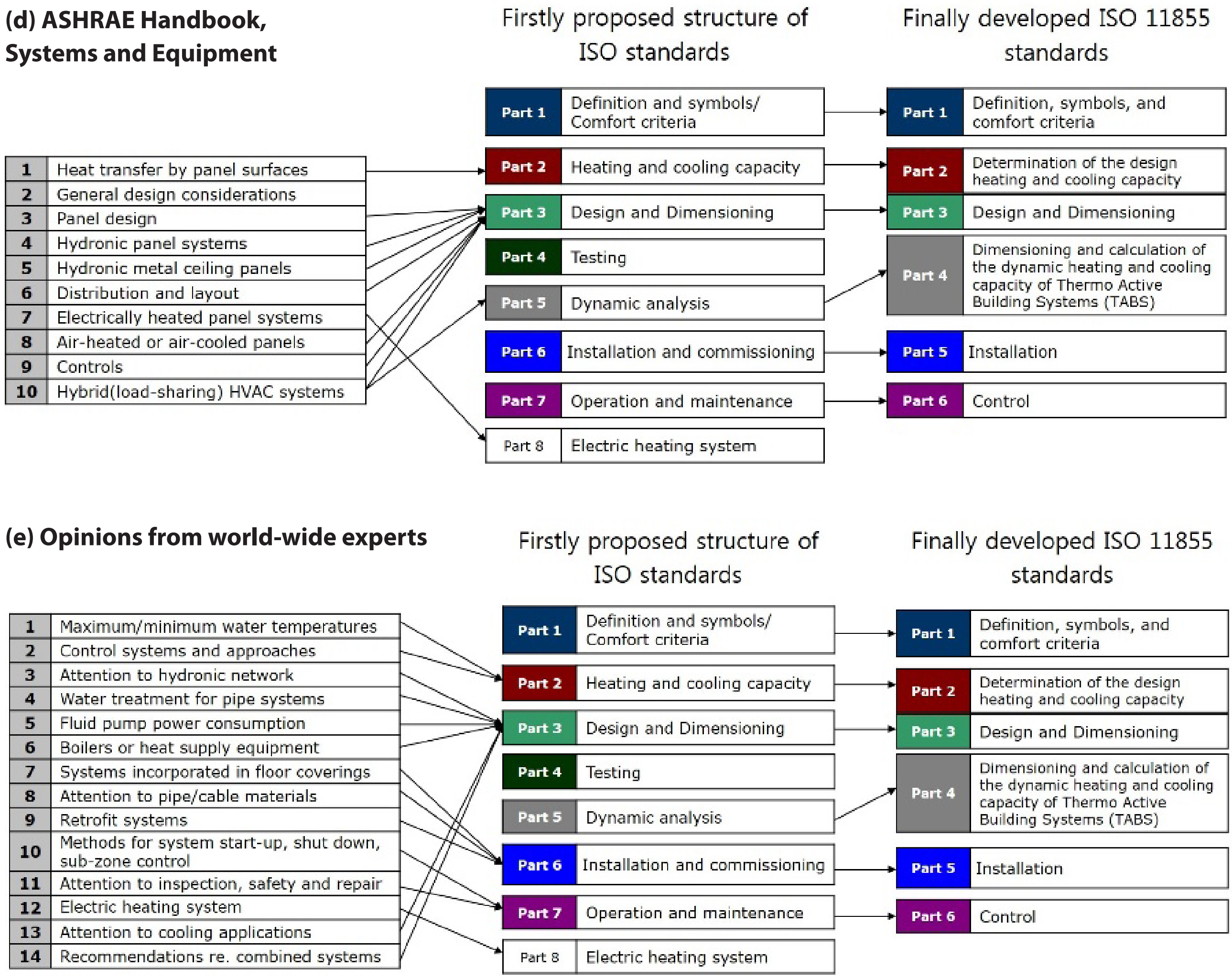


Figure 3. Basic structure of ISO 11855 considering the current EN Standards and other references.

thermal comfort. Radiant heating and cooling systems can be used to directly provide heat to humans as well as to spaces. As long as the occupants are radiantly heated in a radiant heating system, the same comfort level can be maintained with a lower air temperature in comparison to a convective heating system. For radiant cooling systems, maintaining the same comfort level with a higher air temperature in comparison to convective cooling is possible. Therefore, compared with conventional heating and cooling systems, it is possible to reduce the energy loss due to ventilation, and infiltration while maintaining the same comfort level.



Thermal comfort can be defined as the psychological condition that expresses satisfaction with the thermal environment. Therefore, thermal comfort would be evaluated by asking all the occupants if they are satisfied with their thermal environment. However, in order to design and control radiant heating and cooling systems, it is necessary to predict the thermal comfort in a room without resorting to a polling result. To provide an acceptable thermal environment to the occupants, the requirements for general thermal comfort, e.g. predicted mean vote (PMV), operative temperature (OT), and local thermal comfort (surface temperature, vertical air temperature differences, radiant temperature asymmetry, draft, etc.) shall be taken into account. In radiant systems, floor, walls and ceilings can be used as the heat transfer surface for heating and cooling. For this reason, special care shall be paid to the surface temperature limit of the floor and wall with which the occupants can have direct contact.

The floor temperature has a direct impact on the comfort of the feet or buttocks. In ISO 7730, the floor temperature range of 19°C to 29°C is recommended in the space with sedentary and/or standing occupants wearing normal shoes. This is a limiting factor when deciding the capacity of floor heating and cooling systems. For heating, the maximum temperature is 29°C and for cooling, the minimum temperature is 19°C. However, this temperature range of 19°C to 29°C might be changed by the factor of whether the occupants wear shoes or not, or whether they usually sit on the floor or stand up in the occupied zone. Thus, the range of the surface temperature can be different depending on lifestyle habits. For this reason, it is recommended to follow the widely accepted standards of each country when deciding on the optimum range of floor surface temperature. For an electric heating system, an electrically-heated floor may cause discomfort and even skin burn if occupants have prolonged contact with the floor. This is due to the constant supply of heat from an electrical heating source, whereas, for a water based heating system, the increase in surface temperature is limited by the water temperature. Therefore it is important to control the electrical heating source in order to keep the floor surface temperature under the lower limit of discomfort and skin burn. For wall heating, the maximum recommended surface temperature is in the range of 35°C to 50°C. The maximum temperature depends on factors such as whether occupants may easily have contact with the surface or whether buildings are used for more sensitive persons such as children or the elderly. When a skin temperature is 42°C to 45°C, there is a risk of burns and pain. The losses to the rear walls and its influence on neighboring spaces should be taken into account.

Determination of the design heating and cooling capacity

ISO 11855-2 specifies procedures and conditions to enable the heat flow in water based surface heating and cooling systems to be determined relative to the medium differential temperature for systems. The determination of thermal performance of water based surface heating and cooling systems and their conformity to this part of ISO 11855 are carried out by calculation in accordance with design documents and the model. This should enable a uniform assessment and calculation of water based surface heating and cooling systems. The surface temperature and the temperature uniformity of the heated/cooled surface, nominal heat flow density between water and space, the associated nominal medium differential temperature, and the field of characteristic curves for the relationship between heat flow density and the determining variables are given as the result. Based on the calculated average surface temperature at given combinations of medium (water) temperature and space temperature, it is possible to determine the steady state heating and cooling capacity.

ISO 11855-2 includes a general method based on Finite Difference or Finite Element Methods and simplified calculation methods depending on position of pipes and type of building structure. Two types of simplified calculation methods can be applied according to ISO 11855-2. One method is based on a single power function product of all relevant parameters developed from the finite element method (FEM), and another method is based on calculation of equivalent thermal resistance between the heating or cooling medium temperature and the surface temperature (or room temperature). A given system construction can only be calculated with one of the simplified methods. The correct method to apply depends on the type of system, A to G (depending on position of pipes, concrete or wooden construction) and the boundary conditions.

The ISO 11855 series is applicable to water based embedded surface heating and cooling systems in residential, commercial and industrial buildings. The methods apply to systems integrated into the wall, floor or ceiling construction without any open air gaps. It does not apply to panel systems with open air gaps which are not integrated into the building structure. The ISO 11855 series also applies, as appropriate, to the use of fluids other than water as a heating or cooling medium. The ISO 11855 series is not applicable for testing of systems. The methods do not apply to heated or chilled ceiling panels or beams.

Design and dimensioning

ISO 11855-3 introduces the design and dimensioning of floor heating, ceiling heating, wall heating, floor cooling, ceiling cooling and wall cooling respectively. Basically the design and dimensioning methods for radiant floor heating and cooling were described. And wall and ceiling radiant heating and cooling can also be applied to the same procedure except for determination of limit curves because of physiological limitations concerning the surface temperatures of ceiling heating systems.

Floor heating system design requires determining heating surface area, type, pipe size, pipe spacing, supply temperature of the heating medium, and design heating medium flow rate. The design steps for floor heating system are as follows:

- Step 1: Calculate the design heating load Q_N. The design heating load Q_N shall not include the adjacent heat losses. This step should be conducted in accordance with standards for heating load calculation, e.g. EN 12831, based on an index such as operative temperature (OT) (see ISO 11855-1).
- Step 2: Determine the area of the heating surface A_F, excluding any area covered by immovable objects or objects fixed to the building structure.
- Step 3: Establish a maximum permissible surface temperature in accordance with ISO 11855-1.
- Step 4: Determine the design heat flux q_{des}. For floor heating systems including a peripheral area, the design heat flux of peripheral area $q_{des,R}$ and the design heat flux of occupied area $q_{des,A}$ shall be calculated respectively on the area of the peripheral heating surface A_R and on the area of the occupied heating surface A_A.

- Step 5: For the design of the floor heating systems, determine the room used for design with the maximum design heat flux $q_{max} = q_{des}$.
- Step 6: Determine the floor heating system such as the pipe spacing and the covering type, and design heating medium differential temperature $\Delta\theta_{H,des}$ based on the maximum design heat flux q_{max} and the maximum surface temperature $\theta_{F,max}$ from the field of characteristic curves.
- Step 7: If the design heat flux q_{des} cannot be obtained by any pipe spacing for the room used for the design, it is recommended to include a peripheral area and/or to provide supplementary heating equipment. In this case, the maximum design heat flux q_{max} for the embedded system may now occur in another room. The amount of heat output of supplementary heating equipment Q_{out} is determined by the following equation:
- Step 8: Determine the backside thermal resistance of insulating layer $R_{\lambda,ins}$ and the design heating medium flow rate.
- Step 9: Estimate the total length of heating circuit.

If intermittent operation is common, the characteristics of the increase of the heat flow and the surface temperature and the time to reach the allowable conditions in rooms just after switching on the system shall be considered.

Floor cooling system design requires determining cooling surface area, type, pipe size, pipe spacing, supply temperature of the cooling medium, and design cooling medium flow rate. The design steps are as follows.

- Step 1: Calculate the design sensible cooling load $Q_{N,s}$. The design sensible cooling load $Q_{N,s}$ does not include the adjacent heat gains. This step shall be conducted in accordance with standards for cooling load calculation, e.g. EN 15243, based on an index such as operative temperature (OT).
- Step 2: Determine the minimum supply air quantity needed for dehumidifying.
- Step 3: Calculate latent cooling available from supply air and also calculate sensible cooling available from supply air.
- Step 4: Determine remaining sensible cooling load to be satisfied by radiant system. Also designate or calculate the relative humidity and dew point, because cooling system shall operate within a temperature range above the dew point, which shall be specified depending on the respective climate conditions of the country.
- Step 5: Determine the area of the cooling surface A_F, excluding any area covered by immovable objects or objects fixed to the building structure.
- Step 6: Establish a minimum permissible surface temperature in accordance with ISO 11855-1 in consideration of the dew point.
- Step 7: Determine the design heat flux q_{des}. For floor cooling systems including a peripheral area, the design heat flux of peripheral area $q_{des,R}$ and the design heat flux of occupied area $q_{des,A}$ shall be calculated respectively on the area of the peripheral cooling surface A_R and on the area of the occupied cooling surface A_A.
- Step 8: For the design of the floor cooling systems, determine the room used for design with the maximum design heat flux $q_{max} = q_{des}$.
- Step 9: Determine the floor cooling system such as the pipe spacing and the covering type, and design cooling medium differential temperature $\Delta\theta_{H,des}$ based on the maximum design heat flux q_{max} and the minimum surface temperature $\theta_{F,min}$ from the field of characteristic curves.
- Step 10: If the design heat flux q_{des} cannot be obtained by any pipe spacing for the room used for the design, it is recommended to provide supplementary cooling equipment. In this case, the maximum design heat flux q_{max} for the embedded system may now occur in another room.
- Step 11: Determine the backside thermal resistance of insulating layer $R_{\lambda,ins}$ and the design cooling medium flow rate.
- Step 12: Estimate the total length of cooling circuit.

Thermo Active Building Systems (TABS)

ISO 11855-4 allows the calculation of peak cooling capacity of Thermo Active Building Systems (TABS), based on heat gains such as solar gains, internal heat gains and ventilation, and the calculation of the cooling power demand on the water side to size the cooling system as regards the chiller size, fluid flow rate, etc. ISO 11855-4 defines a detailed method aimed at the calculation of heating and cooling capacity in non-steady state conditions. A Thermally Active Surface (TAS) is an embedded water based surface heating and cooling system, where the pipe is embedded in the central concrete core of a building construction. The building constructions embedding the pipe are usually the horizontal ones. As a consequence, floors and ceilings are usually referred to as active surfaces. Looking at a typical structure of the TAS, heat is removed by a cooling system (for instance, a chiller), connected to pipes embedded in the slab. Thermally active surfaces exploit the high thermal inertia of the slab in order to perform the peak-shaving. The peak-shaving is to reduce the peak in the required cooling power, as it is possible to cool the structures of the building during a period while the occupants are absent (during night time, in office premises). This way the energy consumption can be reduced and the lower night time electricity rate can be used. At the same time a reduction in the size of heating/cooling system components (including the chiller) is possible.

TABS may be used both with natural and mechanical ventilation (depending on weather conditions). Mechanical ventilation with dehumidifying may be required depending on external climate and indoor humidity production. The required peak cooling power needed for dehumidifying the air during day time is sufficient to cool the slab during night time. As regards the design of TABS, the planner needs to know if the capacity at a given water temperature is sufficient to keep the room temperature within a given comfort range. Moreover, the planner needs also to know the heat flow on the water side to be able to dimension the heat distribution system and the chiller/boiler.

When using TABS, the indoor temperature changes moderately during the day and the aim of a good TABS design is to maintain internal conditions within the comfort range, i.e. $-0,5 < PMV < 0,5$, during the day, according to ISO 7730. Some detailed building system calculation models have been developed to determine the heat exchanges under unsteady state conditions in a single room, the thermal and hygrometric balance of the room air, prediction of comfort conditions, check of condensation on surfaces, availability of control strategies and calculation of the incoming solar radiation. The use of such detailed calculation models is, however, limited due to the high amount of time needed for the simulations. The development of a more user friendly tool is required. Such a tool is provided in this part of ISO 11855, and allows the simulation of TAS.

Installation

ISO 11855-5 establishes guidelines on the installation of embedded radiant heating and cooling systems. It specifies uniform requirements for the design and construction of heating and cooling floors, ceiling and wall structures to ensure that the heating/cooling systems are suited to the particular application. The requirements specified by this part of ISO 11855 are applicable only to the components of the heating/cooling systems and the elements which are part of the heating/cooling surface and are installed for the heating/cooling systems.

Control

ISO 11855-6 describes the control of hydronic systems to enable all embedded systems to perform as simulated. The design documents shall include specifications for the control system. The control system shall be capable of varying heating or cooling outputs as well as maintaining predetermined room or surface temperatures. The control system shall, if specified, protect buildings and equipment against frost and moisture damage where necessary (when normal comfort temperature level is not required) and prevent condensation from occurring. The design of the control system shall take into account the building, its intended use and the effective functioning of the embedded system, efficient use of energy and avoid conditioning the building at full design conditions when not required.

Due to the high impact that fast varying heat gains (e.g. sunshine through windows) may have on the room temperature, it is necessary that the radiant system control to compensate this by reducing or increasing the temperature difference between room and heated/cooled surface and partly on the difference between room and the average temperature output. For the low temperature heating and high temperature cooling systems, the "self-regulating effect" is significant. The "self-regulating" depends on the average water temperature in the panels. It means that fast change of operative temperature will equally change heat exchange and result in influence of total heat exchange. This impact is bigger for systems with surface temperatures close to room temperature because the change of one degree represents a higher percentage based on a small temperature difference than on a high temperature difference. The self-regulating effect of low temperature heating and high temperature cooling systems supports the control equipment (e.g. individual room temperature control) in maintaining a stable thermal environment providing comfort to persons in the room.

Water based radiant heating and cooling systems need hydronic balancing. The components shall be adjusted in order to ensure the required flow rates. Under dynamic conditions, e.g. during the heating up/cooling down period, it must be ensured that the hydraulic interaction between the different circuits is small (the flow rates in different circuits shall not be greater than the design flow rates). Depending on the situation of the heating/cooling system, the panel distribution system shall be equipped with facilities for degassing and sludge separation.

The control modes of embedded systems are based on three system levels:

1) Local (room) control, where the energy supplied to a room is controlled
2) Zone control normally consisting of several spaces (rooms)
3) Central control where energy supplied to the whole building is controlled by a central system

The control system classification is based on performance level:

1) Manual: The energy supply to the conditioned space is only controlled by a manually operated device
2) Automatic: A suitable system or device automatically controls energy to the conditioned spaces
3) Timing: Function of energy supplied to a conditioned space is shut off or reduced during scheduled periods, e.g. night setback (not necessarily applicable for cooling)
4) Advanced timing: Function of energy supply to the conditioned space is shut-off or reduced during scheduled periods, e.g. daytime with more expensive electricity tariff. Re-starting of the energy supply is optimized based on various considerations, including reduction of energy use (not applicable in commercial buildings).

Conclusion

In general, HVAC systems have been designed as all-air HVAC systems in European countries. Radiant heating and cooling systems can be integrated with general HVAC system by separating the tasks of ventilation and thermal space conditioning. By using the primary air distribution to fulfill the ventilation requirements and the secondary water distribution system to thermally condition the space, the amount of air circulation through buildings can be reduced significantly, because the ventilation air can be supplied by outside fresh air without affecting the recirculation of air. To secure the performance of radiant heating and cooling system, there must be standards for processes and conditions that determine the heating and cooling capacity of radiant heating and cooling systems. The purpose of ISO 11855 lies above all in enabling the radiant heating and cooling systems to perform in accordance with occupants' comfort by providing standards for determining the heating and cooling capacity of radiant heating and cooling systems. These standards may be seen as integrated design standards that make possible the effective design of the entire system by providing standards regarding heat emission calculation, panel design, and system construction. ■

References

ISO 11855-1 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —Part 1: Definition, symbols, and comfort criteria.

ISO 11855-2 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems — Part 2: Determination of the design and heating and cooling capacity.

ISO 11855-3 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —Part 3: Design and dimensioning.

ISO 11855-4 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —Part 4: Dimensioning and calculation of the dynamic heating and cooling capacity of Thermo Active Building Systems (TABS).

ISO 11855-5 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —Part 5: Installation.

ISO 11855-6 Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —Part 6: Control.

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Retrofitting of the HVAC plant of an elderly people residential care home

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About 14% of European population is over 65 years of age, and it is expected that this number will double by 2050. Care homes operate 24 hours a day, 365 days a year, with full occupancy. Some of the best available technologies were compared by an energy and economic analysis for a residential care home for elderly people in Vicenza (North East of Italy).

Keywords: Energy efficiency; HVAC plant; cogeneration; trigeneration; photovoltaic.

Introduction

Among the public (but not only) buildings, residential care homes for elderly people are ones of the potentially most important because of the increasing European population aging. Nowadays about 14% of European population is over 65 years of age, and it is expected that this number will double by 2050 [1]. These figures in Italy assume dramatic values, as actually we have 21.4% of population over 65 (13 million) and we will have 33.6% (18.7 million) by 2050 [2] [3]. Nowadays about 1.5 million elderly live in more than 24,000 care homes in Europe (300,000 elderly live in just less than 6,000 in Italy). These buildings operate 24 hours a day, 365 days a year, with full occupancy. For these reasons reducing energy consumption in residential care homes is important.

In this context, this paper focuses on the study of the energy performance of a quite new residential care home for elderly people in Vicenza (North East of Italy). A preliminary energy audit was carried out in order to obtain appropriate information about energy consumption of the building. Moving from the request of the managers responsible for running the care home, a Trnsys simulation model of the building and heating/cooling plant system was developed aiming at testing different solutions to retrofit the heating/cooling plant.

Building description and energy loads calculation

The considered building is a 29,889 m³ residential care home for elderly people built during the 2002–2004 period and located in Vicenza (North Italy). The main data of the building are reported in **Table 1**. The model of this three floors building (about 2,800 m² each) was implemented using "Trnsys3d for SketchUp" (**Figure 1**) and "TRNbuild".

An annual simulation of both heating and cooling loads was run with one-hour time step: annual heating and cooling energy use of the building were estimated to be 622,000 kWh and 529,000 kWh, with peak loads of 610 kW and 707 kW respectively. Concerning electric loads, there was availability of disaggregated data (on a quarter hourly basis) from local electric distributor for the years 2013 and 2014. Nevertheless, such data was subject to the different environment conditions (e.g. summer 2014 was extremely colder than usual in Italy) and building operating conditions. So, with some approximation based on information found by talks with technical personnel of the home, an hourly electric load profile was built taking into consideration monthly electric consumptions (deduced by electric bills), the actual installed electric power (400 kW) and the real data.

Table 1. Main data of the building.

Zone	Vicenza (North Italy)
Heating degree days	2479
Building destination	Residential care home
Internal volume (m^3)	29,889
Floors	3
External Surface / Heated Volume ratio (m^{-1})	0.35
Heating period	October 15th – April 15th
Cooling period	April 16th – October 14th
Typical temperature set-point, heating period (°C)	24.0
Typical relative humidity set-point, heating period (%)	–
Typical temperature set-point, cooling period (°C)	26.0
Typical relative humidity set-point, cooling period (%)	50%
Ventilation flow rate (m^3 s^{-1})	4.09
External walls transmittance (W m^{-2} K^{-1})	0.517
Floor transmittance (W m^{-2} K^{-1})	0.433
Roof transmittance (W m^{-2} K^{-1})	0.287
Windows transmittance (W m^{-2} K^{-1})	3.155
Average internal heating gains (W m^{-2})	4.0
Patients and staff	160

Heating/cooling plant description and primary energy consumption calculation

The existing heating, ventilation and cooling plant is mixed air/water; it is set up by:

- heating and cooling plant (vapour compression electric chiller and natural gas boilers);
- ventilation and air conditioning plant (many air handling units (AHU) to control relative humidity and for the necessary air changing inside the building; main units are set up by a rotary heat exchanger, pre-heating, cooling and dehumidification and post-heating sections). The common sites of the building (corridors, halls) are served by fan-coils and ventilation air, rooms and bathrooms are served by radiators and ventilation air, service and technical rooms by small AHUs or air heaters;
- air extraction plant (for service rooms).
- The object of the work is the energy and economic analysis of the retrofitting of the heating and cooling plant only. The main components are:
- one air/water vapour compression electric chiller (2 circuits – 4 oil free centrifugal compressors per

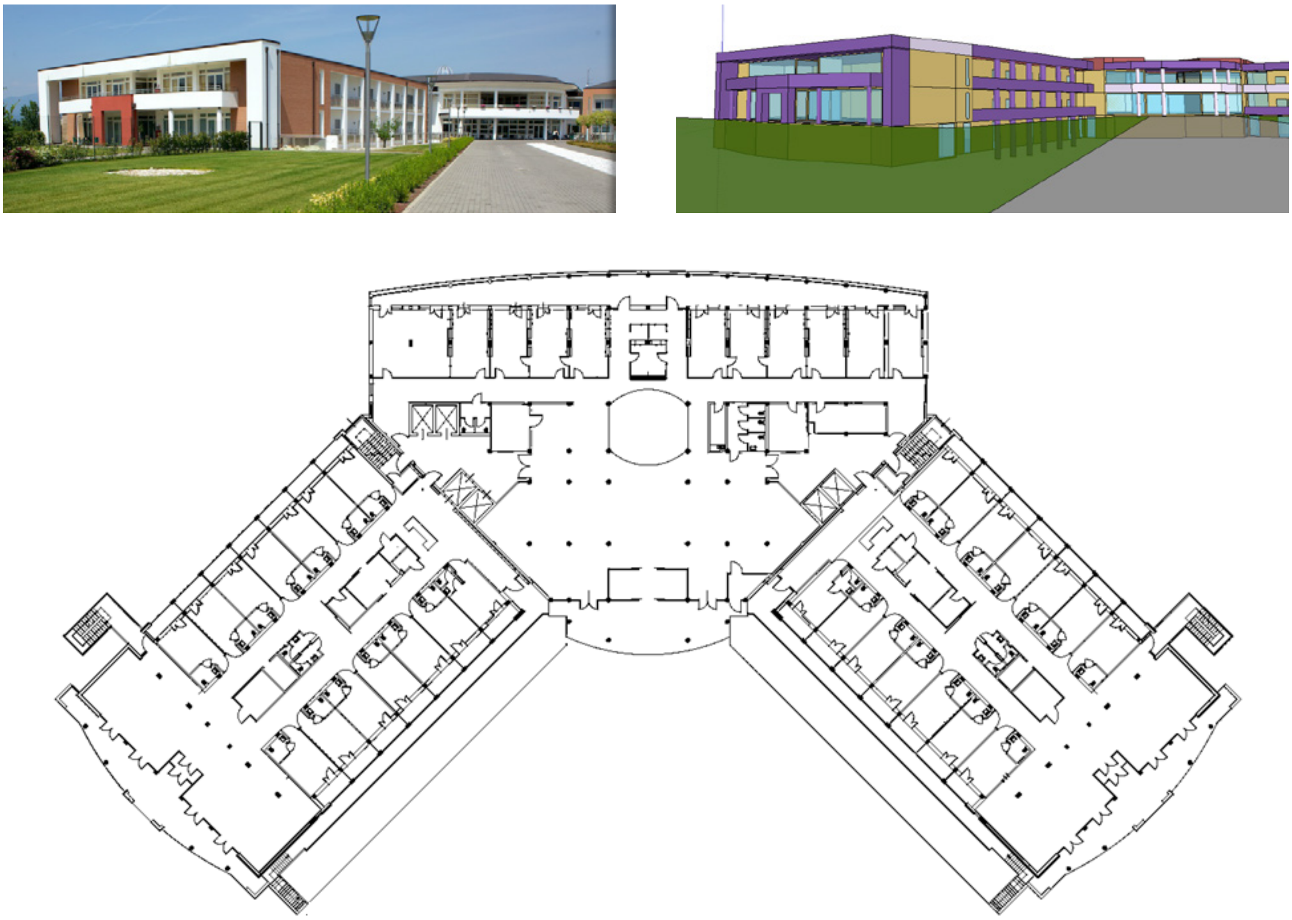

Figure 1. Photo, 3D model and layout (ground floor) of the considered residential care home.

circuit) which provides cold water for fan-coils and cold coils in AHUs. The nominal cooling power is 895 kW, 270 kW is the nominal electric power consumption, EER = 3.31. These performances are labelled for summer external air 35°C and evaporator input/output 12/7°C;

- two natural gas boilers with two-steps burners (nominal useful power 670 kW, minimum useful power 425 kW each) provide the thermal energy for heating, domestic hot water and pre and post-heating in AHUs.

The chiller and the boilers supply three hydraulic circuits. The "hot collector" supplies hot water to the radiators, air heaters, pre and post heating coils in the AHUs. It supplies also the domestic hot water plate heat exchanger that loads a 5,000 l tank. The "hot/cold collector" supplies hot (during heating season) or cold (during cooling season) water to the fan-coils. The "cold collector" supplies cold water to the cold coils of the AHUs (obviously during cooling season only). Reference [4] reports the temperature set points of the different energy uses and the schedules of the main equipment.

The first step was the simulation by Trnsys model of the just described existing plant ("As Is" solution). The validation of the model was carried out by analyzing flow rates and temperatures of the different circuits and by comparing simulated and real energy consumptions of the heating/cooling plant system. **Figure 2** reports the annual profiles of monthly electrical energy (EE) and natural gas (NG) consumptions, both simulated and real (referring to 2012, 2103 and 2014 available energy bills). Considering the variability of the environment and building operating conditions, the concordance between simulated and real data is quite good (the slight overestimate of electrical energy consumption during mid seasons is due to the wider cooling period in the simulations with respect to the reality).

Alternatives for the heating/cooling plant energy retrofitting

Different efficient technologies among the most common known were taken into consideration for the energy improvement of the heating/cooling plant, chosen referring to previous economic estimates in possession of the managers responsible for running the care home. Main technical data of all the equipment considered in the present analysis are reported in [4].

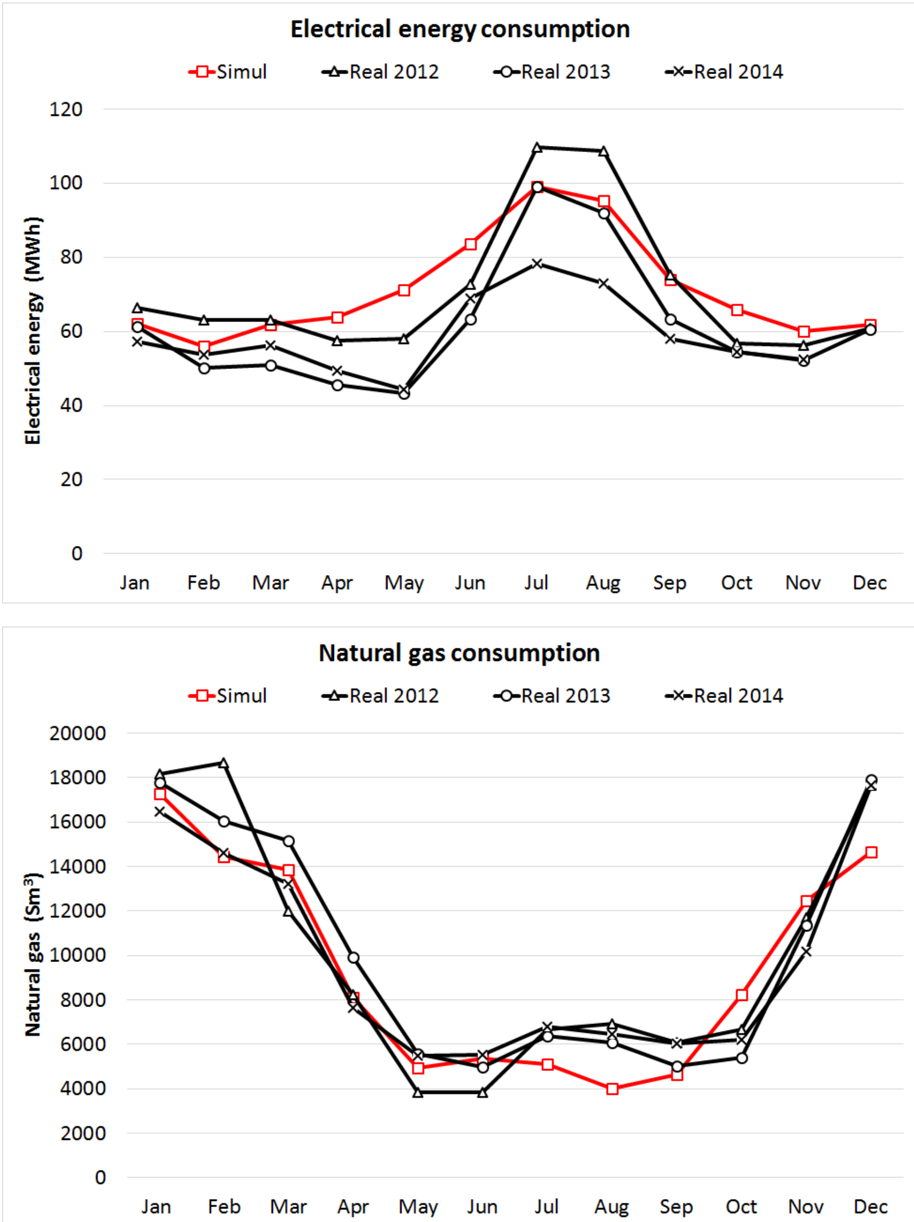

Figure 2. Annual profile of monthly electrical energy and NG consumptions.

For the sake of brevity, extremely concise information is here reported:

- photovoltaics, PV: different kinds of mono and policrystalline modules were considered, varying the peak power in the 230–315 W_p range (14-16.9% nominal efficiency), considering a fixed available area (two terraces of 108 m² each on the roof and three parking shelters for 1,022 m²). In one alternative it was supposed to be able to install more PV modules on the roof (available area 350+350 m²). A penalization factor of –0.60% per year was considered in order to take into account the annual decrease of useful power. All the other parameters and input in Trnsys model were set in order to suitably simulate the PV plant;
- combined heat and power, CHP (cogeneration): natural gas fueled internal combustion engines were considered with two different electrical nominal power (103 and 199 kW) and two different control strategies (electric load following and electric load following but during the 6.00 am-9.00 pm period only). A smaller size one was further considered (60 kW nominal power, electric - load following);
- combined cooling, heat and power, CCHP (trigeneration): in order to extend the operation hours of the cogeneration plants, three of the previous cases were implemented considering to use the heat produced by the engine also during cooling season by coupling suitable sized lithium bromide-water single effect absorption chiller;
- heat pumps: different sizes of vapour compression air/water heat pumps with both electric motor (EHP) and natural gas engine (GEHP) were considered. For the characterization of their performances the Authors referred to the UNI/TS 11300-4 and UNI EN 15316-4-2 method [5] [6] [7].

It was assumed to maintain the two actually present boilers with the double scope of both thermal integration and backup. Finally, primary energy consumptions were calculated taking into account thermal and electrical nominal efficiencies (on Lower Heating Value) for the cogenerators and nominal performances (COP) for the heat pumps, and considering their variations with the partial load (data derived by motors suppliers for the cogenerators – data derived implementing the UNI/TS 11300-4 and UNI EN 15316-4-2 calculation methods for quantifying energy loads and efficiencies of electrical HP-based heating plants).

Investment costs were determined by purchase, installation and first set up, besides costs of the existing plant adaptation [7] [8] [9] [10] (**Table 1**). The same table reports on the ordinary maintenance costs of the different solutions, while such costs for the existing plant are considered to be 5,000 € $year^{-1}$. Reference [4] reports on the costs of energy: natural gas and electricity purchased by the local distributors.

Concerning the management of the electricity produced by the cogenerators and photovoltaic plant besides own consumption, it was considered to be sold to the GSE (Energy Services Manager) at a constant price (fixed in 5.5 c€ kWh^{-1}). For the alternatives here considered, the incomes from the energy efficiency certificates resulted negligible because they did not, or only slightly, satisfy the minimum primary energy saving index provided by the 2004/8/EC directive [11]. Finally, for the primary energy calculations we considered the conversion factors reported in [12] [13], i.e. 1 $kWh_{pr}\ kWh_{th}^{-1}$ and 2.17 $kWh_{pr}\ kWh_{el}^{-1}$ respectively for natural gas and electricity.

Results and discussion

The solutions that maximize NG consumptions (even if minimizing the boilers consumption) are the ones with the installation of the 199 kW_{el} nominal power cogenerators, while the solutions minimizing the consumption in absolute terms concern installing the biggest sized electric heat pumps ("EHP_311 kWt" and "EHP_424 kWt"). The balance, in terms of electrical energy, takes into account the "self-produced" and the "sold to grid" electricity: the solutions that minimize the purchase from the grid (by more than 20 times with respect to the "As Is" plant) are the ones that maximize the self-production ("2, 199 kW – el. following" with and without trigeneration).

A more comprehensive comparison, carried out in terms of primary energy (PE), is reported in specific terms (per square meter of building surface) and in absolute terms as well in **Table 2**. Self-produced electrical energy accounts for a negative value in terms of consumption, so the best solutions foresee the installation of 199 kW_{el} cogenerator (coupling the absorption chiller allows a very slight improvement). These solutions allow to reduce by 63% the PE consumption. Anyway, if one looks at the benchmark for Italy suggested by [1] (234 kWh m^{-2} $year^{-1}$), all the CHP and CCHP alternatives and two PV solutions are performant. On the other hand, heat pumps, both electric and natural gas driven, are not advantageous at all and, in some cases, they lead to an increase of the PE consumption. This is due to the penalized operation of air/water heat pumps in the Vicenza winter

Table 1. Costs of the different heating/cooling plant retrofitting solutions (*Refer to the HP nominal thermal power at the condenser at A7/W45 conditions) (**Electric motor efficiency=0.95 – Internal combustion engine efficiency=0.3 [7]).

Alternative	Cost per kW (Electric/Thermal/Cooling) installed	Electric/ Thermal/ Cooling installed power	Set up and adaptation	Total investment cost	Annual ordinary maintenance cost
As Is	Sunk cost	-	-	-	5000 €
1, Solon Black 220/16 mono (265 W_p)	On the roof: 1,300 € kW_p^{-1} On the parking shelter: 1,700 € kW_p^{-1}	33.9 kW_p^{-1} 156.9 kW_p^{-1}	-	310,792 €	3400 €
2, Solon Black 220/16 poli (230 W_p)	On the roof: 1,300 € kW_p^{-1} On the parking shelter: 1,700 € kW_p^{-1}	29.4 kW_p^{-1} 136.2 kW_p^{-1}	-	269,744 €	3400 €
3, Abba Solar ASP 60 245-250 poli Plus (250 W_p)	On the roof: 1,300 € kW_p^{-1} On the parking shelter: 1,700 € kW_p^{-1}	32.0 kW_p^{-1} 148.0 kW_p^{-1}	-	293,200 €	3400 €
4, Renesola 156 mono (275 W_p)	On the roof: 1300 € kW_p^{-1} On the parking shelter: 1700 € kW_p^{-1}	35.2 kW_p^{-1} 162.8 kW_p^{-1}	-	322,520 €	3400 €
5, Renesola Virtus II JC315M-24/Abs poli (315 W_p)	On the roof: 1300 € kW_p^{-1} On the parking shelter: 1,700 € kW_p^{-1}	30.2 kW_p^{-1} 156.2 kW_p^{-1}	-	304,920 €	3400 €
1, Solon Black 220/16 mono (265 W_p) _Plus	On the roof: 1,300 € kW_p^{-1} On the parking shelter: 1,700 € kW_p^{-1}	110.2 kW_p^{-1} 156.9 kW_p^{-1}	-	410,008 €	3400 €
1, 103 kW – el. following	1,800 € kW_{el}^{-1}	103 kW_{el}	10000 €	195,400 €	0.020 € kWh_{el}^{-1}
2, 199 kW – el. following	1,400 € kW_{el}^{-1}	199 kW_{el}	10000 €	288,600 €	0.020 € kWh_{el}^{-1}
3, 103 kW – el. following – day time only	1,800 € kW_{el}^{-1}	103 kW_{el}	10000 €	195,400 €	0.020 € kWh_{el}^{-1}
4, 199 kW – el. following – day time only	1,400 € kW_{el}^{-1}	199 kW_{el}	10000 €	288,600 €	0.020 € kWh_{el}^{-1}
5, 60 kW – el. following	2,000 € kW_{el}^{-1}	60 kW_{el}	8000 €	128,000 €	0.020 € kWh_{el}^{-1}
2, 199 kW – el. following – trigeneration	Cogenerator: 1,400 € kW_{el}^{-1} Abs. chiller: 500 € kW_c^{-1}	Cogenerator: 199 kW_{el} Abs. chiller: 147 kW_c	30000 €	382,100 €	0.020 € kWh_{el}^{-1}
3, 103 kW – el. following – day time only – trigeneration	Cogenerator: 1,800 € kW_{el}^{-1} Abs. chiller: 600 € kW_c^{-1}	Cogenerator: 103 kW_{el} Abs. chiller: 70 kW_c	25000 €	252,400 €	0.020 € kWh_{el}^{-1}
5, 60 kW – el. following – trigeneration	Cogenerator: 2,000 € kW_{el}^{-1} Abs. chiller: 625 € kW_c^{-1}	Cogenerator: 60 kW_{el} Abs. chiller: 50 kW_c	20000 €	171,250 €	0.020 € kWh_{el}^{-1}
EHP_99 kWt*	500 € kW_{el}^{-1}	29 kW_{el}	20000 €	34,600 €	10 € kW_{el}^{-1}
GEHP_99 kWt* **	250 € kW_{th}^{-1}	153 kW_{th}	30000 €	68,239 €	5 € kW_{th}^{-1}
EHP_209 kWt*	500 € kW_{el}^{-1}	58 kW_{el}	20000 €	48,900 €	10 € kW_{el}^{-1}
GEHP_209 kWt* **	250 € kW_{th}^{-1}	300 kW_{th}	30000 €	105,042 €	5 € kW_{th}^{-1}
EHP_311 kWt*	500 € kW_{el}^{-1}	86 kW_{el}	20000 €	63,000 €	10 € kW_{el}^{-1}
GEHP_311 kWt* **	250 € kW_{th}^{-1}	458 kW_{th}	30000 €	144,479 €	5 € kW_{th}^{-1}
EHP_424 kWt*	500 € kW_{el}^{-1}	116 kW_{el}	20000 €	78,000 €	10 € kW_{el}^{-1}
GEHP_424 kWt* **	250 € kW_{th}^{-1}	618 kW_{th}	30000 €	184,438 €	5 € kW_{th}^{-1}

Table 2. Annual Primary Energy consumption of the different solutions, both absolute (MWh) and specific (kWh m^{-2}).

Alternative	Total PE		Saving
	(MWh)	(kWh m^{-2})	
As Is	2,944	351	–
1, Solon Black 220/16 mono (265 W_p)	2,015	240	32%
2, Solon Black 220/16 poli (230 W_p)	2,058	245	30%
3, Abba Solar ASP 60 245-250 poli Plus (250 W_p)	2,058	245	30%
4, Renesola 156 mono (275 W_p)	1,942	231	34%
5, Renesola Virtus II JC315M-24/Abs poli (315 W_p)	2,138	255	27%
1, Solon Black 220/16 mono (265 W_p)__Plus	1,694	202	42%
1, 103 kW – el. following	1,236	147	58%
2, 199 kW – el. following	1,101	131	63%
3, 103 kW – el. following – day time only	1,524	181	48%
4, 199 kW – el. following – day time only	1,317	157	55%
5, 60 kW - el. following	1,689	201	43%
2, 199 kW – el. following – trigeneration	1,092	130	63%
3, 103 kW – el. following – day time only – trigeneration	1,581	188	46%
5, 60 kW – el. following – trigeneration	1,720	205	42%
EHP_99 kWt	2,999	357	–2%
GEHP_99 kWt	3,067	365	–4%
EHP_209 kWt	2,911	347	1%
GEHP_209 kWt	3,064	365	–4%
EHP_311 kWt	2,916	347	1%
GEHP_311 kWt	3,111	370	–6%
EHP_424 kWt	2,931	349	0%
GEHP_424 kWt	3,135	373	–6%

climate with frequent defrosting necessity ([9] [10] [14]) and especially due to the high temperature of water to be produced at the condenser (70°C) serving the radiators and the pre and post heating coils of AHUs.

The interest rate, the inflation rate and the period of the economic analysis were fixed at 4%, 1.8% and 20 years respectively. From the economic point of view, the comparison between the different alternatives did not consider the investment cost of substituting the existing plant, as it was considered to be a sunk cost (**Table 1**). The best solutions in terms of trade-off between maximum differential (investment alternative versus "As Is" solution) net present worth (NPW) and minimum discounted pay-back period (DPB) are the CHP/CCHP ones with installed electrical power around or less than 100 kW_{el}. In these cases, NPW is around 630–670 k€ and DPB is between 3 and 4 years (**Figure 3**); these are very interesting results, also considering that we are talking about the plant of a public building stock (so payback periods can be quite longer than in industry sector). These are also the solutions that allow the minimization of thermal energy dissipation by the cogenerator. From this point of view, it is interesting to observe that a further increase of the economic viability of such alternatives would be obtained by operating the cogenerator with the thermal load following logic.

Concerning the "2, 199 kW" solution it is worth to stress that coupling the absorption chiller can improve the economic viability, but even more advantageous is the operation during the day only (respectively NPW increases from 195 to 332 to 423 k€ and DPB decreases from 10.9 to 9.6 to 7.1 years). Photovoltaics is also interesting, even if it allows smaller NPWs (200-300 k€) and longer DPBs (9-11 years). Should be possible to use more surface on the roofs ("1, Solon Black 220/16 mono (265 W_p)__Plus" solution) the economic viability would increase (NPW=440 k€, DPB=8.6 years). It is interesting to observe that heat pump solutions are not advantageous at all as they present a greater than "As Is" solution annual cash flow for the reasons explained before.

In reference [4] these conclusions are completed by a sensitivity analysis in order to compensate for the uncertainty of some of the parameters here considered, such as natural gas and electrical energy costs.

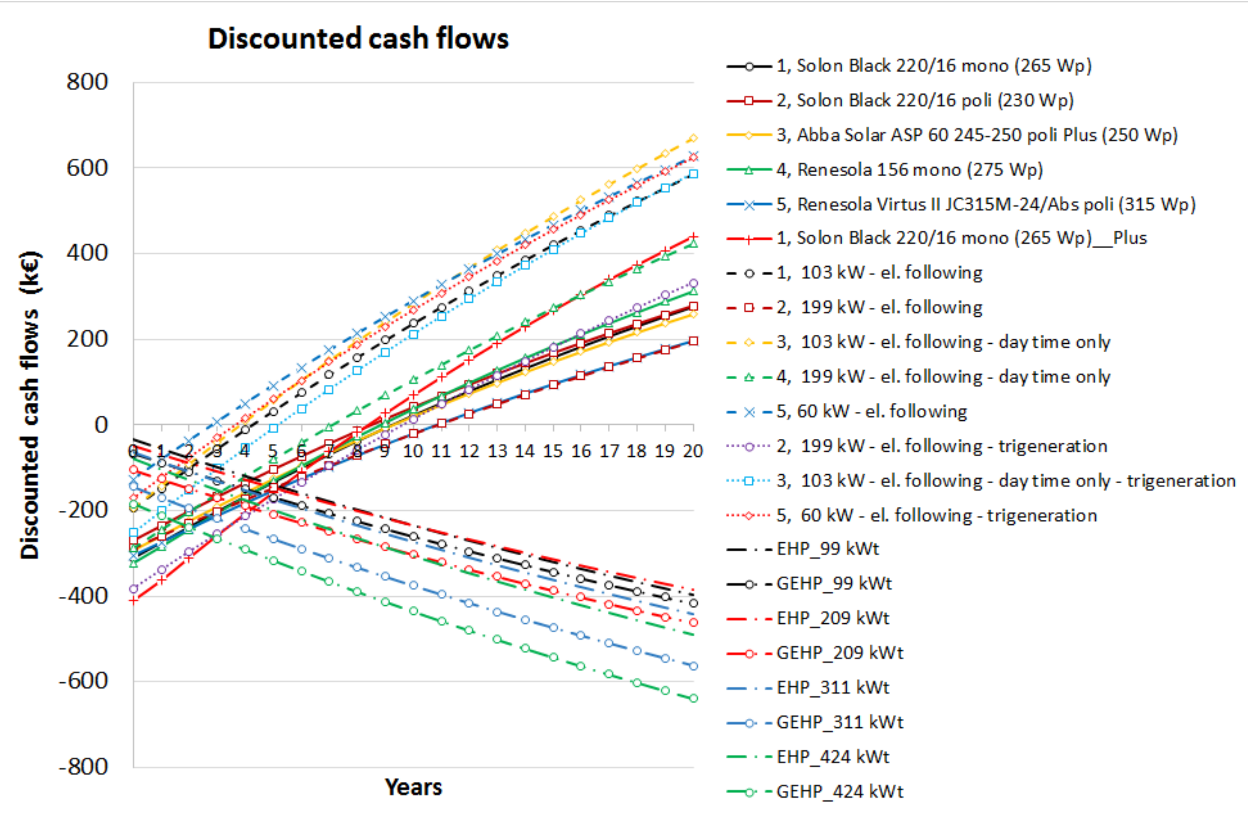

Figure 3. Discounted differential (between the alternatives and the "As Is" solution) cash flows of the different solutions. NPWs can be read at the end of the period of analysis (20 years), DPBs by the interceptions of the curves with the x-axis.

Conclusions

Many energy efficiency interventions can be thought to be implemented in residential care homes for elderly people as the greatest part of them are old buildings, i.e. built before regulations on buildings energy performance and economic incentives. In this sense, interventions on the building (e.g. retrofitting of the opaque and transparent surfaces by thermal insulation and windows substituting) are the first ones that should be implemented; installing a solar thermal plant and substituting the old lighting appliances by more efficient ones should be the second ones. In more recent (and so more energy performant) buildings some retrofitting interventions in heating/cooling plant can be analyze. Photovoltaics, cogeneration, trigeneration, electric and gas engine heat pumps were considered in this study and the energy and economic viability were evaluated. Cogeneration with small size engines (with respect to the installed electric power by local distributor) result the most advantageous solutions, whereas coupling a single-effect absorption chiller do not significantly improve the advantage. Photovoltaics as well allows an interesting energy saving with respect to the existing plant, even if with longer payback periods. Air/water heat pumps (the most economic and widespread diffused ones) are not advantageous at all in this case because of the high temperature at condenser and because of the cold and humid winter climate typical of the Po Valley (that implies frequent defrosting of the evaporator coil). The main conclusions of this study will be delivered to the managers responsible for running the residential care home in order to make energy efficient informed decisions. ■

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Effects of different heat and cold generation scenario (Heat pump versus district heating and chiller) in a building with variable air volume versus demand controlled ventilation systems




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The objective is to verify the operation of an installed system in an OfficeRetail building located in Sweden. The focus is on comparing the energy saving achievable with different ventilation systems with focus on keeping or improving the indoor comfort condition. The systems taken into account are: a constant air volume, a variable air volume and a demand controlled ventilation system. The energy saving is analysed considering two different heat & cold generation scenarios (heat pump versus district heating and chiller) in order to make the evaluation in terms of purchased energy and economy. The findings show the importance of continuous monitoring, the advantages of the choice of an advanced ventilation system and the importance of the chosen generation systems considering energy and economical costs.

Keywords: Demand controlled ventilation, Energy production, Energy distribution.

Introduction

Considering demand controlled ventilation, constant air volume and variable air volume systems two types of generic HVAC system can be categorized:

- constant air volume (CAV) - ventilation with constant airflow. This type of ventilation system is characterized by one or two manual operations (e.g. ON/OFF), chronological management or control of supply/extract temperature;
- variable air volume (VAV), in this case the control is managed in a manual or continuous mode with prefixed models or time steps.

A building's HVAC system must ensure a good indoor air quality and both thermal and acoustic comfort based on (EN ISO 7730, 1994; EN 15251, 2007). A variable air volume system with automatic control based on the real need, automatically adapting the supply air temperature and supply airflow to keep the target conditions in a building is defined as demand controlled ventilation (DCV). In other words, a DCV system is designed with the aim of supply a quantity of fresh air fitted with the need in every situation, ensuring the right quantity of fresh airflow and right environmental conditions in terms of temperature and relative humidity and air exchange. The parameters controlling a DCV system are: temperature and relative humidity, carbon dioxide (CO_2) and/or volatile organic compounds (VOC).

Office-Retail building Engelsons in Falkenberg (Western Sweden) was built in 2009. It is used as an office, retail with a packing area and a warehouse, and is considered as a multifunctional building with the indoor climate conditions controlled by a ventilation system and heating/cooling based on air and water: air diffusers with constant airflow and climate beams with constant volume on the air side and variable on the water side. The building envelope is made of steel and concrete (**Table 1**), all insulated in order to reduce the thermal losses.

The **HVAC system** is a constant air volume system (CAV) with three independent air handling units which are equipped with high efficiency rotary heat recovery: TA/FA1 with 1,000 l/s (3,600 m³/h) serves a warehouse, TA/FA3 with 1 200 l/s (4,300 m³/h) for a rented office and TA/FA2 with 1,800 – 2,400 l/s (6,500 – 8,650 m³/h) is the only monitored air handling unit and it serves as an office, a retail and a packing area (**Figure 1**). The evaluation of indoor climate and energy monitoring in the Engelsons can be found at (Errico 2014).

All of the offices are heated and cooled by climate beams. In the common areas (e.g. corridors) air diffusers with constant airflow are installed. The retail area is heated by air diffusers with constant airflow and cooled by central air, and post-installed fan coils. The warehouse is heated only with air diffusers specially designed for large spaces. The packing area is air heated via air diffusers and cooled with an active beam above the work desk, a split system to compensate the heat peaks during summer, and during winter time additional fan coils are turned on to satisfy the thermal balance and reach the indoor set temperature.

Lowering supply of fresh air inside the building (as a sum of airflow based on design occupancy rate and designated airflow per square meter) so called night mode allows energy saving by recirculation of the most part of indoor air (already at 20°C) with minimum fresh air supply of 0,35 l/s/m² (1,26 m³/h) based on (Boverket, 2011). The temperature difference between the recirculation air and the water in the coils is lowered. This means that the power supplied to the air is lower as well as the energy needed for heating the airflow from 14°C (possible temperature after the recovery) to 20°C. This will also allow the decreasing of the temperature by 2K, i.e. the extract air has to reach 18°C before the heating system increases the thermal power of water flow.

Energy production for heating and cooling is supplied by a ground coupled double effect heat pump of 55 kW with a COP = 4,85 and a EER = 3,85 that during winter time takes heat from the ground, whereas during

Table 1. Overview of main buildings´ characteristics as used for design of HVAC system.

Building	Value	Characteristics	U-value [W/(m.K)]	Surface [m²]
Floor surface	2,203 m²	Walls	0.315	1,483
Number of storeys	2	Floor	0.146	2,203
Height of each storey	3.8 m	Ceiling	0.193	2,203
Floor surface	2,203 m²	Windows	1.2	106

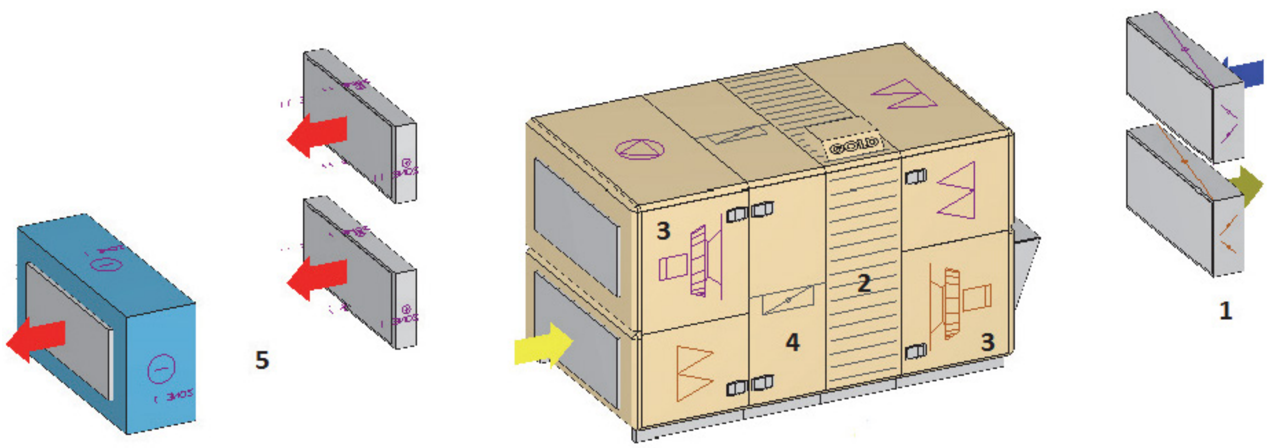

Figure 1. TA/FA2 dampers (1), recovery (2), fans (3), bypass section (4) and coils (5).

the summer it rejects the heat absorbed by the building back to the ground in order to avoid the thermal unbalance of it. As an auxiliary system is used an electric boiler of 42 kW and a classic split system cools the package area during the summer season.

Methods

The simulation scenarios for various ventilation systems (**Table 2**) are used to evaluate the actual state of the building in a verified simulated model (CAV), and potential improvements if installing more controlled systems such as VAV, DCV and DCV Class 1 with indoor climate set points according European standard (EN 15251 2007). The VAV system is modelled with reducing SFP and allowing a temperature gap of ± 2°C (cooling or heating mode) during the unoccupied period at nights. As for the DCV model, VAV model setup was applied and also a variation was allowed for the airflow from the minimum legal of 0,35 l/(s·m²) to a maximum of the design value for each zone. In addition, the schedule of the people presence was used considering a daily occupancy profile based on (Johansson 2005), occupancy rate of 50%. The energy modelling was done in a dynamical simulation tool IDA ICE (EQUA 2014).

It is particularly important to split the energy saving between ventilation (fans), heating and cooling energy, since fans are always powered by electric source, in comparison with the thermal capacity that can be produced in several ways. In the analysed system both fans and thermal are powered by electric sources respectively directly and by heat pump (HP). Now, if the thermal capacity was produced differently, e.g. district heating and chiller (DH–CH), the percentage of energy reduction for the aim, ventilation, heating or cooling was constant, but the differences in the production will affect the economical evaluation because of the different efficiency of the production and price for the different sources.

Energy production scenarios, economic evaluation and energy prices are used in evaluation of two scenarios:

- Installed HP: ground coupled double effect heat pump with COP = 4,85 and EER = 3,85. In that case the only energy entering in the building is the electric source.
- Scenario DH–CH: heating source is district heating (DH) and 28 kW air-water chiller (CH) with a EER = 3,25 measured for an outdoor temperature of 35°C.

The electric energy price (epp.eurostat.ec.europa.eu 2014) and for the district heating price (Falkenberg-energi 2014) are reported in **Table 3** with the exchange ratio €/SEK = 8,874 (Google 2014).

Results

Evaluation of energy consumption in different ventilation systems show a progressive decrease of the energy used for the entire conditioning system due to the improvement of the ventilation system only. Looking at energy results the energy can be split based on the components usage such as energy for ventilation (and air circulation), heating and cooling, everything in electric

Table 2. Simulation scenarios.

Name	Description
Scenario 1: CAV	Actual state of the building
Scenario 2: VAV	CAV system with night mode switched on
Scenario 3: DCV	Demand controlled ventilation system with 50% occupancy rate and the night mode switched on
Scenario 4: DCV Class 1	DCV with zones' set point for reach the Class 1 building

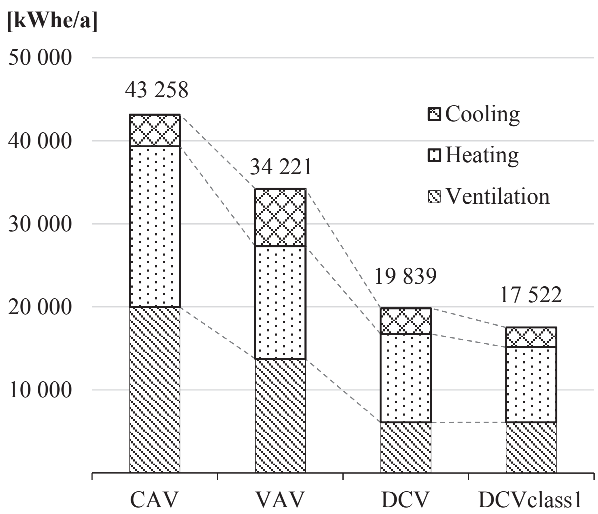

Figure 2. Overall annual simulated energy consumption for different systems´ scenarios.

Table 3. Energy price for electricity and district heating.

Electricity		District heating					
		Price category	Annual consumption	Fix costs		Variable costs	
[€/kWh$_e$]	[SEK/kWh$_e$]		[MWh$_t$/a]	[€/a]	[SEK/a]	[€/kWh$_t$]	[SEK/kWh$_t$]
0.135	1.2	(1)	80	2,569	22,794	0.08	0.70
		(2)	193	5,586	49,570	0.08	0.70

kWh per year. In total an overall energy reduction of 54% can be seen in the comparison DCV–CAV and of 42% in DCV–VAV. The largest energy reduction can be seen for ventilation, with a reduction of 56% (from VAV to DCV) and the impressive reduction of 70% (from CAV to DCV). When adjusting the indoor temperature in order to achieve the Class 1, an additional reduction of 12% can be seen from DCV to DCV Class 1 showing the importance of the correct regulation to achieve energy saving. The split of the overall consumption in heating, cooling and ventilation (energy for the air circulation) is fundamental for the economic analysis and the evaluation of the payback period in the installation of a ventilation system instead of another.

Comparison between installed HP and scenario (DH–CH) in operating cost shows the additional investment cost of a DCV system compared to a CAV system is approximately 12% and only 8% more compared to a VAV system (Errico 2014). HVAC operating costs play large factor in operating of a building, and it is based on expected modelled energy consumption for various ventilation systems based on partial costs (ventilation, heating and cooling), efficiency of production methods (COP for heat pump and EER for heat pump with EER for air-water chiller) and respective energy prices (for district heating and electricity). For the heating and cooling costs (**Figure 3**) large differences can be seen especially for the heating expenses. Moreover, if in both the scenarios the cooling operating costs rise and drops by exactly the same percentage (+81%, -55%, 23,6%) following the upgrade from CAV to DCV Class 1, the percentage of reduction in the heating expenses are not coupled because of the construction of the price of the thermal energy supplied by the district heating.

In **Figure 3** it is interesting to see how much the internal set points in the same installation affect the heating operating cost (DCV → DCV Class 1). Usually, to keep a higher comfort level inside a building, it is necessary to use more energy and therefore create higher costs. Here we can see an energy reduction with decrease of operating costs. Because of that in the process of optimization from CAV to DCV Class 1 there is an additional final overall operating cost reduction from DCV to DCV Class 1 of 12% in installed HP case and of 9% in scenario (DH–CH). This additional reduction in the operating cost, achieved only by setting the correct internal temperature set points, highlights the clear importance that the right indoor temperature set points have on operating cost in systems.

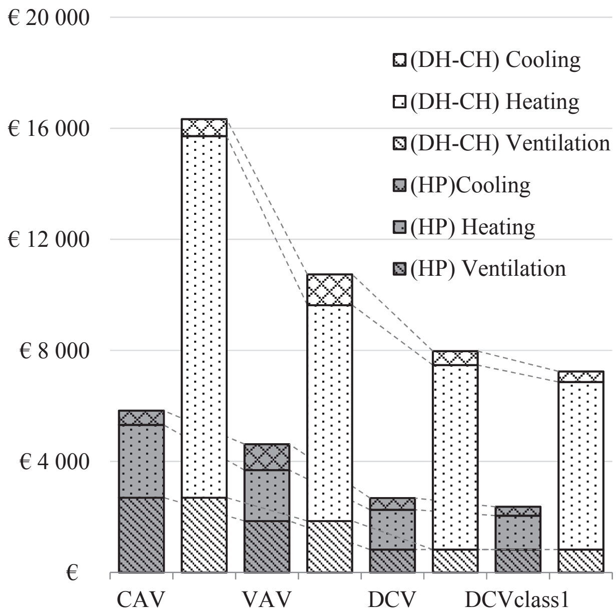

Figure 3. Comparison between the production scenarios (HP; DH–CH) in the different system scenarios CAV, VAV, DCV and DCV Class 1.

Comparison between real installed HP and scenario (DH–CH) in regards of payback period (**Figure 4**) it can be seen that with a scenario (DH–CH) instead of the installed HP there is evidence in the reduction of the payback period. Reduction that amounts approxi-

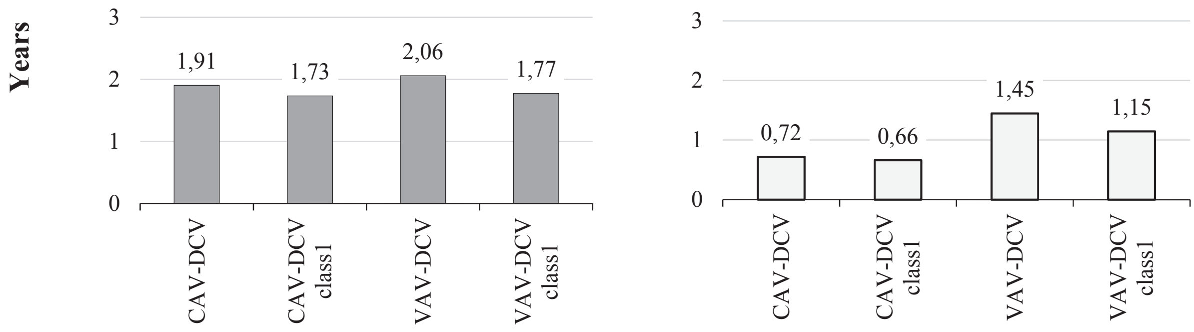

Figure 4. Payback period: installed HP and theoretical scenario (DH–CH).

mately to 60% in the system improvement scenarios from CAV to DCV and from CAV to DCV Class 1 and approximately to 30% in both VAV to DCV and VAV to DCV Class 1. In all the analyses it appears that the profitability of a DCV system is higher in the scenario (DH–CH) compared to the installed HP case. This consideration leads to think that the DCV system is more suitable if the production system is not at the top of the efficiency class. In both considered production scenarios (HP and DH–CH), the economical convenience is more remarkable in case of a system improvement scenario CAV–DCV instead of a system improvement scenario VAV–DCV.

Discussion and conclusions

The comparison of the CAV, VAV and DCV systems shows an energy reduction of over 50%, in the best case (from CAV to DCV) and will reach and keep the indoor temperature set points. It is important to state that this reduction is achieved simulating a conservative occupancy rate equalling to 50% (note that occupancy rate is around 28–30% in a typical office and around 25% occupancy inside a typical retail building). That means that the energy saving achievable with a DCV system could be even larger than the modelled one in this work. The energy to reach the indoor temperature set points cannot be separated from an economic assessment that has to evaluate the expense for the purchased energy.

The first production system is represented by the real installed ground coupled heat pump (HP), the second, based on a Swedish situation, is composed by district heating and chiller (DH–CH). With the HP the electric energy purchased is converted in thermal for heating passing through the COP and in the cooling energy trough the EER of the heat pump (constant EERs). In the DH–CH case the heating energy was considered to be used directly as purchased from the district heating distribution net, the cooling energy was considered supplied from the chiller passing through its EER, also here considered as constant. It is easy to understand that the HP system is more efficient than the DH–CH. Even though based on the presented results it can be stated that the payback period is shorter if the installed production system is inefficient, yet in both production scenarios the payback periods are rather short, in most of the system improvement scenarios less than two years. Because of that, considering that the additional investment is a small value compared to the overall cost, about 10%, and the short payback period, it can be stated that the choice of a DCV system instead of a CAV or VAV ventilation system is always economically profitable to ensure the good climate in buildings, not only in new buildings but also in retrofits. ■

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Acoustic evaluation of floating floors with housekeeping pads

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Mechanical and electrical equipment rooms are one of the main sources of noise and vibration in buildings. In high-rise buildings, it is usually inevitable to locate equipment rooms in mid-floors rather than placing them far from noise sensitive areas such as basements or separate structures. The noise from the mechanical equipment such as chillers, circulation pumps, and air handling units in these spaces can travel via and through structure to adjacent occupant spaces. Structure-born noise from the machinery excitation transmitted as impact sound and vibration can be isolated by choosing proper vibration isolators. Yet, air-borne and flanking noise transmission from the flooring should still be carefully treated. Installing a floating floor provides high levels of air-borne and flanking sound reduction in such cases. A floating floor is either constructed by using an air gap or a resilient layer. Spring or rubber type mounts are utilized to provide an air gap. A composite sound transmission loss value for such types of floating floor applications are calculated and presented in this paper.

Keywords: sound transmission, housekeeping pads, floating floors, sound insulation, equipment noise.

Introduction

High rise multistory buildings involving concrete and steel frames are embarked in many countries. Increase in sound insulation performance requirements result in cost oriented and technically practical solutions [1]. The most effective noise control measure is to locate indoor technical rooms as far away as possible from noise-sensitive areas. However, mechanical equipment rooms in high-rise multistory buildings are typically located on intermediate floors, close to the occupied areas they serve. In such cases, appropriate constructive layers should be selected for walls, ceilings, and floors once the amount of noise is assessed within the mechanical equipment rooms. For floorings, floating concrete floors are usually required to separate mechanical spaces from noise-sensitive spaces that are below the mechanical room [2].

Floating floor is a technical term which implies that the flooring is separated from the structure so that it has no rigid connection with surrounding building elements such as walls, floors and columns. This is achieved by using various insulation materials such as rubber mount isolators, resilient layers, flanking bands and strips. As a term, floating floor may refer to various floor isolation methodologies that can be adopted by using these products. Floorings raised on steel constructions in data centers and laminate parquet floorings installed on resilient layers are also called floating floors, but in our case we will be mostly dealing with concrete slabs raised on rubber mounts or springs as used in most mechanical rooms.

Floating Floor Applications in Mechanical Spaces

Insulation for the flooring in mechanical spaces should be chosen according to the equipment type, equipment weight, noise level and adjacent spaces intended purpose of use. Unnecessary and overqualified insulation may result in excess amounts of investment costs. If the main purpose of floor insulation is to overcome impact noise caused by the machinery, then using vibration isolators, resilient layers or rubber pads is probably a better choice since primary objective to install a floating floor with resilient mounts and air cavity is to prevent airborne sound transmission.

Impact noise insulation performance

Floating floors in mechanical rooms are generally not designed as part of a vibration isolation scheme for plant equipment. Floating floors with resilient mounts consist of concrete slab which is completely disconnected from surrounding building elements by vertical flanking strips to separate it from walls and columns, and resilient mounts to support it above the structural floor. The resilient mounts chosen mostly determine the overall impact noise isolation performance of the floating floor application. Assuming an ideal condition in which flanking transmission are neglected and there are no sound bridges, impact sound insulation improvement can be calculated from equation (1).

$$\Delta L_n \approx 10\log_{10}\frac{2.3\rho_{s_1}^2\omega^3\eta_1 c_{L_1} h_1}{ns^2} \qquad (1)$$

where ρ_{s1} is the surface weight, η_1 is the internal loss factor, c_{L_1} is the longitudinal wave velocity, h_1 is the thickness of the floating slab, n is the number of resilient mounts per unit area, and s is the stiffness of the mounts used [3].

It is possible to achieve the same or even better impact sound insulation performance with a similar floating floor construction by using a resilient layer instead of rubber mounts. We can consider this case as a locally reacting floating floor. Thus, we can use the following equations for the calculation of improvement in impact noise insulation performance of a floating floor with a resilient layer under ideal circumstances [3].

$$\Delta L_n \approx 20\log_{10}\left[1+\left(\frac{f}{f_0}\right)^2\right] \qquad (2)$$

where f is the frequency. The natural frequency f_o of the system is

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{s}{\rho_{s1}}} \qquad (3)$$

where s is the dynamic stiffness of the resilient layer and ρ_{s1} is the surface weight of the floating slab [3].

Comparing resilient mounts and resilient layers impact noise performance from properties of the available products in the market, it is clear that we can achieve similar impact noise performances by choosing appropriate products according to their mechanical properties (**Figure 1**). Also, vibration isolation products can be used when the foundation or base of a vibrating machine is to be protected against large unbalanced forces or impulsive forces [4]. However, resilient mounts and elastic underlays performance vary a lot when we are dealing with airborne sound insulation. Even when the so called impact noise, structural vibrations and flanking transmissions are damped by vibration isolation, airborne noise transmission can still be a problem.

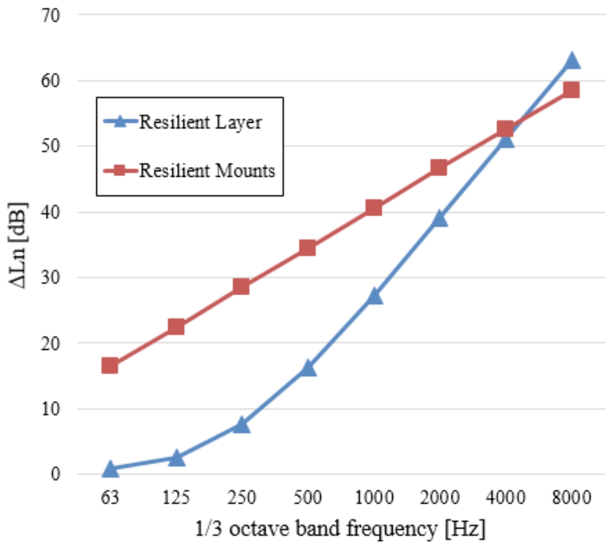

Figure 1. Comparison of improvement in impact noise performances of floating floor systems constructed with a resilient layer and rubber mounts.

Airborne noise insulation performance

The heavy equipment should be properly supported to account for additional loads such as seismic loads [5]. Therefore, heavy equipment such as a chiller is usually fastened to a housekeeping pad which is anchored to the structural load bearing slab in floating floor applications (**Figure 2**). It is possible to overcome impact noise and vibration transmission caused by the machine by using an elastic or resilient member between the machinery and the foundation. The problem is, it is usually questioned whether the floating floor that surrounding the housekeeping pad is doing any good in terms of acoustic insulation since its plinth base already creates a short cut for airborne noise transmission through the cross section of the plinth itself.

Floating floors and housekeeping pads have different sound transmission losses. For the ease of our calculations we adopt Goesel's empirical method of double partitions to predict the floating floors sound transmission loss [6]. Calculating the transmission loss of two constituent single partitions R_I and R_{II} assuming that there are no structure-borne connections, and the gap is filled with porous sound-absorbing material, the airborne sound transmission through the floating floor can be calculated from equation (4).

$$R_{FF} \cong R_I + R_{II} + 20\log_{10}\left[\frac{4\pi f \rho_0 c_0}{s}\right] \quad (4)$$

where ρ_o is the density and c_o is the speed of sound in air trapped in between the gap, s is the dynamic stiffness per unit area of the gap, d is the gap thickness, and R_{FF} is the overall sound reduction performance of the floating floor system.

To calculate isotropic single layered structures sound reduction performances, calculation method described in EN 12354: Annex B is adopted [7]. Assuming that floating floor and plinth structure are exposed to the same average sound intensity on the source side, we can calculate the composite transmission loss from equation (5).

$$TL_c = -10\log_{10}\left(\frac{S_{FF}\times 10^{-R_{FF}/10} + S_P \times 10^{-R_P/10}}{S_{FF}+S_P}\right) \quad (5)$$

where S_{FF} is the surface area of floating floor system, S_P is the surface area of the plinth structure, and R_P is the sound transmission loss of plinth base structure.

Contribution of Housekeeping Pad to Sound Transmission

We evaluate a mechanical room with the equipment described above installed within. We consider a single rigid base of plinth structure made of concrete with a height of 400 mm which is surrounded by a floating concrete slab of 100 mm. Load-bearing concrete slab has a 200 mm thickness as usual in most mechanical spaces and the air gap between the floating slab and the load-bearing concrete slab is considered to be 50 mm. Composite transmission loss is calculated according to the method described for different surface area of housekeeping pad for a fixed area of 200 m² mechanical space.

As it appears, there is a considerable difference between the insulation performance of a whole floating floor and a floating floor that encloses a housekeeping pad (**Figure 3**).

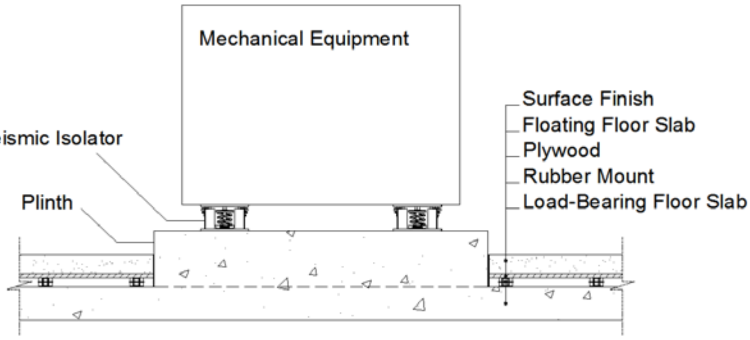

Figure 2. ASHRAE compatible floating floor design for heavy equipment.

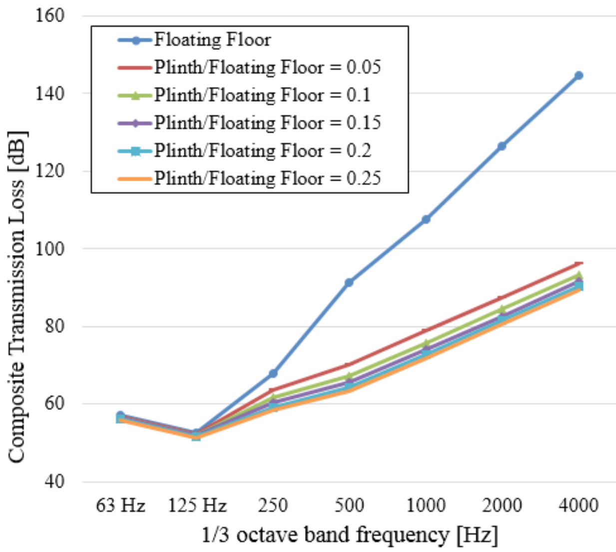

Figure 3. Change in composite transmission loss for varying surface area of housekeeping pad to a fixed 200 m² surface area of a floating floor system.

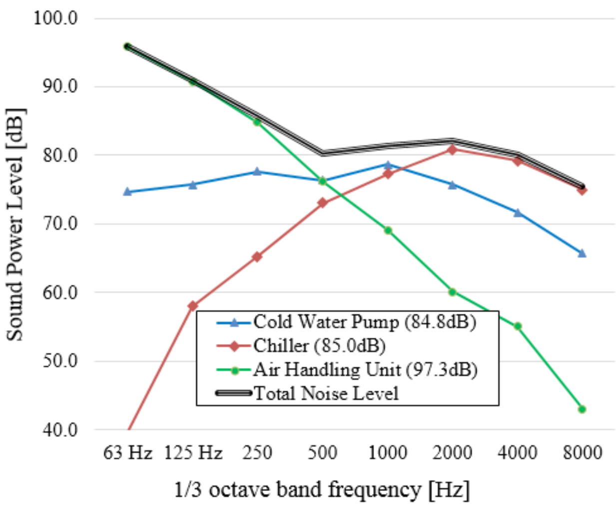

Figure 4. Sound power levels of cooling equipment in a mechanical room.

However, once a rigid base of plinth structure is built within a mechanical space, increasing the surface area of the plinth base does not affect insulation performance in a significant way. At this point, the question is whether the performance of floating floors that enclose a plinth base structure is efficient under real working conditions.

Most equipment manufacturers give single value representations of their products noise levels. Unfortunately, we have to work on broad band – or at least one-third octave band – responses of the relevant machinery to design a working isolation system. Therefore, if it is not possible to make measurements on site, having an archive of measurement results of spectral noise levels of common machinery can be an advantage to start with a reasonable design. As an example, we consider a cooling room with a cold water pump, a chiller and an air handling unit (**Figure 4**). Even though spectral noise characteristics of these three units vary, their combination gives us a flatter response.

Assuming that the total noise within the mechanical space is transmitted through the flooring to an adjacent space, the difference between the total sound power level (SWL) of the equipment and the composite transmission loss of flooring gives us an idea of sound insulation performance of various floating floor systems with and without housekeeping pads. As expected, having a monolithic floating floor is advantageous. Existence of a housekeeping pad causes an increase in noise between 500 Hz and 1 kHz. However, the change of housekeeping pad surface area does not affect noise transmission dramatically (**Figure 5**).

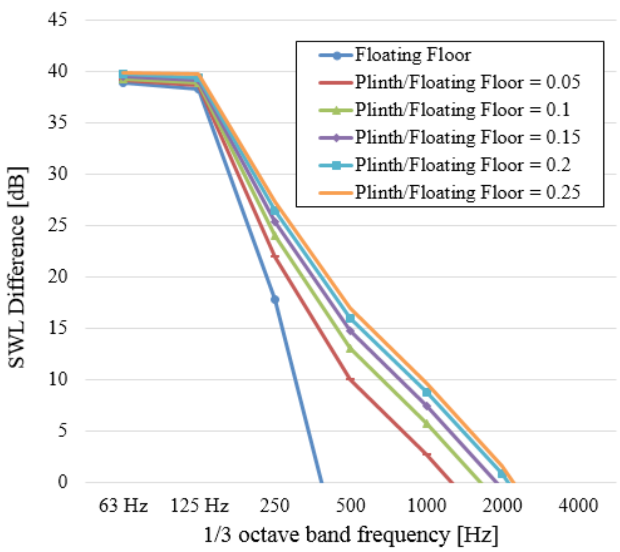

Figure 5. Insulation performance comparison between various floating floor systems.

Conclusion

The plinth structures contribution to airborne sound insulation is investigated and with some simple calculations available in literature it has been found that - for a realistic case - the contribution of housekeeping pads to noise transmission is mostly in between 500 Hz and 1 kHz. Presence of a housekeeping pad causes a conspicuous increase in noise transmission compared to a monolithic floating floor design. However, if a floating floor design is made considering the equipment noise levels from the beginning and appropriate slab thicknesses and insulation materials are chosen, it is expected that the transmission loss should not vary much according to the changing plinth base surface area. For future work, further analysis and a more detailed model should be developed to investigate floating floors with plinth base structures. It is recommended to investigate more about such composite structures contribution to airborne and impact noise transmission especially in mechanical spaces. ■

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United nations conference on climate change
COP21/CMP11

COP21 – Decarbonising the build environment by 2050

Aftermath of COP21

After many years of disappointing UN climate change negotiations COP21 delivered a historic, ambitious and balanced agreement. This is the first major multilateral deal of the 21st century setting out a global action plan to put the world on track to avoid dangerous climate change by limiting global warming to well below 2°C. The European Union has played an important role in brokering the agreement in Paris, where 195 countries adopted a new universal, legally binding global climate deal. Now it's time for every stakeholder to take concrete actions from global to local level to ensure that the Paris agreement is implemented in real life in the coming decades.

COP21 has crossed many thresholds needed for the planning and implementation of global climate change actions. One of them is that it gave a clear message that from the technology point of view the smooth transition to near-zero emissions in buildings is feasible. A key message of the COP21 side events was to feature available solutions enabling the achievement of the ambitious zero emission targets. Another novelty of COP21 was that for the first time ever the energy sector – including among others buildings energy efficiency and construction sector – was involved in the UN climate discourse. This approach was followed by many side events organized in Paris during the two weeks of negotiation.

Symposium "Zero Emissions by 2050: How to decarbonise the built environment"

9 December 2015 a symposium was organised on the decarbonisation of the build environment by 2050. The COP21 side event was initiated by the Edward Mazria, the internationally recognized architect and founder of Architecture 2030, a think tank aiming at rapidly transforming the building sector from a major contributor of greenhouse-gas emissions to a central part of the solution to the global-warming crisis. The event was hosted by the French Building Federation and co-organised by global architecture, construction and urban planning NGO-s.

After setting the scene by a high level panel of World Bank, World Economic Forum and the Global Word Building Council the symposium was structured in 3 sessions featuring flagship initiative panels in the domains:

1. City/District Building Initiatives
2. Building Design and Planning Tools
3. Innovative Building Sector Financing

Building sector – the elephant in the room

The panellists stressed that the decarbonisation of the built environment is a key in the climate process and that construction sector innovation can accelerate the process even more. The event focused on cities and urban environment where the importance of the building sector's decarbonisation is even more obvious. The first panel presented the global challenges faced by the cities around the world and especially in China and North America. By 2030 urban population will increase by 1.1 billion people, while China and North-America will be in charge of 38% of the global CO2 emission by 2050. The elephant in the room are buildings, which have a massive share within the total emissions in urban areas. As an example 73% of New York's CO2 emission comes from buildings. The solutions and answers to this challenge have to be global with cities as a priority. The Green Building Council announced its global commitment to deep refurbish 2.1M m2 of buildings by 2030 and train construction sector workers to be able to answer the demand. 3 GBC chapters from Canada, Africa and Australia launched a Net zero certification scheme for buildings.

The panellists agreed that the only ways to force the market to invest in the decarbonisation of buildings are the prescriptive codes and mandatory performance requirements that are incentivized. The good news is that the change has started. By 2016 building codes in the world became by 50% stricter compared to the previous decades. Even China is committed to introduce stricter building codes and launch investments to comply with them in the big cities was confirmed by the Chen Zhen, the Secretary-General of the China Exploration and Design Association Architecture Branch (CEDAAB).

Hal Harvey, CEO of Energy Innovation, a US think tank promoting sustainable urban planning, presented their excellent guidelines elaborated for sustainable urban planning for metropoles in China, while Brett Phillips, Co-Founder and Chairman of the Seattle 2030 District project presented an innovative and interpectoral local initiative for the decarbonisation and sustainable development of Seattle, Washington. The speakers pointed out that achieving 80% CO2 emission reduction via buildings in urban areas is not a technical, neither a financial challenge. It's a practice challenge requiring training, skilled professionals and smart / innovative practice. And it also requires cooperation between the different involved public and private sector stakeholders.

In the second panel IT companies presented innovative building design and planning tools providing architects and designers with tools to design high energy performing buildings and energy refurbishments. The event was closed by a panel presenting innovative building sector financing tools.

The symposium was an excellent example of the novel dialog between UN stakeholders and the building energy efficiency sector as it brought together the high level UN climate change community and environmental policy makers with energy policies and the related sectors such as buildings & construction industry, architects, designers, public and private finance sector, NGO-s and technology providers in the same discussion. It also highlighted flagship initiatives that are on the way to decarbonise major cities by 2015. This is the way we need to go to successfully implement the COP21 political agreement in practice.

ANITA DERJANECZ
REHVA Policy and Project Officer

Nordbygg is the platform we are using to enter the Swedish market

Nordbygg 2014 marked the start of our big push to enter the Swedish professional construction market. Our participation was successful. At Nordbygg 2016 we will follow up by further establishing our position and launching new, sustainable products.

Jonas Brennwald, Regional Vice President North-West Europe at Grohe, views Nordbygg as the platform for the company's major push to enter Sweden.

A lot of interest in the right target groups

"There was a lot of interest and we made some good contacts in our most important target groups – architects, specifiers and installers. We successfully spread our message that Grohe is one of the top brands in the world for faucets and sanitary fittings for the bathroom and kitchen and that we had fully adapted our products for the Swedish market," says Jonas Brennwald.

After Nordbygg 2014 Grohe successfully marketed its water and energy calculator, which helps property owners monitor and influence their water and energy consumption.

"We also started collaboration with Bravida, the leading installation and service company, which will now be launched," explains Jonas Brennwald.

Positioning and new products at Nordbygg 2016

At Nordbygg 2016 Grohe will have a large stand to take its investment in the Swedish market to the next level.

"Next time we will focus on cementing our position as a global market leader with a long list of new products that have been adapted to the Swedish market. Our design is an important factor, but it is primarily innovations in sustainability that will be in focus," explains Jonas Brennwald.

Grohe has a very large in-house development department and invests a lot of resources into designing fittings and systems that optimize the use of water and energy in the bathroom and kitchen.

"There is an inherent conservativeness in this area that we want to challenge. At Nordbygg 2016 we will show that Grohe is constantly creating new, innovative and sustainable products and we will demonstrate their potential," says Jonas Brennwald.

Nordbygg at a glance

Nordbygg is held in March/April every even year at Stockholmsmässan and attracts around 900 exhibitors and 53,000 visitors. Next event will take place 5-8 of April in 2016. Nordbygg is organized in partnership with leading associations in the Swedish industry. We also collaborate with a number of other industry organizations in various projects related to Nordbygg.

Nordbygg has the position as the Nordic region´s largest and most dynamic meeting place for the construction and real estate industry, offering new knowledge and business opportunities. The event offers everything from discussions about construction conditions and urban planning to products and services for property production, maintenance and management.

FOR THERM – 7th trade fair for heating, alternative sources of energy and air conditioning

20-24 September 2016
Prague – PVA EXPO PRAGUE,
Czech Republic

FOR THERM is an annual Czech trade fair which is held in Prague together with four concurrent fairs – FOR ARCH, FOR WOOD, FOR STAV and BAZÉNY, SAUNY & SPA. This set of construction-related fairs is the single most attended and most important building event in the Czech Republic. In 2015, more than **74,000 visitors** have attended and **830 exhibitors** from **13 different countries** have participated in this exhibition. Due to its location in Prague, an important center of Central-European business, FOR THERM is a unique opportunity for both domestic and foreign exhibitors to present their services and products within an international competition. If you want to address the Czech region, FOR THERM is the one place you need to meet new partners, dealers, suppliers and customers.

The event is organized under the auspices of the President of the Czech Republic, REHVA and the Association of Building Entrepreneurs of the Czech Republic. A rich program will accompany the trade fair.

Webpage: www.for-therm.cz/en

The Cold Climate 2015 conference and the industrial ventilation conferences in China during 19-28.10.2015

This article summarizes the two international conferences: the 8th International Cold Climate HVAC (heating, ventilation and air-condition) Conference (Cold Climate 2015) (October 21~23, 2015) in the seaside city of Dalian, China and the 11th International Conference on Industrial Ventilation (26-28.10.2015) in Shanghai, China. The Cold Climate conference attracted 180 participants from 12 countries and the Industrial Ventilation conferences attracted 210 participants from 15 countries.

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Norwegian University of Science and Technology, Norway

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The 8th International Cold Climate HVAC Conference (21-23.10.2015), Dalian, China

The Cold Climate conference is a series conference initiated by the Scandinavian Federation of Heating, Ventilation and Sanitary Engineering Associations (SCANVAC) in 1994. This year's conference was organized by Dalian University of Technology, Tsinghua University and Technical Research Centre of Finland (VTT). The theme of this conference is sustainable buildings and the energy utilization in cold climate. There were altogether about 180 participants in the conference. They came from three continents (Europe, America and Asia): 12 countries (Denmark, Finland, Norway, Sweden, German, Russia, the Republic of Kazakhstan, USA, Canada, Korea, Japan and China). All participants represented 70 different universities and institutions around the world, thirty percent of the delegates were from overseas. The conference received 179 papers and 145 of them were finally accepted.

Photo 1. The SCANVAC President, Per Rasmussen, gave a speech in the opening ceremony on 21.10.2015.

In total 113 papers were presented by oral presentation and 18 by posters. In the conference, a REHVA workshop, which was purposed to disseminate the new REHVA Guidebook - mixing ventilation, was organized by Risto Kosonen.

Photo 1 shows that the SCANVAC President, Per Rasmussen, gave a speech in the opening ceremony on 21.10.2015 and Photo 2 shows that he was in the Workshop of Indoor climate design for nearly Zero Energy Buildings (NZEB) in cold climate.

The 11th International Conference on Industrial Ventilation (26-28.10.2015) in Shanghai, China

The 11th International Conference on Industrial Ventilation (Ventilation 2015) was held on October 26 to 28, 2015 in Shanghai, China. This conference was organized by Tongji University, co-organized by VTT (Technical Research Center of Finland) and Tsinghua University. Prof. Xu Zhang chaired this conference. More than 210 participants originated from 15 different countries/regions attended this triennial event to celebrate the 30 years' anniversary of this international conference series on industrial ventilation. In total 125 papers were peer-reviewed and included in the Proceedings of this conference.

During the opening ceremony, besides Prof. Goodfellow was awarded for his 30 years' service to industrial ventilation society. Meanwhile, Prof. Wei Xu and Prof. Xu Zhang from CCHVAC were awarded by Prof. Karel Kabele, President of REHVA, for their outstanding services to the collaboration between REHVA and CCHVAC (see Photo 3).

The Rehva President, Prof. Karel Kabele, was invited to give keynote speech in the plenary session. The title of his keynote speech was: Ventilation for better IEQ in Nearly Zero Energy Buildings (see Photo 4).

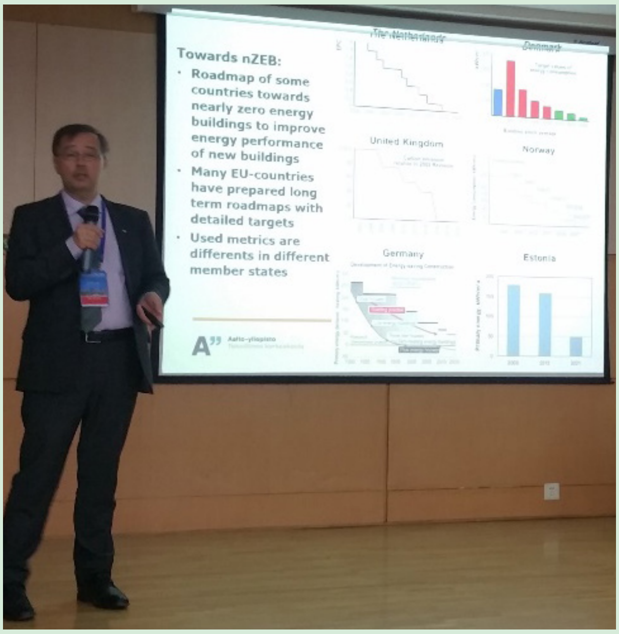

***Photo 2.** Risto Kosonen was presenting the presentation about Nearly Zero Energy Building in the Workshop of Indoor climate design for nearly Zero Energy Buildings (NZEB) in cold climate.*

***Photo 3**. Opening ceremony 26.10.2015: REHVA's award – REHVA's Certificate of Merit to: Prof. Wei Xu, President of CCHVAC and Prof. Xu Zhang, Tongji University, Vice president of CCHVAC.*

***Photo 5**. SCANVAC workshop: Advanced airflow distribution methods for reduction exposure to indoor pollution 27.10.2015.*

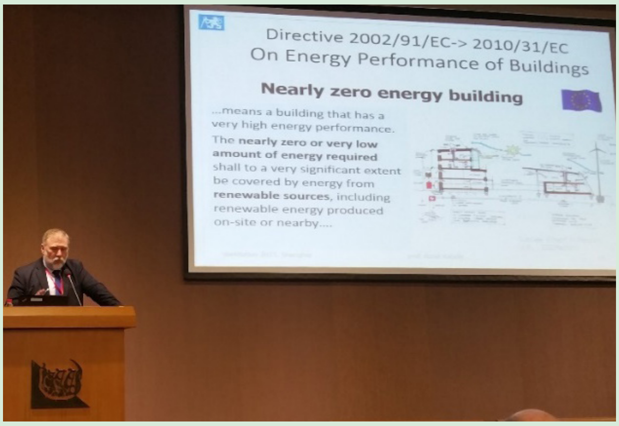

***Photo 4**. Keynote speech from the REHVA President Prof. Karel Kabele in the Industrial Ventilation 2015 conference.*

During the conference, a SCANVAC workshop was organized entitled: Advanced airflow distribution methods for reduction exposure to indoor pollution. Four presentations were made by Prof. Peter Nielsen from Aalborg University, Prof. Arsen Melikov from DTU, Prof. Risto Kosonen from Aalto University and Prof. Guangyu Cao from NTNU (see Photo 5).

Remarks

The two conferences provided wide and solid platforms for experts to share and exchange ideas and insights in developing a healthy, productive, safe and comfort indoor environment in an energy efficient manner. The Cold Climate conference in Dalian included a few main topics including zero energy building, sustainable district heating, renewable energy and low exergy energy utilization. Discussions also covered: whether different countries have different zero energy building roadmaps, how to heat and cool a zero energy building, whether the type of space heating system is the key method to improve the efficiency of CHP system, what is the fourth generation district heating system which is the future trend of district heating system, what can be done in HVAC field facing the era of big data, etc. During the Industrial Ventilation conference in Shanghai, five domains of ventilation science and engineering were covered: 1) occupational health; 2) ventilation and sustainable development; 3) industrial ventilation and pollutant control; 4) air distribution design, measurement and products; and 5) specialized ventilation applications. ■

Halton

Halton Vario – Latest total indoor climate solution

Halton Vario is the latest total indoor climate solution from ventilation products to controls for rooms, zones and at the system level. Halton also verifies the performance of your system so you can be sure that your building's indoor climate is exactly as designed from day one.

Halton claim that Vario chilled beam solution not only can reduce energy consumption by up to 50% but also that this solution can be up to 50% more energy-efficient than conventional air-conditioning systems. For the first time in the history of chilled beam systems, the system monitors space usage, controlling and adjusting cooling and ventilation according to the demand. Thanks to the smart controls the indoor environment conditions are maintained at an optimal level.

Radically lower churn costs with a fully flexible system. Applying Halton Vario solution office spaces can be converted in to a meeting room (and vice versa) in 15minutes or less.

Halton Vario controls ventilation, room air temperature and air quality demand-based to provide "A-class" indoor environmental quality that is specified in regular cited international standards (ISO EN 7730, EN15251 and CR 1752).

Comfortable thermal conditions and indoor air quality create productive working conditions thus users' complaints on indoor conditions are minimal.

This solution not only makes green buildings a reality, it makes them more energy efficient, more flexible to layout changes and more comfortable to work in than ever before.

More information: www.halton.com

Swegon

Great savings with the new generation PARAGON

Besides friendly staff and great location, one of the top priorities of hotel guests is a comfortable room climate – requiring perfect supply of fresh air, the right air temperature and low noise level. As a solution to all these requirements, Swegon has created the PARAGON comfort module, providing the perfect indoor climate in a highly energy efficient way. The solution has proven itself in hotel rooms, as well as nursing homes and hospital rooms, since year 2009 with over 13000 units installed.

The secret behind the PARAGON is to use the air pressure from a centrally placed air handling unit, together with small nozzles inside PARAGON, to create an induction effect, distributing the air along the ceiling. This results in a highly efficient mix of fresh air and room air, which together with a waterborne heat exchanger make sure the room is well ventilated at the right temperature, by adding heating or cooling according to demand. With the PARAGON solution, there are no noisy fans, condensation problems or cold draught in the room.

Now the new generation of PARAGON is here, with a unique Compact Change Over-valve integrated in the unit, increasing each unit's cooling capacity about 20%, and the heating capacity about 60%.

Product Manager Jonas Åkesson sees a great potential in the new product generation; "The gain in capacity can be used in several ways, one is to adjust the cooling and heating water temperatures, allowing for a more economical operating mode of the heat pump or chiller. For a typical heat pump, this may convert into an electrical energy saving of about 25%."

Jonas Åkesson also points out the advantages from a building process perspective; "PARAGON's increased capacity makes product dimensioning easier and provides a more robust system design, without any compromises concerning the indoor climate. In the end this makes PARAGON a crucial factor for hotel guest satisfaction and retention."

More information: http://www.swegon.com/

Eu.bac certifies Belimo actuator/valve combination

LR24ALON fitted on a 6-way zone valve.

The French CSTB test lab accredited by eu.bac has tested the control accuracy (CA value) of the LR24ALON rotary actuator with integrated single room controller in combination with the 6-way zone valve in accordance with European standard EN15500 and awarded it the eu.bac certificate and AA energy efficiency label.

Control accuracy test

The control accuracy was tested in the heating/cooling ceiling application intended for a 6-way zone valve and achieved the CA values of 0.3K for the heating sequence and 0.2K for the cooling sequence.

Certification process

The actual application test in the lab also involves the manufacturer (licence holder) being assessed by the accredited eu.bac test lab. On 16 July 2015 the head office of BELIMO Automation AG in Hinwil/Switzerland was audited by CSTB. The assessment of the relevant production line played a key role in this.

Direct connection and integrated single room controller

The FTT-10A transceiver fitted in the LonMark®-certified LR24ALON actuator permits a direct connection to LonWorks®. A single room controller (Functional Profile LonMark® #8060) is integrated into the actuator in order to reduce the costs of implementation of individual room controls. Thanks to an additional output variable, the 6-way zone valves fitted in the heating/cooling ceilings can be controlled directly by the LR24ALON. Additional controller versions, for CO_2 for example, can be implemented on request.

Free LNS plug-ins available on Belimo website

Free-of-charge LNS plug-ins can be downloaded from the Download Center at www.belimo.ch for the integration of LON actuators in LonWorks® systems. These work with common binding tools based on LNS.

Heating/cooling ceiling application

Both the heating and cooling control sequences can be activated with just one 6-way zone valve from Belimo (**Fig. 1**). As a result, just one actuator is needed for this assembly. This means fewer cables and data points and less installation work.

The system diagram (**Fig. 2**) shows a heating/cooling ceiling with room temperature sensor. Window contact, occupancy switch and dew point monitor are integrated directly by means of LonWorks®. The room temperature is controlled via the single room controller integrated in the LR24ALON actuator.

More information: www.belimo.eu

Grégory Picard, Certification Manager CSTB (2nd from left) with the production line managers at BELIMO Automation AG head office, Hinwil/Switzerland.

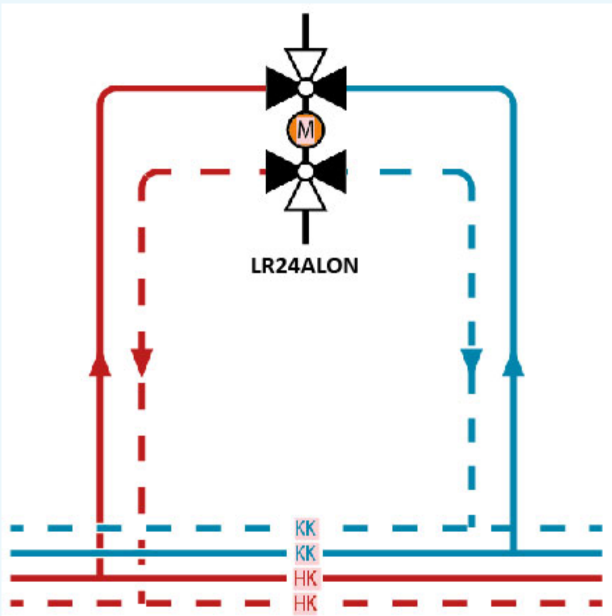

Fig.1

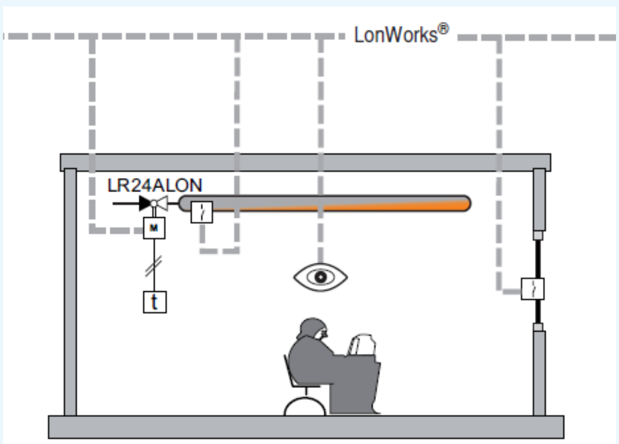

Fig.2

VDI- Standards published in January and February 2016

VDI 3805 part 17 (2016-01). Product data exchange in the building services; Drinking water system assemblies

Based on VDI 3805 Part 1, the standard describes a manufacturer and IT system independent and unified data format for the exchange of product data for drinking water system assemblies used in building services. German / English language.

VDI 3805 part 2 (2016-01). Product data exchange in the building services; Heating value assemblies

Based on VDI 3805 Part 1, the standard describes a manufacturer and IT system independent and unified data format for the exchange of product data for heating value assemblies used in building services. German/ English language.

D VDI 3805 part 23 *(Draft)* (2016-01). Product data exchange in the building services; Ventilation devices for flats

Based on VDI 3805 Part 1, the standard describes a manufacturer and IT system independent and unified data format for the exchange of product data for ventilation devices for flats. Only in German available.

VDI 6012 part 1.2 (2016-01). Integration of distributed and renewables-based energy systems in buildings; Fundamentals; System selection

This standard provides a method that can be used to determine which combination of regular, regenerative and/or possibly fossil-fueled energy systems could be installed in new or existing buildings. Included Excel spreadsheets facilitate the application of the described method. German / English language.

VDI 6012 part 1.4 (2016-01). Integration of distributed and renewables-based energy systems in buildings; Fundamentals; Fixing of solar modules and solar collectors on buildings

The use of solar energy is increasingly an integral part of energy generation. Both for photovoltaics and solar heating many systems are available that in part greatly differ in appearance and design. A viable and reliable mounting of modules and panels on buildings is indispensable required and creates inter alia, the preconditions for a desired long period of operation of these systems. Different connection and fastening systems are used for this purpose in practice. By attaching solar panels on buildings, the modules and panels and the mounting systems are exposed to external influences, such as force effects caused by wind, snow and temperature fluctuations, or other weather conditions. Moreover, the existing structurally optimised building structure and its protective functions, e.g. from rain or fire, may be effected by individual attachment means. The standard provides help on a professional and proper construction and gives advice for a proper selection of available mounting systems and mounting means and, therefore, helps to ensure the intended design and use of a building and the operation of the solar system to be used. German / English language.

VDI 6022 part 6 BER (2016-01). Ventilation and indoor-air quality; Air humidification on decentralised devices; Planning, construction, operation, maintenance

This standard applies to standalone units for the intended and local humidification of air as well as for decorative water-carrying devices (such as fountains, cascades and water walls) which affect the air humidity in a room. Units originally intended for residential use, which are used in workplaces, are also subject to the requirements specified by this standard. The standard factors the particular hazards incurring from such units arising from, e.g., the supply of unfiltered microbiologically contaminated breathing air and insufficient maintenance. German / English language.

D VDI 3805 part 100 *(Draft)* (2016-02). Product data exchange in the building services; Systems

Based on VDI 3805 Part 1, the standard describes a manufacturer and IT system independent and unified data format for the exchange of product data for systems. Only in German available.

VDI 6210 part 1 (2016-02). Demolition of civil constructions and technical facilities

The standard defines the procedures and assessment criteria for the planning and execution of demolitions of civil constructions and technical facilities for all involved personnel. It applies to the demolition of stationary and non-stationary variable structural and technical systems. The standard describes the planning, implementation and follow-up of such work as well as retrieving, deployment, (interim) storage, treatment and handling of the accumulating materials and waste. It does not include the requirements imposed on the reuse of resulting materials or the recycling or the disposal of waste. German / English language.


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Events in 2016

Conferences and seminars 2016

Jan 23-27	ASHRAE Winter Conference	Orlando, Florida, USA	www.ashrae.org/orlando
Feb 24-26	World Sustainable Energy Days	Wels, Austria	www.wsed.at/en/world-sustainable-energy-days/
March 8-11	Sustainable Built Environment - SBE 2016	Hamburg, Germany	www.sbe16hamburg.org
March 16-18	9th International Conference Improving Energy Efficiency in Commercial Buildings and Smart Communities (IEECB&SC'16)	Frankfurt, Germany	http://iet.jrc.ec.europa.eu/energyefficiency/node/9096
March 31-April 2	12th International HVAC+R Technology Symposium	Istanbul, Turkey	www.ttmd.org.tr/sempozyum2016/eng/
May 22-25	12th REHVA World Conference - CLIMA 2016	Aalborg, Denmark	www.clima2016.org
May 30-June 3	CIB World Building Congress 2016 Intelligent built environment for life	Tampere, Finland	http://wbc16.com
June 22-24	Central Europe towards Sustainable Building Prague 2016	Prague, Czech Republic	www.cesb.cz
July 3-8	Indoor Air 2016	Ghent, Belgium	www.indoorair2016.org
August 21-24	12th IIR Natural Working Fluids Conference	Edinburgh, United Kingdom	www.ior.org.uk
September 21-23	International Conference on Solar Technologies & Hybrid Mini Grids to improve energy access	Frankfurt, Germany	www.energy-access.eu
October 23-26	IAQVEC 2016: international conference on indoor air quality, ventilation & energy conservation in buildings	Seoul, South Korea	www.iaqvec2016.org

Exhibitions 2016

January 25-27	2016 AHR Expo	Orlando, Florida, USA	www.ahrexpo.com
February 2-5	Aqua-Therm Moscow	Moscow, Russia	www.aquatherm-moscow.ru/en
February 24-26	Aqua-Therm Novosibirsk	Novosibirsk, Russia	http://www.aquatherm-novosibirsk.ru/en
March 1-4	AQUATHERM Prague	Prague, Czech Republic	www.aquatherm-praha.com/en/
March 13-18	Light and Building	Frankfurt, Germany	http://ish.messefrankfurt.com
March 15-18	Mostra Convegno Expocomfort	Milan, Italy	www.mcexpocomfort.it/
April 5-8	Nordbygg	Stockholm, Sweden	www.nordbygg.se
April 20-22	Aqua-Therm St-Petersburg	St-Petersburg, Russia	www.aquatherm-spb.com/en
May 4-7	ISK-SODEX 2016	Istanbul, Turkey	www.sodex.com.tr/
October 12-14	FinnBuild	Helsinki, Finland	www.messukeskus.com/Sites1/FinnBuild/

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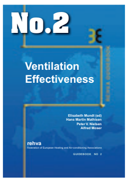
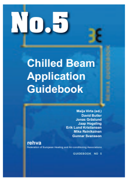
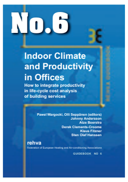


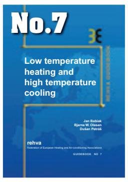
Ventilation Effectiveness. Improving the ventilation effectiveness allows the indoor air quality to be significantly enhanced without the need for higher air changes in the building, thereby avoiding the higher costs and energy consumption associated with increasing the ventilation rates. This Guidebook provides easy-to-understand descriptions of the indices used to measure the performance of a ventilation system and which indices to use in different cases.



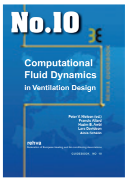
Chilled Beam Cooling. Chilled beam systems are primarily used for cooling and ventilation in spaces, which appreciate good indoor environmental quality and individual space control. Active chilled beams are connected to the ventilation ductwork, high temperature cold water, and when desired, low temperature hot water system. Primary air supply induces room air to be recirculated through the heat exchanger of the chilled beam. In order to cool or heat the room either cold or warm water is cycled through the heat exchanger.



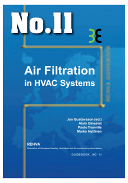
Indoor Climate and Productivity in Offices. This Guidebook shows how to quantify the effects of indoor environment on office work and also how to include these effects in the calculation of building costs. Such calculations have not been performed previously, because very little data has been available. The quantitative relationships presented in this Guidebook can be used to calculate the costs and benefits of running and operating the building.



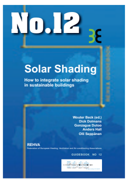
Low Temperature Heating And High Temperature Cooling. This Guidebook describes the systems that use water as heat-carrier and when the heat exchange within the conditioned space is more than 50% radiant. Embedded systems insulated from the main building structure (floor, wall and ceiling) are used in all types of buildings and work with heat carriers at low temperatures for heating and relatively high temperature for cooling.



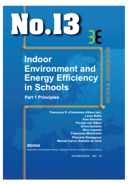
Computational Fluid Dynamics in Ventilation Design. CFD-calculations have been rapidly developed to a powerful tool for the analysis of air pollution distribution in various spaces. However, the user of CFD-calculation should be aware of the basic principles of calculations and specifically the boundary conditions. Computational Fluid Dynamics (CFD) – in Ventilation Design models is written by a working group of highly qualified international experts representing research, consulting and design.



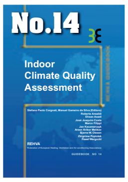
Air Filtration in HVAC Systems. This Guidebook will help the designer and user to understand the background and criteria for air filtration, how to select air filters and avoid problems associated with hygienic and other conditions at operation of air filters. The selection of air filters is based on external conditions such as levels of existing pollutants, indoor air quality and energy efficiency requirements.



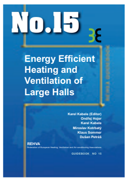
Solar Shading – ***How to integrate solar shading in sustainable buildings.*** Solar Shading Guidebook gives a solid background on the physics of solar radiation and its behaviour in window with solar shading systems. Major focus of the Guidebook is on the effect of solar shading in the use of energy for cooling, heating and lighting. The book gives also practical guidance for selection, installation and operation of solar shading as well as future trends in integration of HVAC-systems with solar control.



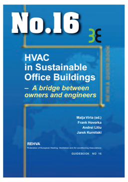
Indoor Environment and Energy Efficiency in Schools – ***Part 1 Principles.*** School buildings represent a significant part of the building stock and also a noteworthy part of the total energy use. Indoor and Energy Efficiency in Schools Guidebook describes the optimal design and operation of schools with respect to low energy cost and performance of the students. It focuses particularly on energy efficient systems for a healthy indoor environment.



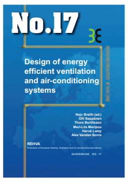
Indoor Climate Quality Assessment. This Guidebook gives building professionals a useful support in the practical measurements and monitoring of the indoor climate in buildings. Wireless technologies for measurement and monitoring have allowed enlarging significantly number of possible applications, especially in existing buildings. The Guidebook illustrates with several cases the instrumentation.



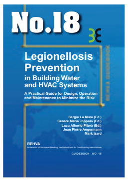
Energy Efficient Heating and Ventilation of Large Halls. This Guidebook is focused on modern methods for design, control and operation of energy efficient heating systems in large spaces and industrial halls. The book deals with thermal comfort, light and dark gas radiant heaters, panel radiant heating, floor heating and industrial air heating systems. Various heating systems are illustrated with case studies. Design principles, methods and modelling tools are presented for various systems.



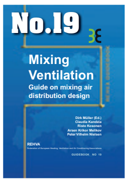
HVAC in Sustainable Office Buildings – ***A bridge between owners and engineers.*** This Guidebook discusses the interaction of sustainability and heating, ventilation and air–conditioning. HVAC technologies used in sustainable buildings are described. This book also provides a list of questions to be asked in various phrases of building's life time. Different case studies of sustainable office buildings are presented.



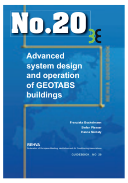
Design of energy efficient ventilation and air-conditioning systems. This Guidebook covers numerous system components of ventilation and air-conditioning systems and shows how they can be improved by applying the latest technology products. Special attention is paid to details, which are often overlooked in the daily design practice, resulting in poor performance of high quality products once they are installed in the building system.



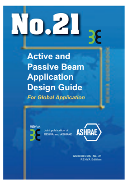
Legionellosis Prevention in Building Water and HVAC Systems. This Guidebook is a practical guide for design, operation and maintenance to minimize the risk of legionellosis in building water and HVAC systems. It is divided into several themes such as: Air conditioning of the air (by water – humidification), Production of hot water for washing (fundamentally but not only hot water for washing) and Evaporative cooling tower.



Mixing Ventilation. In this Guidebook most of the known and used in practice methods for achieving mixing air distribution are discussed. Mixing ventilation has been applied to many different spaces providing fresh air and thermal comfort to the occupants. Today, a design engineer can choose from large selection of air diffusers and exhaust openings.



Advanced system design and operation of GEOTABS buildings. This Guidebook provides comprehensive information on GEOTABS systems. It is intended to support building owners, architects and engineers in an early design stage showing how GEOTABS can be integrated into their building concepts. It also gives many helpful advices from experienced engineers that have designed, built and run GEOTABS systems.



Active and Passive Beam Application Design Guide is the result of collaboration by worldwide experts. It provides energy-efficient methods of cooling, heating, and ventilating indoor areas, especially spaces that require individual zone control and where internal moisture loads are moderate. The systems are simple to operate and maintain. This new guide provides up-to-date tools and advice for designing, commissioning, and operating chilled-beam systems to achieve a determined indoor climate and includes examples of active and passive beam calculations and selections.

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