The Potential of Façade-Integrated Ventilation Systems in Nordic Climate

Advanced decentralised ventilation systems as sustainable alternative to conventional systems

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Summary

The work evaluates the applicability of façade-integrated ventilation systems in a Nordic context.

For this purpose the state of the art of façade-integrated ventilation (FIV) and demands for ventilation system in Norway and criteria for an comprehensive evaluation are identified. In this framework agreements between national requirements and system-specific performance are assessed. The evaluation investigates indoor environment and comfort with focus on aspects of indoor air quality. Energy efficiency and emission efficiency are evaluated by comparison with a centralised ventilation system. Implications of FIV in operation are outlined in respect to usability, maintenance and life cycle costs. Furthermore, aspects of building integration regarding requirements on the built environment and the aptitude of FIV for flexibility and typologies are examined.

The used tools in this work include "ESP-r", "Simien" for dynamic simulation of building performance and energy performance as well as "Ökobau.dat" for the determination of building component related emissions.

The results of the evaluation show that current systems do not comply with all requirements of the Norwegian building code and related regulations. Some aspects need adaptation to local requirements. However, good performance and many possibilities can be expected in other fields e.g. indoor environmental comfort and user satisfaction since advanced principles are exploited.

The technology has an enormous potential. It might be an alternative if there is demand for high expectations on indoor environment and conventional ventilation system are not applicable. The technological limits of façade-integrated ventilation are not reached yet. Possibilities of further development of the concept itself and related technologies are outlined in the work

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1. Introduction

Background

As buildings become more and more thermally super-insulated and airtight the ventilation accounts for an increasing portion of heat loss and energy consumption. Strategies have been developed to decrease the energy demand connected with ventilation. In case of mechanical ventilation systems the focus lies on the optimisation of transported volume, treatment and distribution of air within the system. The discussion about minimised airflow rates and demand-controlled operation is in the centre of attention. (Hegger et al., 2008)

Other concept like hybrid ventilation integrate ´free´ natural ventilation to provide a robust and sustainable strategy. Aim is to benefit from the best of both, mechanical and natural ventilation. Highlighted is the positive response from users, the possibility of individual control and the transparency of the ventilation's response. Less dependency on mechanical systems and increased flexibility while reducing costs are strong motivations in this respect. (Heiselberg, 2002)

Primary goal of most of these concepts is to ensure a comfortable indoor environment with acceptable indoor air quality. Especially in non-domestic buildings high requirements on the indoor air quality meet a high energy demand. (Hegger et al., 2008)

Decentralised ventilation systems are proposed to be a state-of-the-art technology which has shown favourable performance in the field. Flexibility and individual control are combined with high energy efficiency and reduced need for space. In the last decade the technology has proven to be mature and successful in operation. This good performance was experienced so far only in the context of Central Europe. If it can be also an an option in another climate like Norway will be investigated in this work.

Objectives

The principal question is:

Is façade-integrated ventilation technology applicable in Nordic climate?

which can be further specified in a subset of questions:

- **Are national requirements and standards met with particular focus on indoor environmental criteria?**
- **Does it represent an energy- and emission-efficient alternative to conventional centralised state-of- the-art ventilation systems?**
- **What are the consequences for building design and planning when decentralised ventilation technology is considered for application?**

Methods & Tools

A comprehensive evaluation of façade-integrated ventilation (FIV) technology will be conducted. The research work is performed in four phases:

1. Characterising the state-of-the-art of FIV technology
2. Specifying categories and criteria of the evaluation
3. Identifying the requirements for ventilation systems in Norway
4. Assessing agreements between FIV systems and national requirements considering the specific characteristics of FIV, which includes:
   - Performing a comparison with state-of-the-art system
   - Investigating the influence of specific conditions in parametric studies

Sources

Information about the current state of the art were found in reports on research, professional and scientific literature. Brochures, presentations and data sheets of commercially available FIV units were explored to find detailed specifications for the evaluation. Manufacturers have been contacted for further information. Response was limited but could provide more details. (The gathered information about current FIV units is available in Appendix B)

The selection of criteria has been guided by regulations and standards which provide required target values. Superordinated framework is the "Veiledning om tekniske krav til byggverk" (in this work referred to as TEK 10). A general guideline for the design of ventilation systems has been found in EN 13779. The standards EN 15251 and ISO 7730 for indoor air quality issues as well as NS 3031 and "Prosjektrapport nr. 42; Kriterier for passivhus- og lavenergibygg – Yrkesbygg" (Dokka et al., 2009; in this work referred to as PR 42) for energy performance requirements provide additional benchmarks if not mentioned in TEK 10.

Assessment tools

Literature

Most of the work is literature-based. Related scientific and professional literature of recent years is used for the assessments where other tools are not available.

In particular the "DeAL; Evaluierung dezentraler außenwandintegrierter Lüftungssysteme; Abschlußbericht." (Mahler et al., 2008; in this work referred to as DeAL) is to be highlighted as reference. This research project has investigated profoundly decentralised ventilation systems with direct access to built case studies over several years and has drawn data mostly from measurements in situ and user surveys. Most of the comparative data is sourced from this document. In many cases of the evaluation results from these surveys will be discussed in the first place and then supplemented and juxtaposed with own results. The case studies were anonymised in the DeAL study. The names were replaced by a three-digit number starting with 201 which is also kept in the diagrams of this work.

Building simulation

For whole year energy performance simulations the software "Simien" version 5.010 is used. "Simien" is based on the the method for dynamic simulation as described in NS 3031 and has been validated against EN 15625 (Programbyggerne, 2008). (Figure 1)

The detailed simulation and parametric studies are performed with "ESP-r" utilising the airflow network model. "ESP-r" was developed at the Energy Systems Research Unit (ESRU) at the University of Strathclyde (ESRU, 2012). In this work the most recent distribution 11.11 for GNU Linux was used. Optical properties for the glazing and shading ("complex fenestration constructions") are obtained from "GSLedit" (see Lomanowski, 2008) using specifications for glazing and shading from "Pilkingtion" and "Warema". (Figure 2)

One single exemplar room is simulated instead of an entire building. This approach reduces complexity, improves quality control and allows flexibility for multi-criteria assessments (Hand, 2010). The exemplar room represents a generic office room for three persons with two FIV units in the uppermost floor of a fictitious office building in Norway. The generic room is not project-specific allowing the simulation of conditions in new buildings as well as in energy-focussed refurbishment of existing buildings. The setting can be considered as worst-case scenario. The room is exposed to outdoor conditions both, on the façade and via the ceiling (roof) while having a high occupancy (8.64 m² per person). A similar setting was also used by "Transsolar" in DeAL (2008). For further details and discussion of the input parameters see Appendix A.

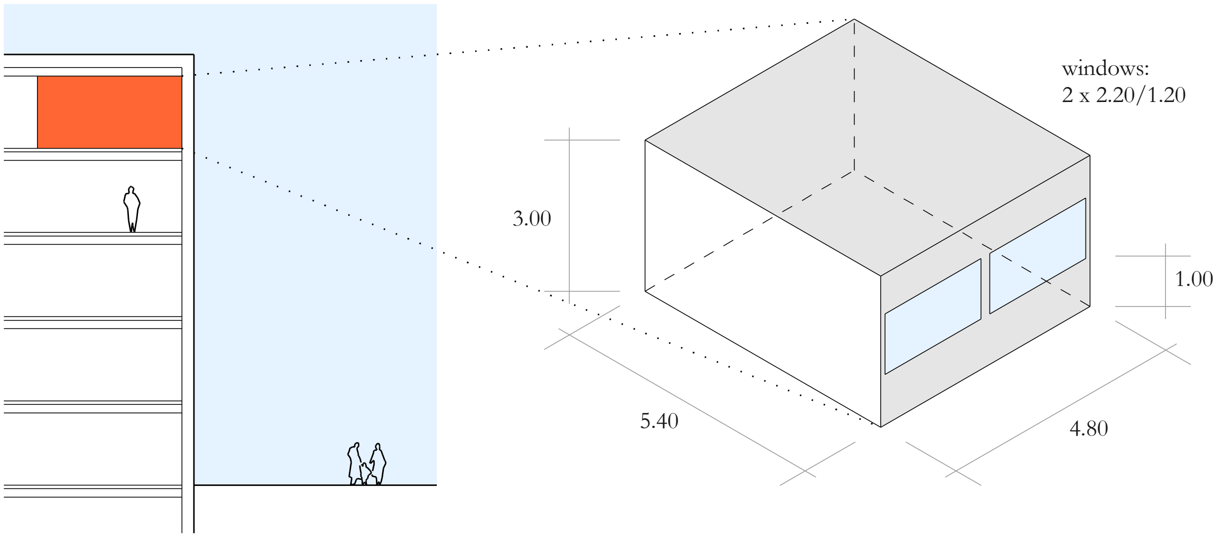

Figure 1: Test room for "Simien" simulations

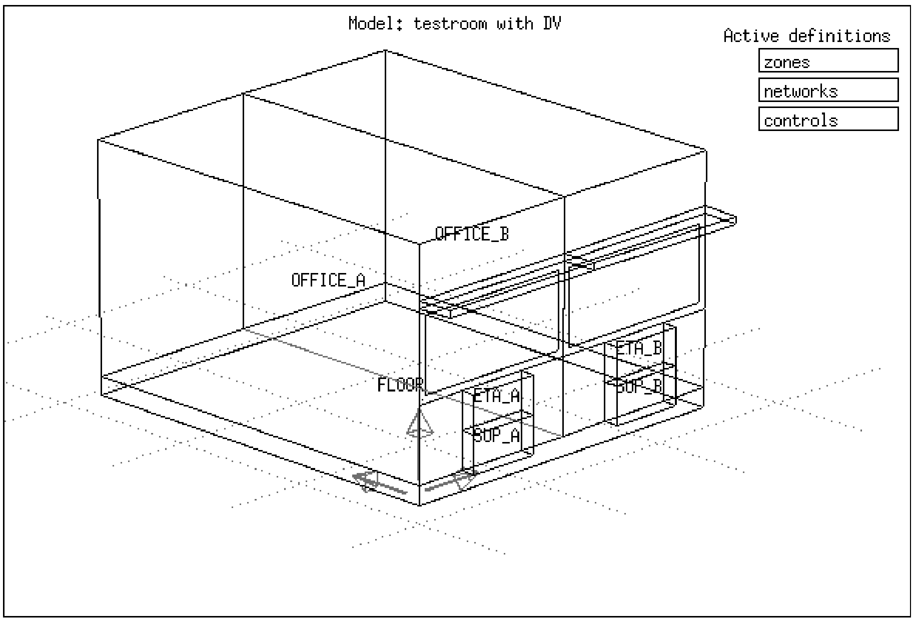

Figure 2: Exemplar room in "ESP-r"

Emission balance

A full inventory of all relevant building components would require full architectural and technical design of a case study for both, a decentralised and a centralised ventilation system. This cannot be provided. However, components of the ventilation systems are used for an approximate comparison. A dataset for FIV units to obtain the global warming potential is found only in the cradle-to-gate database "Ökobau.dat 2011" of the German Federal Ministry of Transport, Building and Urban Development available at BMVBS (2012). The validity of the dataset for non-domestic buildings has been confirmed on request by "PE International" who is the author of the underlying LCA dataset (Braune, 2012). Also the embodied emission dataset for the centralised ventilation system is sourced from the same database for reasons of comparability. The lifetime of 20 years for both systems is given by the database (BMVBS, 2012). Emissions related to operation are obtained from the simulations in "Simien". Operational and embodied emissions origin form different sources. Therefore they cannot be amalgamated in a total emission balance.

Economics

Costs are considered in a life cycle perspective. Novakovic et al. (2007) identify costs for management, operation, maintenance, development and services as additional expenditures to investment costs. Therefore investment, operational and maintenance costs will be estimated.

Manufacturers have been contacted to provide actual prices with mild response. The costs depend on the features of the units and can therefore determined only for the individual project (Plugge, 2012a). Hence literature values for the year 2006 are used (DeAL, 2008) and adjusted with the present price index for construction of buildings from the German statistical office (DESTATIS, 2012).

Available literature was reviewed regarding maintenance routines and expenditures in Norwegian conditions but eventually discarded as no up-to-date data has been accessible. Instead, a spot-check in form of an interview with HVAC engineer Jan Ove Jakobsen at NTNU was conducted. The discussed building is the "Realfagbygget", a multi-functional educational building comprising lecture halls, offices, laboratories, library and a cafeteria. The building was opened in 2000 (Statsbygg, 2012) and can be regarded as state-of-the-art. According to Jakobsen (2012) the ventilation is a demand controlled VAV system.

2. State of the Art

Definition

A distinctive definition of the concept of decentralised ventilation systems is found in DeAL (2008):

- **The supply air is provided decentralised, i.e. via openings in the façade. In some examples also the exhaust air is discharged via the façade.**
- **The conditioning of supply air (and recirculating air) is decentralised, i.e. in every deployed unit.**

Figure 3 shows the two principal solutions of decentralised ventilation systems.

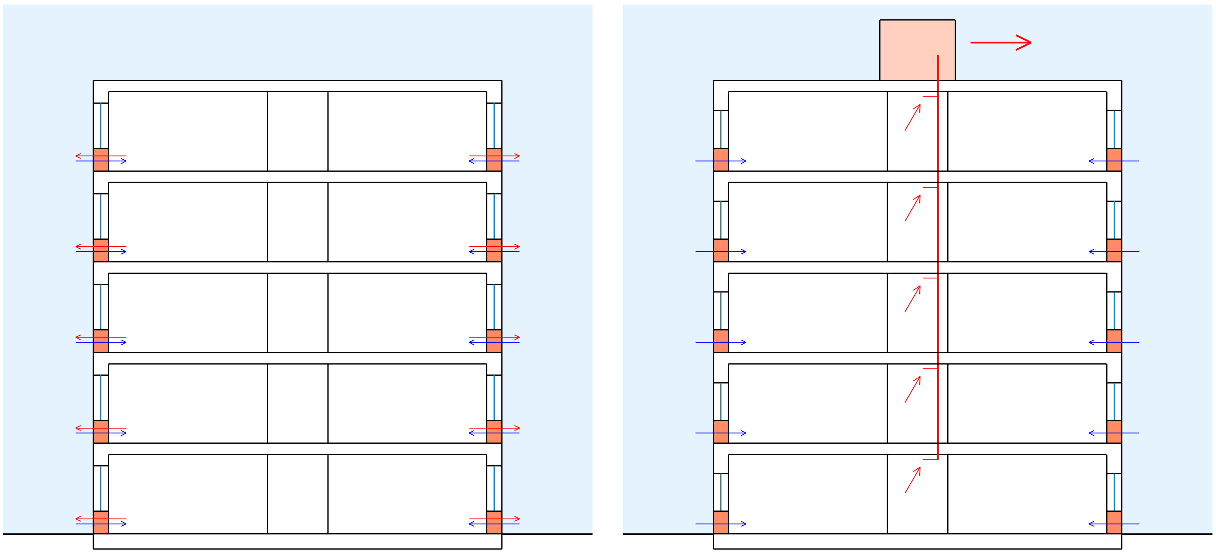

Figure 3: Decentralised ventilation, room-wise (left) or with centralised (right) exhaust (after Mahler and Himmler, 2008)

In the first layout the heat recovery takes place within the individual decentralised unit. Thus a central system is only necessary for core areas and spaces without access to a façade or special requirements e.g. sanitary rooms. In the second layout the used air is extracted to the central parts of the building and discharged collectively which allows the possibility of a central heat recovery with an air-water heat exchanger. The recovered heat is then used e.g. in a low-temperature space heating system.(DeAL, 2008)

It is common to use the term "façade-integrated ventilation" systems to highlight the building integration aspect and the room-wise functionality. Using FIV as terminology also avoids the risk of confusion with the concept of decentralised air handling units for separate parts of the building wings or storeys (compare with Halvarsson, 2012). (DeAL, 2008)

Historically, the concept of FIV derives from the principle to take in fresh air in every room through the façade and precondition the incoming air actively by a heating system or passively e.g. in the intermediate space of a double skin façade. Background for the further development was the need for individual control of indoor environment in the single office rooms. The concept is widely used in Central Europe and described as "Danish model" and "German model" in Dokka et al. (2003).

The airflow in early examples of buildings with façade-integrated ventilation systems was driven by inducing an incoming flow of outdoor air by means of under-pressurising the room with an centralised, fan-driven extract. In these systems fan-less "passive" FIV units served mainly for conditioning the air. Later projects used fan-driven "active" units to increase the airflow rates and allow their individual control. The most prominent built example is the "Posttower" in Bonn embodying an enormous conceptional and technological leap in many fields of engineering. (Joneleit, 2005); DeAL, 2008).

According to Mahler and Himmler (2008) 30 to 40 buildings with FIV technology dating after 2000 were in operation in the year 2008.

Units with combined air supply and discharge and heat recovery will be in the focus of the work. By doing so the impact of circumstances and surrounding influences can be limited and the evaluation can focus on the performance of the FIV units.

Characteristics

Types

FIV units can be classified by their location in the façade. (figure 4)

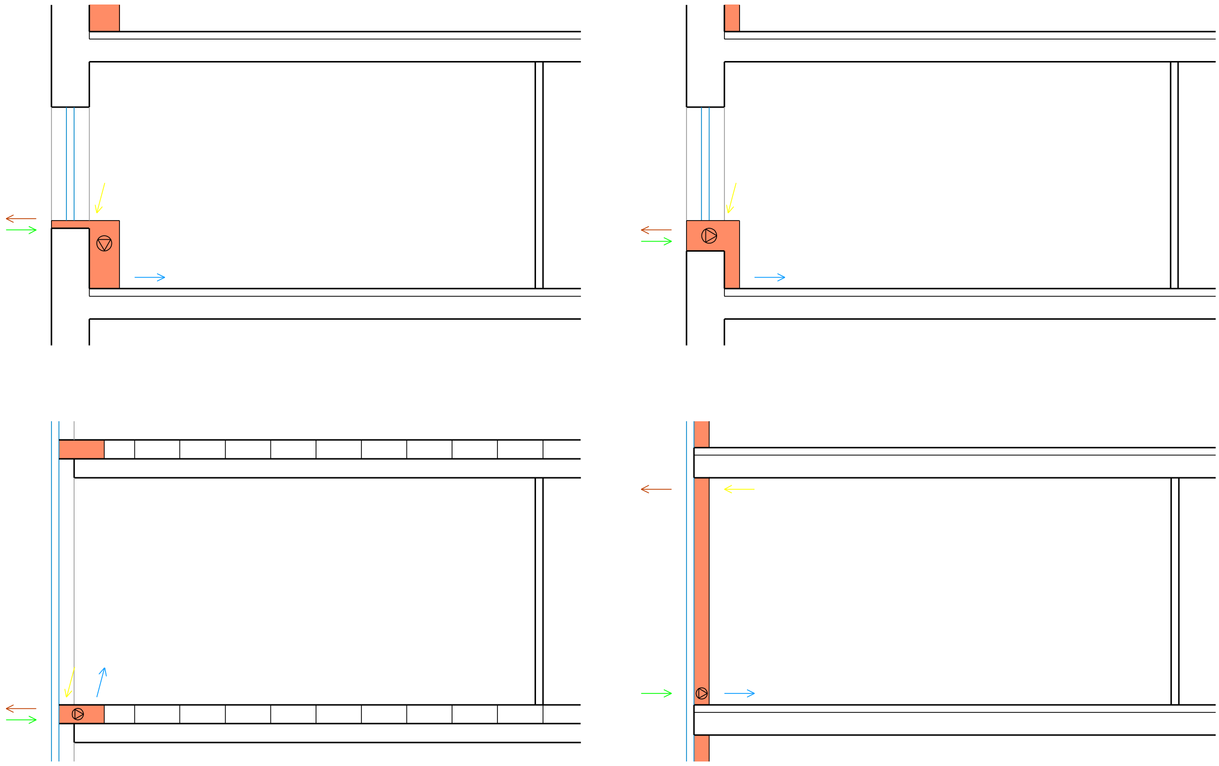

Figure 4: Locations of FIV units: sill-mounted, jamb-mounted, raised floor, wall-integrated (after Hegger et al., 2008)

The group of sill-mounted units is subdivided in units where only the air intake and the air exhaust protrude the façade and units where the entire functional unit for ventilation is to a great extend situated horizontally in the ´jamb´ of the window. They are approximately 0.90 to 1.10 m wide and project approximately 0.20 to 0.30 m into the room.

FIV units integrated in the raised floor are up to 1.25 m wide and 0.60 to 0.65 m deep whereof less than 0.30 to 0.35 m form the grille for air supply and extract. The remaining 0.30 cm are walkable and covered by the raised floor construction. The height of the FIV units is between 18 and 25 cm which in return also defines the height of the necessary raised floor construction.

Wall-integrated (vertical) FIV units are usually project-specific custom-made units. They are characterised by little depth to be integrated in the structural glazing system. One commercially available unit is approximately 16 cm thick and intended for use together with a vacuum-insulation panel on the outer side. (See Apendix B)

System components

A FIV unit is a highly integrated architecture of several functional units in a very limited space. A range of commercial out of the box units is available from several manufacturers. However, project-specific solutions are common adapting the number and specifications of functional units and their components to individual settings. An order with a minimum volume of 100 units is necessary for custom-made technology (Plugge, 2012a). Consequently individual solutions are possible starting at an approximate 12000 m^3/h airflow rate given the common maximum airflow rate of 120 m^2/h for the smallest units.

The following enumeration of functional modules based on Trox (2009) describes the modular construction of one unit.

Air supply unit

This module controls the airflow and pressure conditions of the air supply. Dampers ensure one-way operation and shut-off the airflow if outdoor conditions are not desirable. Flow rate controllers limit the air volume to the needed amount. The fan (usually radial fans with energy-saving EC motors) is ideally equipped with a sound damper. Another function is the removal of fine particulates by means of an air filter with the filter classes F 5 to F 9.

Heat exchanger unit

The conditioning of supply air is water-borne. 2-pipe or 4-pipe systems are available. 4-pipe systems provide cold and warm supply water simultaneously and are therefore more suitable for intermediate periods where heating and cooling loads night occur (DeAL, 2008). The module also comprises the corresponding valves and a supply air temperature sensor. Besides the convective air-conditioning some units can also employ static heating without the use of fans in unoccupied periods. The cooling capacity is limited by working as ´dry´ cooling since only transient condensation liquid can be handled within the unit.

Extract air unit

A coarse particulate air filter (e.g. filter class G 3) is preceded to avoid damaging the unit and the heat recovery unit. The subsequent order of fan and dampers is similar to the supply air unit.

Heat recovery unit

As heat recovery unit usually plate recuperators are in use. Therefore efficiencies are limited to approximately 55..60 percent. The use of plate heat recovery also entails the risk of icing of the unit. Therefore often solutions with bypass are employed. Pre-heating for frost protection which is common in Norway has not been integrated but can be considered on request (Plugge, 2012a).

Secondary air unit

The use of secondary air is a common strategy to reduce the energy demand. The air is usually mixed with fresh outdoor air and possibly conditioned in the heat exchanger. During night-time the unit may operate merely with secondary air to heat or cool the building outside operation times.

PCM unit

Recently, some manufacturers introduced the possibilities to integrate phase change materials (PCM) as latent heat storage. The supply air flows flows through a PCM stack where the air is cooled before being supplied to the room. During the process the PCM (paraffin) melts which can take several hours. Hence the functionality can stretch up to 10 hours. At night, the cool outdoor air consolidates the PCM again. (Trox, 2009 and Emco, 2012)

Additionally, FIV units can be equipped with components for humidification and dehumidification. Due to the complexity of this feature these components are used only in special cases e.g. in museums and the uppermost floor of highly glazed buildings. (DeAL, 2008)

Environmental control

Figure 5 gives an overview of possible capabilities of FIV technology to control the indoor environment. (after Mahler and Himmler, 2008)

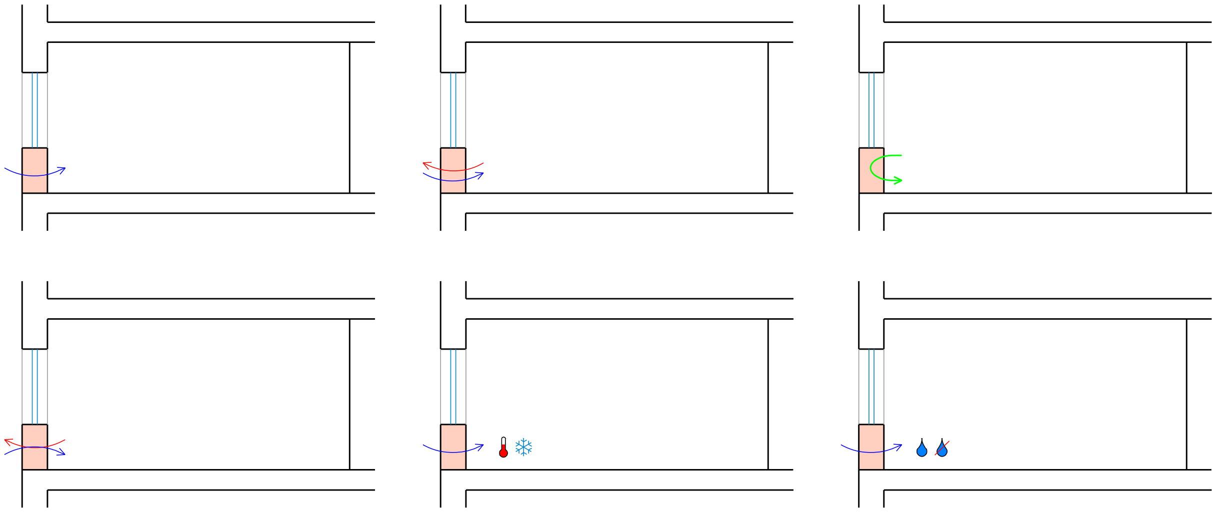

Figure 5: Functionalities: air supply, air supply/exhaust, secondary air, heat recovery, heating/cooling, de-/humidification (after Mahler and Himmler, 2008)

FIV units are usually part of a ´slender´ HVAC concept. To satisfy the heating and especially the cooling loads FIV systems are usually combined with thermo-active building systems (TABS) in the concrete slabs. The quick responding FIV units cover the ventilation heat losses while the slowly reacting activated building elements cover the remaining transmission heat losses. For this purpose the planar TABS allow low operation temperatures and therefore use of renewable energy sources like ground via boreholes or groundwater heat pumps. (Hirn, 2009)

Another common practice is a mixed installation of FIV units. Units supplying conditioned fresh air alternate with units with air-conditioning function only. These can be FIV units in secondary air mode or convection heating/cooling units. Since manufacturer usually provide both units this concept is characterised by visual uniformity despite the technology mix. (DeAL, 2008)

As main advantage of FIV technology is considered that ventilation is provided only when needed i.e. when the space is occupied and air has to be conditioned to the extent which is required to reach indoor comfort. This was in accordance with the obsolete DIN 1946 part 2 (now replaced by EN 13779) where ventilation demand is only defined per persons and pollution from materials is not considered. Room-wise control allows that only occupied spaces are ventilated while unoccupied rooms are controlled not more than to keep the room temperature within defined limits. (Sefker, 2006)

Users have the possibility to control the indoor temperatures and ventilation rates to their liking. The user interfaces are located in wall-mounted sockets or integrated in the FIV units in the rooms. A variety of settings and user control options is possible. Manual settings (tellingly called "comfort") allow individual control of room temperature and airflow rates. Automatic preferences ("stand-by") have two functions. Protective settings prevent the building from damage by ensuring minimum or maximum temperatures also outside occupancy or protect the FIV e.g. by activating a bypass to avoid icing of the plate heat recovery. Other automatic settings control the indoor environment based on set-points similar to centralised ventilation systems. A variety of control sensors (time, presence, CO2, window switch, etc.) are used in operation facilitating demand-controlled ventilation operation. Room-wise control allows special settings. One example is "absence" mode. The room temperature is then not fixed to a set-point but is allowed to wander within a defined range to ´prepare´ the room for expected occupancy. Integration in the building management system and central control is possible by a data bus to information networks in addition the local control possibilities. (DeAL, 2008; Emco FLH, 2010; LTG FVM, 2009; Trox, 2009)

Advantages & Disadvantages

Hirn (2009) summarises the assets and drawbacks connected with FIV technology.

Advantages

- **ventilation only activated when room occupied**
- **energy savings by demand control and window ventilation**
- **individual ventilation and temperature control**
- **higher user satisfaction**
- **lower demand for technical spaces and suspended ceilings**
- **tenant- and zone-specific billing**

Disadvantages

- **problems with noise and draught at high airflow rates**
- **demanding humidification/dehumidification**
- **higher effort for maintenance and filter replacement**
- **restricted possibilities of heat recovery within units**
- **higher energy demand in buildings with centralised control**
- **influence of outdoor conditions on performance of units**

3. Evaluation

Four categories of evaluation criteria have been determined:

- **Indoor environment**
- **Environmental impact**
- **Operation & life cycle**
- **Building integration**

The evaluation of thermal comfort, humidity, acoustic conditions and foremost aspects of indoor air quality assesses the accomplishment of desired indoor environment and comfort conditions.

The environmental impact is evaluated by investigating the energy efficiency and emission efficiency. Therefore FIV technology is confronted with a centralised system and the performances are compared.

Issues regarding operation and maintenance are addressed by reviewing the hygienic and technical aspects of installation and the operability of the units in everyday use by users. An examination of investment, operational and maintenance costs can help to determine the economical viability of the systems.

Lastly, the influence of FIV units on the planning and design of space plan and building components and on the design process as such are evaluated. Spatial requirements regarding room geometry and façade as well as implications on the programme of the building are discussed.

Indoor Environment

User satisfaction

The human sensation is invaluable as starting point for the assessment of comfort conditions. A survey among tenants of five buildings 201 to 209 was conducted in the framework of DeAL (2008) and has shown mostly contentment with the indoor environment in spaces with FIV. (Figure 6)

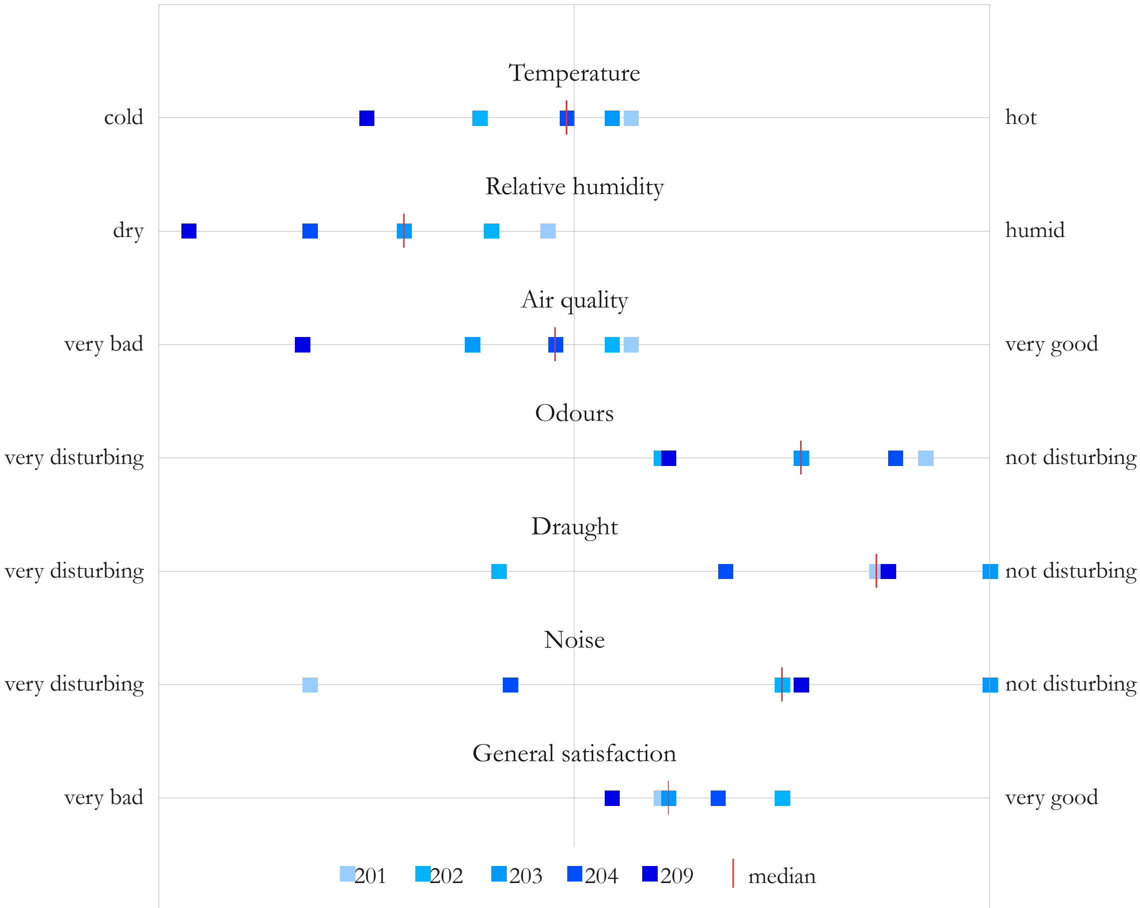

Figure 6: User satisfaction based on surveys among tenants (after DeAL, 2008)

The majority of users is very satisfied with the indoor environment. Only building 209 is generally problematic regarding comfort conditions. Other buildings have single issues which are related to faulty operational settings. Fixing those would improve the contentment. The thermal comfort is mainly perceived as balanced. In one building (202) too low supply air temperatures lead to issues related to temperatures and draught. Hence also the air quality was regarded low.

The air in the buildings is predominately evaluated as too dry. Humidification as air treatment was installed in none of the examined rooms. (DeAL, 2008)

In general, indoor air quality was perceived as neutral – neither too good nor too bad. Odours of new interior and furniture in newly occupied buildings influence the perception of odours while buildings which are in use for a while the odour nuisance is considered small. The acoustic environment seems to depend strongly on the building. One case study (203) with excellent results regarding noise and draught uses passive FIV units in an open-plan office. (DeAL, 2008)

Thermal comfort

Measurements in DeAL (2008) showed good results regarding the capability of FIV systems to keep the room air temperature within the comfortable range of 22 to 26 °C as required by the relevant DIN 1946 part 2. Additionally, relevant data (operative temperature, relative humidity, air velocity) was evaluated to determine the thermal state of the body as a whole according to ISO 7730. Figure 7 shows the predicted percentage of dissatisfied index (PPD) of the examined case studies for the time of measurement.

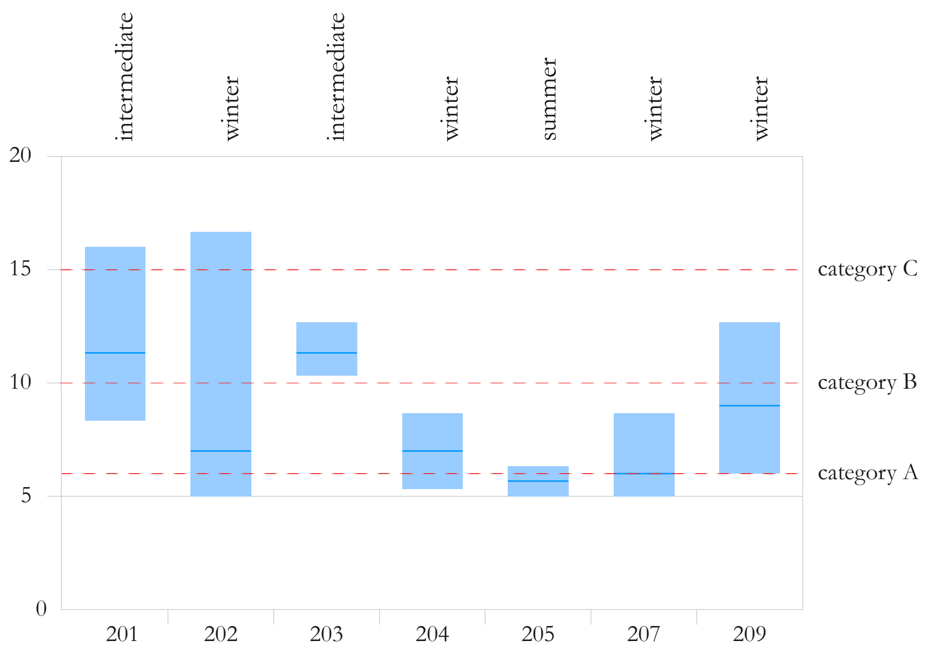

Figure 7: Measured PPD indices in buildings with FIV (after DeAL, 2008)

The measured values for the intermediate season are in the range of ISO 7730 category C and are higher than for the other seasons. The results for winter conditions generally show a wide band of extrema within the building with an average in category B. Only one building was examined during summer. Category A could be achieved then. A possible reason can be found in the results for the investigations of draught in the buildings. Three measured buildings in category B and C had significantly higher draught rate above 15 %. In two of those buildings the problems were caused by wrongly calibrated units with excessive airflow rates or too low supply air temperatures. (DeAL, 2008)

Operative temperature

The Norwegian building code TEK 10 § 13-4 requires an operative temperature between 19 and 26 °C for light work. The minimum temperature shall be satisfied every time, meaning also in the unoccupied period. 22 °C are preferably not exceeded during heating season. 50 hours per year above 26 °C are admissible with reference to the meteorological statistical annual data (according to TEK 10 is the outdoor temperature 50 hours above 26 °C in the standard climate for Oslo/Blindern). In NS 3031 the set-point temperature is set to 21 °C during and 19 °C outside occupancy. Set-point temperature for cooling is 22 °C.

Maintaining thermal comfort cannot be provided with ventilation supply temperature alone in FIV systems. The ruling principle of displacement ventilation is ideal for cooling but should not used as an all-air system as most authors report unanimously. Radiant heaters, floor heating and convective systems close to the floor like baseboard heater are proposed as heating systems for displacement ventilation (Skistad, 1994; Sodec, 2002; Chen and Glicksman, 2003).

The interaction of FIV with the several heating/cooling systems can be compared. The base case represents an almost ideal system with convectors for heating and a night cooling strategy with quick response to prevent overheating. A first case study investigates ventilation preheating with secondary air during the night which is a strategy often connected to FIV systems. Also the combination of FIV systems with thermally activated building systems (TABS) is very common (DeAL, 2008) and provides a second case study. Moreover, radiant floor heating/cooling can be investigated. Radiant floor heating/cooling and TABS are of special interest because also cooling loads can be covered in summer in addition to heating loads in winter.

The base case will be examined more extensively for the coldest and the hottest day in the given climate data (31.12. respectively 31.07.), a very cold week (15. - 21.01.), a hot week (23. -

29.07.) and an erratic week in the intermediate season (23. - 29.04.). The investigations of the other cases studies focus on the performance on the coldest and the hottest day.

In the simulations of the base case a control strategy is applied which is intended to run perennial without seasonal alteration. From 18:00 to 7:00 the ventilation heating switches to free-floating if the indoor room dry bulb temperature (used as set-point temperature in all cases) is above 19 °C. Night cooling from 18:00 to 7:00 with 60 m³/h outdoor air is activated if the dry bulb temperature is above 20 °C. Heat recovery is established as fan running with 55 % airflow rate of the supply fan between the zones for extract air ETA and the zones for supply air SUP to transport heat from the extract to the supply (see also figure 2). This control is switched off if 24 °C dry bulb temperature are reached from 7:00 to 18:00 respectively 20 °C from 18:00 to 7:00. A similar strategy is found in Høseggen (2008).

The operative temperatures and the heating loads for the cold week in January are presented in figure 8.

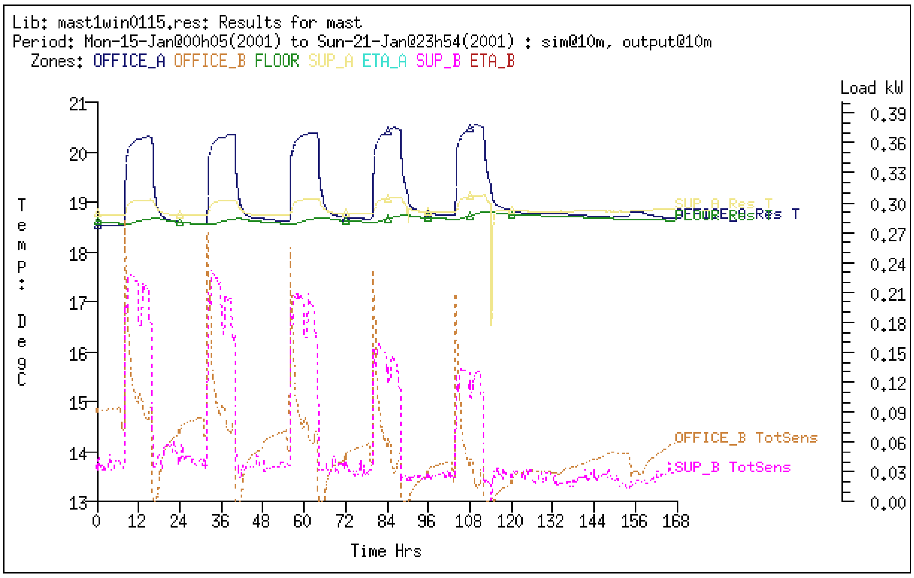

Figure 8: Convective heating (base case), operative temperatures and heating/cooling load in winter week 15. - 21.01.

The operative temperature varies between 18.5 and 20.6 °C in the office room. The room temperature increases rapidly with and plunges with the heating period. A short shut-off of the ventilation heating according to the above-mentioned strategy is noticeable on Friday afternoon.

The coldest day in the weather file is the 31st December with temperatures as low as -17 °C. It can be seen as worst case (figure 9).

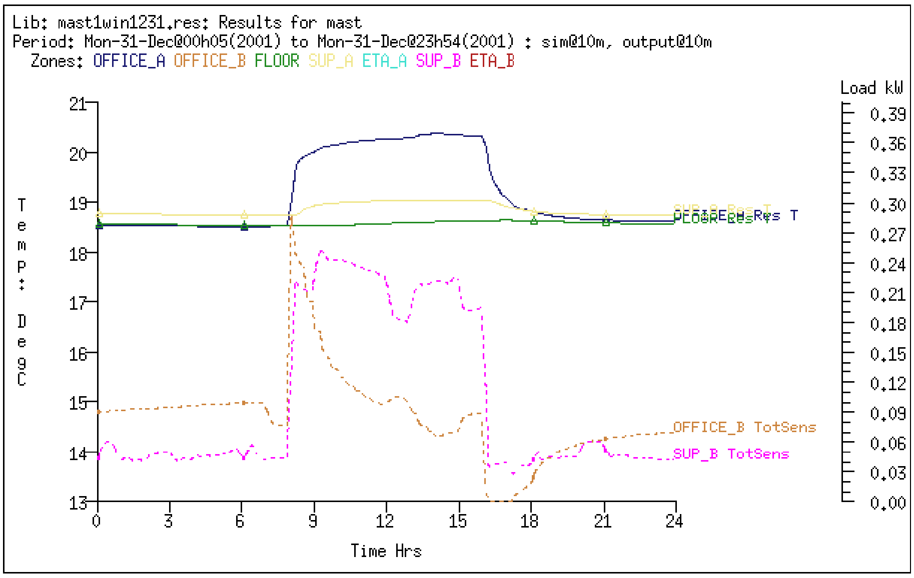

Figure 9: Convective heating (base case), operative temperatures and heating/cooling load on 31.12.

20.4 °C is the maximum operative temperature in the room. 940 W total heating capacity (36.3 W/m²) are necessary in the morning for heating. Thereof 250 W can be assigned to each of the FIV units to maintain the supply air temperature of 19 °C.

The results of the typical summer week are shown is figure 10.

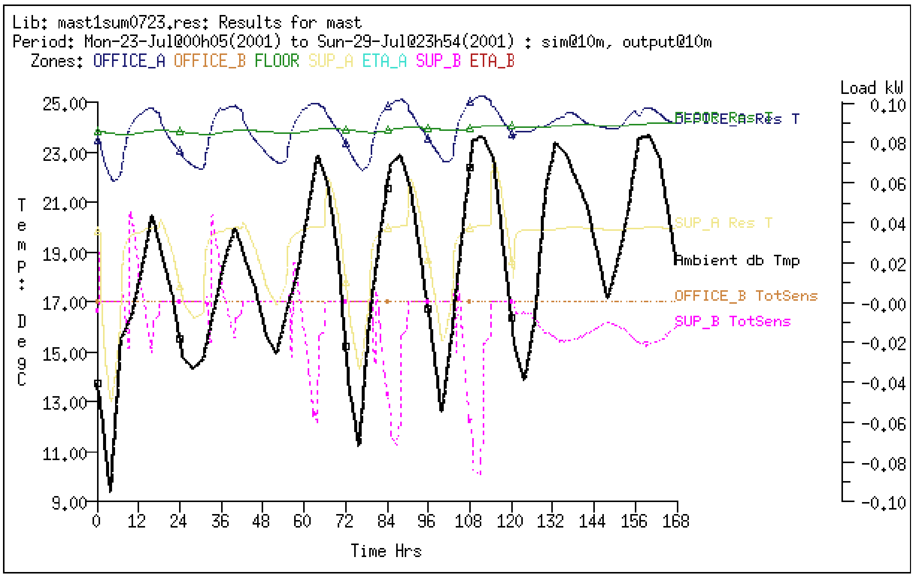

Figure 10: Convective heating (base case), operative temperatures and heating/cooling load in summer week 23. - 29.07.

The operative temperatures during the day do not exceed 25.2 °C. Also during the night the temperatures do not drop below 21.9 °C. Notable are the minor cooling demands followed by small heating demands on some days in the early working hours. This is due to the operation of the heat recovery which is still activated in the beginning and then switched off as high room temperatures are reached. The deactivation again requires heating as the outdoor temperatures are still below 19 °C.

The 31st July is the hottest day in the climate file with outdoor air temperature reaching up to 28 °C. The results are presented in figure 11.

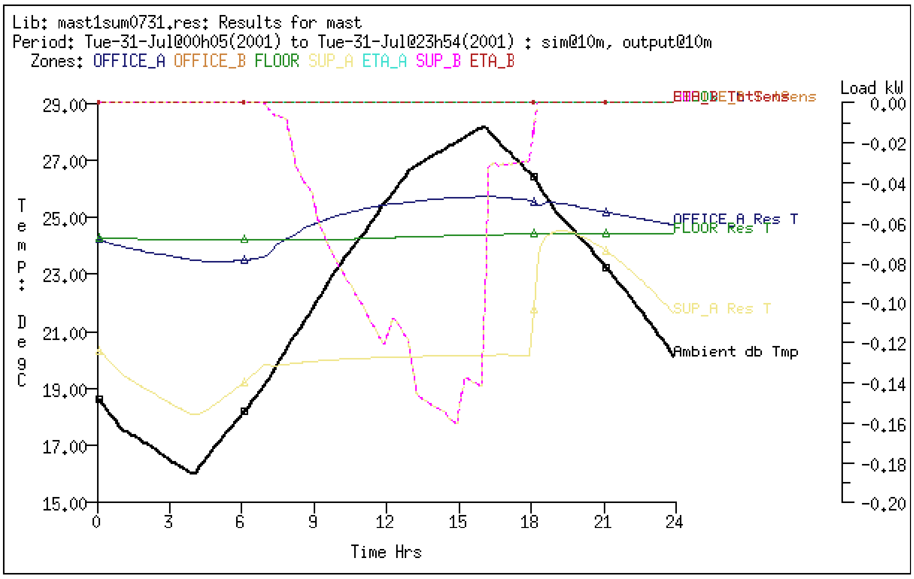

Figure 11: Convective heating (base case), operative temperatures and heating/cooling load on 31.07.

The operative temperature remains does not exceed 25.7 °C. 320 W (12.3 W/m²) total cooling capacity are necessary to ensure this. Since the outdoor air used for night cooling after 18:00 is still 24 °C warm the room temperature remains above 23.5 °C.

Figure 12 shows results for the week in the intermediate season where both high and low outdoor temperatures (+21 °C as well as -3 °C) are present.

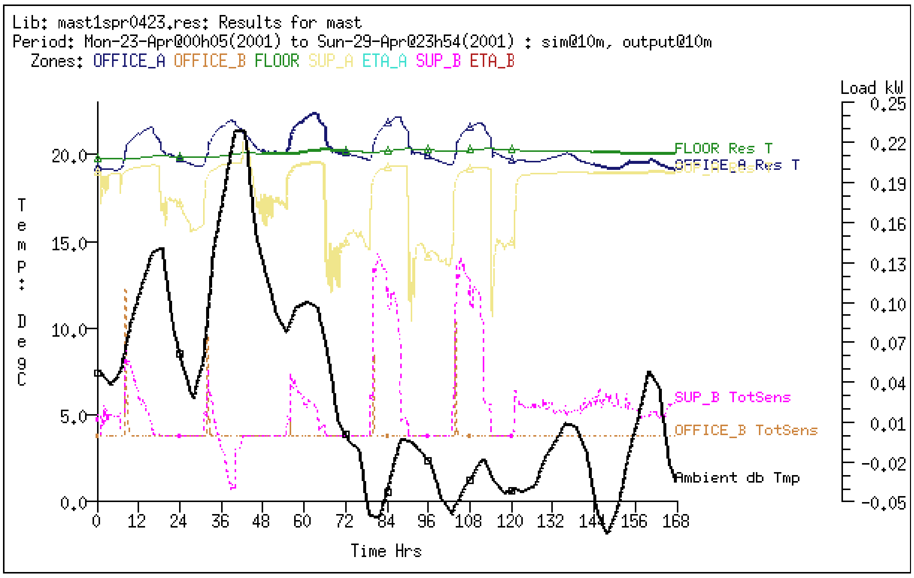

Figure 12: Convective heating (base case), operative temperatures and heating/cooling load in spring week 23. - 29.04.

Operative temperatures below 22.3 °C are maintained when the outdoor temperature raises to 21 °C. Space heating is only required at the very beginning of the working hours at 8:00. Both, heating and cooling loads for the ventilation have to be covered on Tuesday which addresses the necessity of quick response. Thus a 4-pipe system to supply both hot and cold water to the heat exchanger is favourable. Cooling loads are covered easily by night cooling. However, the set-point room temperature for night cooling is crucial for a successful application as the on-off pattern in the supply air temperature indicates.

Ventilation preheating in an office building in Norway was studied by Wachenfeldt (2007). With a supply air temperature during the day of 20 °C the operative temperatures remained above 20 °C at -20 °C outdoor temperature. During the night a preheating strategy with 30 °C warm supply air conditioned the office space.

This strategy resembles the operation of FIV systems employing secondary air for heating/cooling outside occupancy. Heating is provided with 35 °C supply air temperature (set-point temperature) from 18:00 to 7:00 during the night. In summer the supply air temperature is reduced to 16 °C. During occupancy the supply air temperature is 20 °C in winter and 19 °C outside the heating season to operate with displacement ventilation. The airflow rates are linked to the minimum ventilation rates.

Results for the coldest winter day are shown in figure 13.

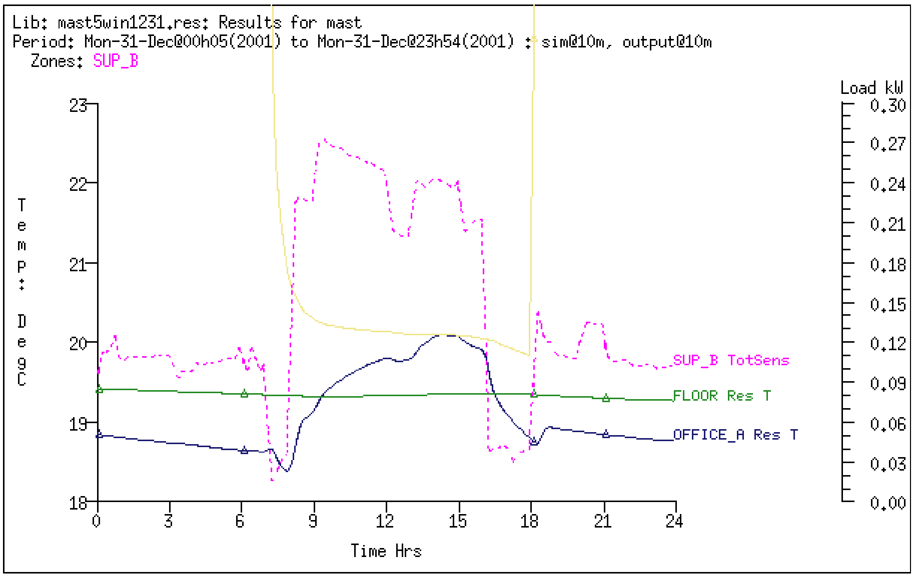

Figure 13: Ventilation preheating, operative temperatures and heating load on 31.12.

Although the supply air temperature during occupancy is increased by 1 Kelvin the operative temperatures remain low. On the coldest day barely 20 °C are reached at the end of the occupancy. Constant ventilation heating is required during the weekend to maintain a stable temperature.

Figure 14 shows the situation on the hottest summer day.

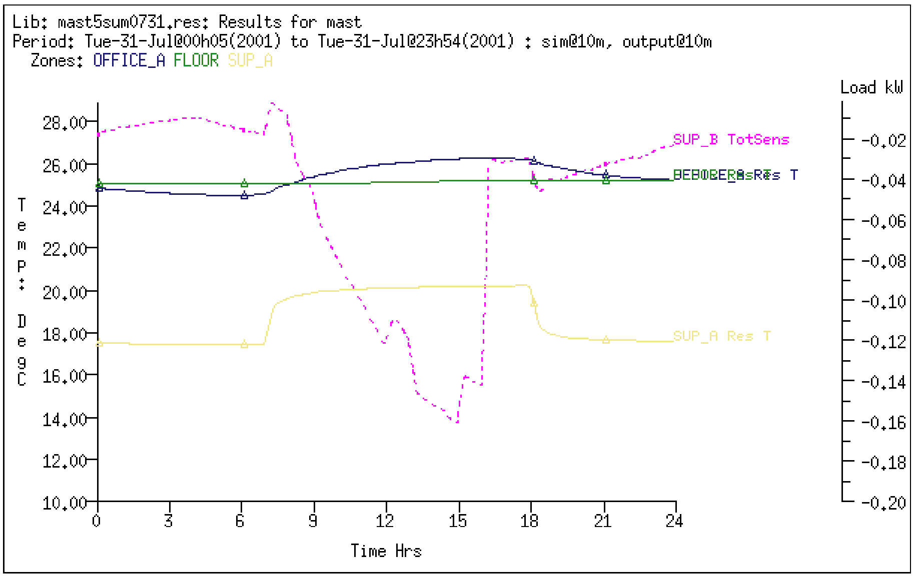

Figure 14: Ventilation preheating, operative temperatures and heating load on 31.07.

The temperature in the room reaches above 26 °C at the end of the working hours. This was also detected during the warm summer week. The supply air temperature of 16 °C outside occupancy could not be lowered as it is regarded as the minimum to omit the risk of condensation within the FIV unit (Plugge, 2012a). Therefore additional measures are necessary if higher loads would occur.

Thermally activated building systems (TABS) can be used to cover stationary cooling loads of 30 to 40 W/m² and heating loads of 25 to 30 W/m² with very low temperature supply water temperatures. Limiting factor for TABS is the temperature of the ceiling surface. On one hand, condensation is possible if the dew point temperature on the surface is reached. On the other hand, comfort criteria in ISO 7730 regarding radiant temperature asymmetry limit the surface temperature to a range of 21 to 25 °C. (Pfafferott and Kalz, 2007)

In order to achieve similar thermal conditions as in the base case a total of 300 W heat is injected (respectively 100 W extracted in the summer case) in the middle of the concrete slabs in the ceiling and in the slab below the raised floor. The TABS are activated during the night between 16:00 and 7:00 from Sunday to Friday. Night cooling with outdoor air is deactivated.

Figure 15 shows the results for the winter season.

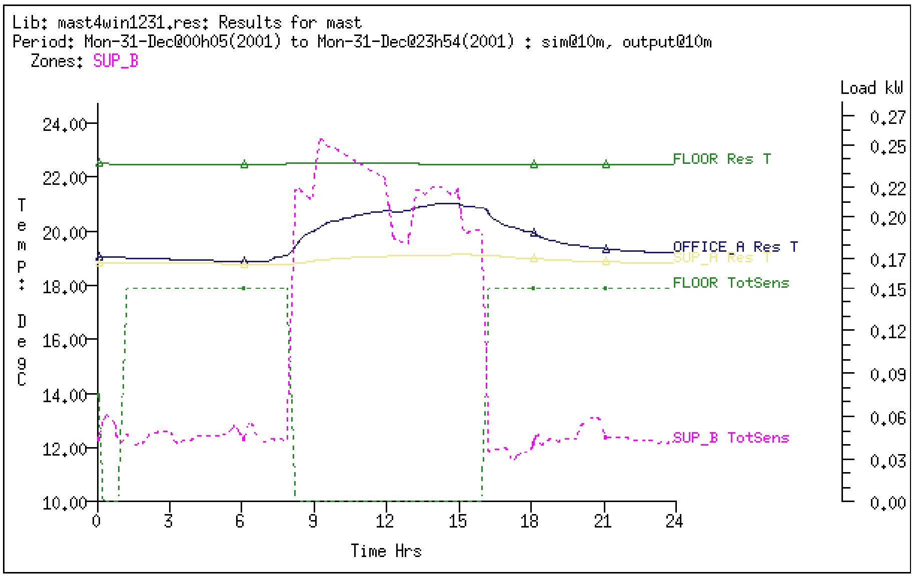

Figure 15: TABS, operative temperatures and heating load on 31.12.

TABS maintain the room temperature during the night. Only the discharge of heat from the slabs and internal gains supply heat during the day which results in a gradient increase of the operative temperature with peak in the afternoon. A steady increase of the room temperature in the office and under the raised floor over the week from Monday to Friday is observable rendering heating on weekend is unnecessary.

Results for the summer day are presented in figure 16.

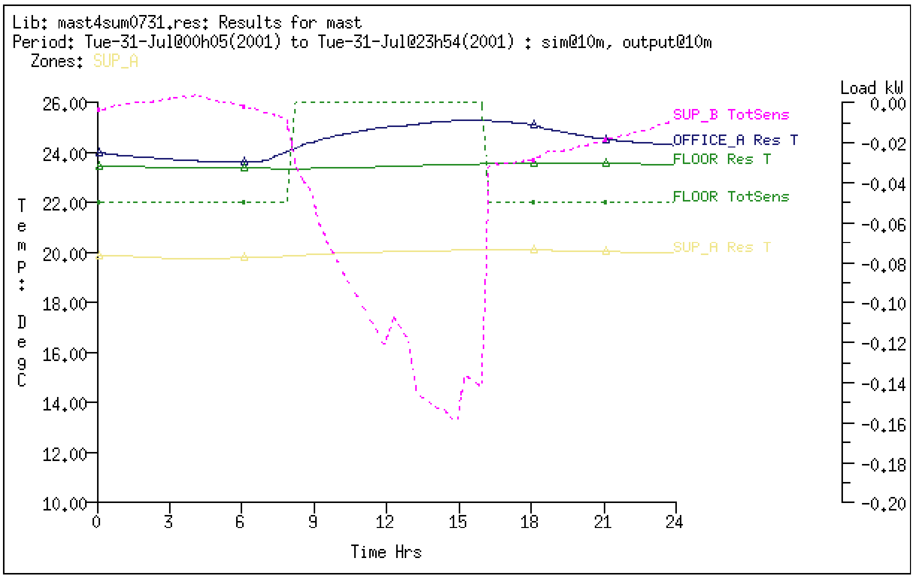

Figure 16: TABS, operative temperatures and heating load on 31.07.

During summer TABS control the operative temperature very efficiently. Only 2 W/m² heat is extracted from the concrete slabs which is very little compared to the possible 30..40 W/m². Increasing the cooling capacity to 6 W/m² results in room temperatures below 23 °C also during the hottest day. The cooling capacity of the ventilation remains unchanged.

The effect of TABS on the surface temperatures and conduction heat fluxes of the ceiling and floor as well as of the slab under the raised floor is shown in figures 17 to 20 compared with base case with convective heating.

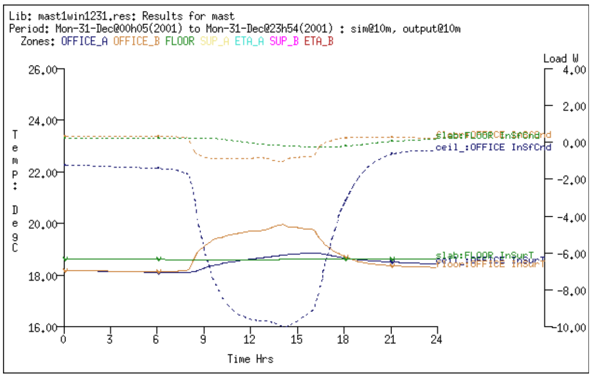

Figure 17: Convective heating, surface temperatures and heat fluxes on 31.12.

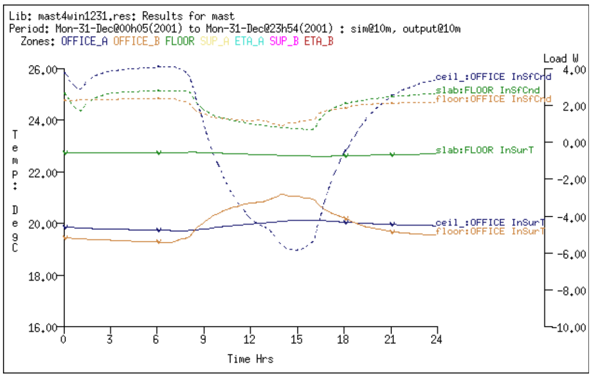

Figure 18: TABS, surface temperatures and heat fluxes on 31.12.

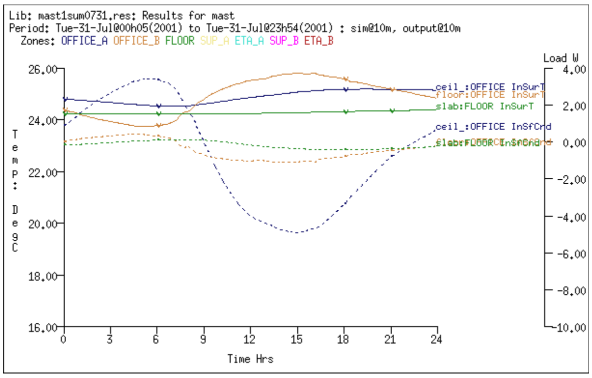

Figure 19: Convective heating, surface temperatures and heat fluxes on 31.07.

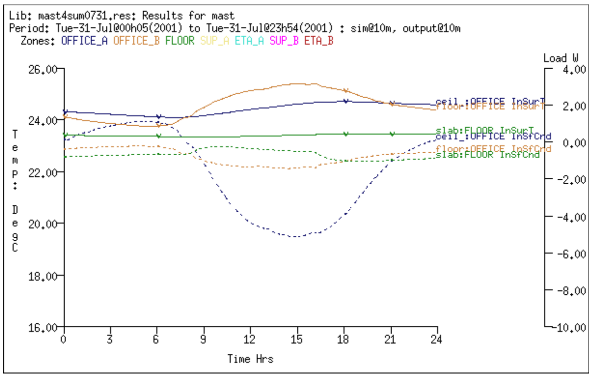

Figure 20: TABS, surface temperatures and heat fluxes on 31.07.

In the winter scenario the reduced heat flux to the outside (negative values) during occupancy and the heat release to the room (positive loads) during the night are noticeable. The surface temperatures are generally higher than in the base case.

During the summer day the heat is retained outside occupancy. The surface temperatures are 0.5 Kelvin lower than the base case. The effect of TABS can be compared to the night cooling strategy which is applied in the base case. Figures 21 and 22 show the convective heat fluxes on ceiling and floor of the office. The discharge of heat is measurable effect especially in the first hours of the day. Notwithstanding, the TABS show a higher performance at very low capacity as more heat is discharged by the ceiling when comparing figures 19 and 20.

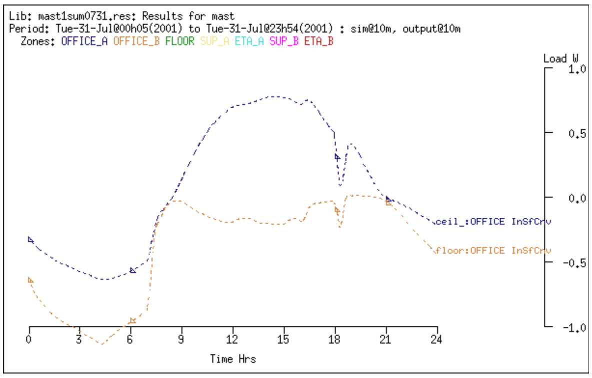

Figure 21: Convective heating, conv. heat fluxes on floor and ceiling

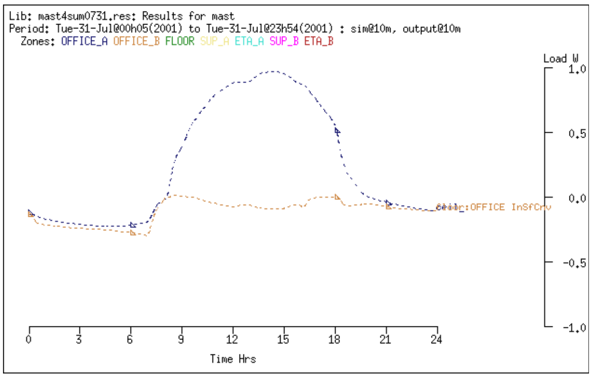

Figure 22: TABS, convective heat fluxes on floor and ceiling

Skistad (1994) and Causone et al. (2010) put forward that radiant floor heating and displacement ventilation do not interfere with each other. Causone et al. (2010) obtained results from experimental tests in the environmental chamber that heating loads up to 60 W/m² and cooling loads up to 76 W/m² can be covered with radiant heating/cooling without having a negative impact on the comfort requirements of displacement ventilation. This is interesting when a raised floor is not desired or cannot be installed due to a low ceiling height as for refurbishment. Comfort criteria according to ISO 7730 for radiant floor heating/cooling is a surface temperature of the floor which remains within the range of 19 to 29 °C.

The simulation seeks to achieve similar thermal conditions in the room like the base case. Hence set-point temperature is 25 °C. 750 W (58 W/m²) are injected or extracted from the layer in the floor construction. Instead of a raised floor the adjacent zone is another office room with similar thermal conditions as the examined office room.

The results for winter are shown in figure 23. Heating occurs sporadically outside occupancy but also in the first working hours. The delay between injection and measurable surface temperature is evident. In general, the room temperature fluctuates steadily between 19 and 21 °C during the week. The simulation performed in the coldest week shows for the Saturday without occupancy that no intervention of the heating system is measurable despite outdoor temperatures between -2 and +5 °C.

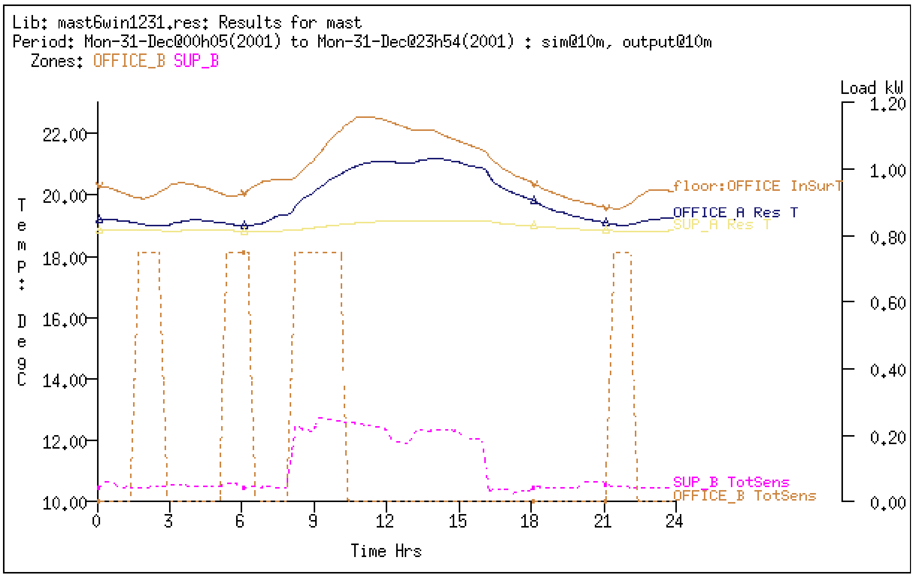

Figure 23: Radiant floor heating, operative temperatures and heating/cooling loads on 31.12.

During summer the activity pattern changes (figure 24).

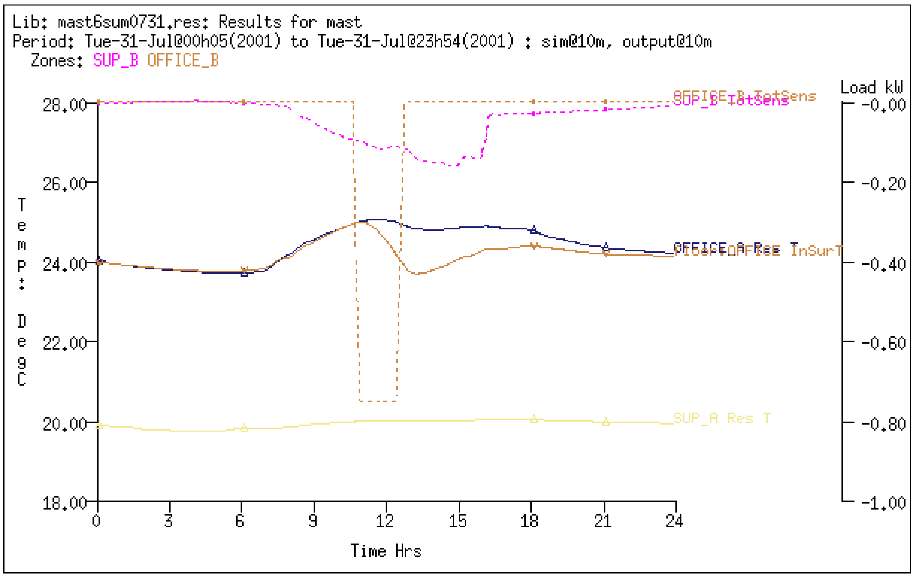

Figure 24: Radiant floor heating, operative temperatures and heating/cooling loads on 31.07.

In the cooling period radiant cooling is present during occupancy to cover peak temperatures around noon. The floor temperature decreases and remains cooler than the room until the end of occupancy.

Predicted percentage of dissatisfied (PPD)

The PPD can be used as an indicator to evaluate the thermal state of a user as a whole. Figures 25 to 32 show the PPD's of the case studies for the coldest and the hottest day. The air velocity is set as 0.1 m/s. Clothing in summer is assumed as 0.5 clo and in the other cases as 1.0 clo in accordance with EN 15251.

In winter the three latter alternative systems delay achieving comfort conditions in the beginning of the main occupancy. Ventilation preheating has problems to reach PPD category B (10 %) in general. TABS show fast process at the beginning of occupancy but slow down during the morning. The radiant floor achieves comfort faster in the morning. The reference case of convective heating shows instant response but also fast decay after the heating period. In summer, the two planar systems show best results while the ventilation system show increasingly poorer performance during the day.

In summary, the practice of ventilation preheating with secondary air during the night in combination with displacement ventilation during the day shows unsatisfactory results in Norwegian conditions. Additional heating and cooling during occupancy or other strategies are necessary. An alternative can be convective heating. Then the location of the heating devices and the interaction of the supplied heated air with the airflow pattern of the supply air unit has be investigated further. However, this cannot be done within the scope of this work.

Planar heating/cooling systems show very good results. TABS maintain comfortable conditions in the heating and the cooling season although operating only outside the occupied periods. Prominent are the very low required heating and cooling capacities. There is even more potential of TABS since the capacities used in this simulation are far from the maximum capacities. The radiant system has the advantage to respond also during the occupancy where only punctually demand has to be covered for a few hours. However, for both systems further investigation is necessary. In the intermediate season heating and cooling loads may occur alternatingly. Hence additional measures are necessary since changeover operation with planar systems is difficult to establish.

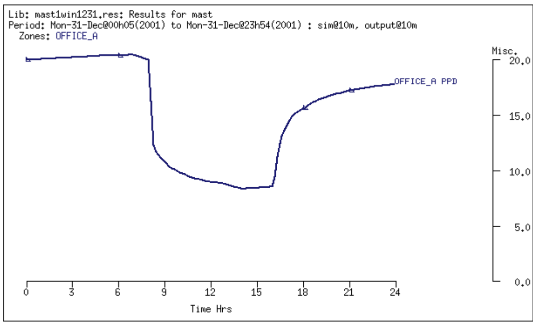

Figure 25: Convective heating, PPD index on 31.12.

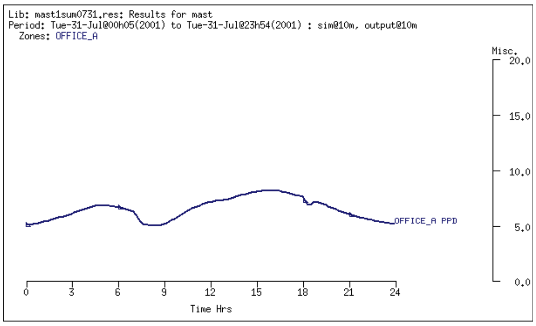

Figure 26: Convective heating, PPD index on 31.07.

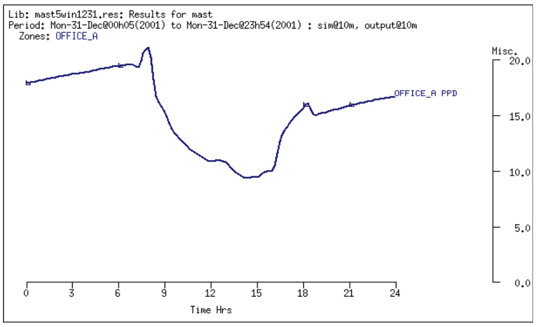

Figure 27: Ventilation preheating, PPD index on 31.12.

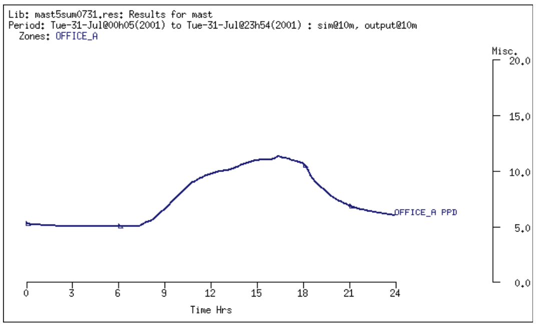

Figure 28: Ventilation preheating, PPD index on 31.07.

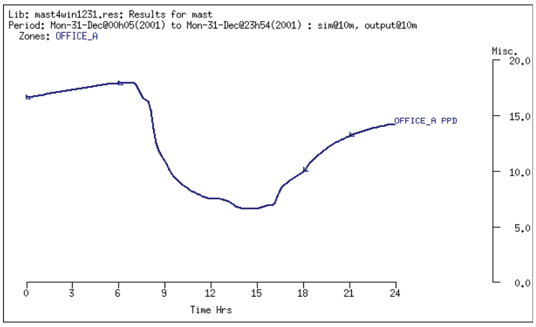

Figure 29: TABS, PPD index on 31.12.

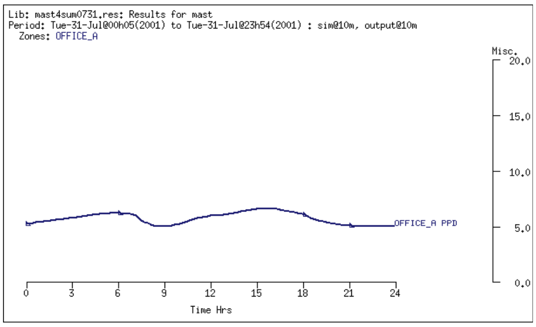

Figure 30: TABS, PPD index on 31.07.

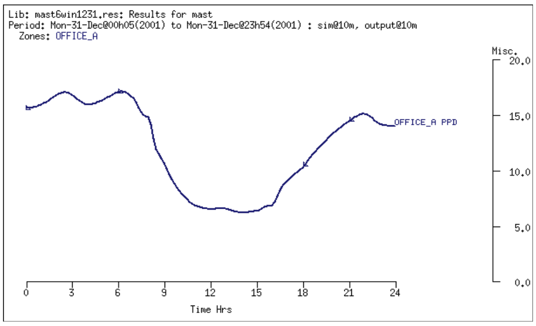

Figure 31: Radiant floor heating, PPD index on 31.12.

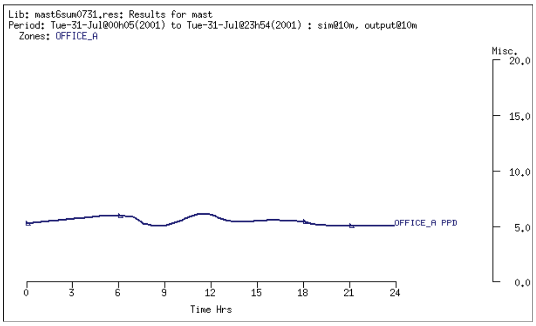

Figure 32: Radiant floor heating, PPD index on 31.07.

Indoor Air Quality

The potential of FIV can be investigated by the examination of how effective the ventilation air is supplied and distributed, the provided ventilation rates and how the system design ensures good indoor air quality. The CO_2 level can be used as indication for the air quality in the room

Provision of air

FIV units can be designed for mixing ventilation or displacement ventilation. However, mixing ventilation with outlets located at the façade is not considered favourable due to the low airflow rates and associated air velocities. Therefore, and because of its multiple advantages, displacement ventilation is preferred. (Sefker, 2008)

The use of displacement ventilation has implications on comfort and design. The principle is better suited for higher rooms. Only sufficient clear height allows proper stratification of fresh, cooler air at the bottom and the impurer, warmer air above as the air volume related to a plume increases with distance from the source. Skistad (1994) identifies room heights above 3.00 m as advantageous and 2.30 m as absolute minimum ceiling height and. Chen and Glicksman (2003) referring to previous research find a bottom line of 2.70 m room height to remove common cooling loads in office buildings.

The vertical temperature gradient as the important parameter of displacement ventilation is constrained by comfort requirements. ISO 7730 sets strict limits to the temperature difference between feet and head to avoid local discomfort. Therefore the vertical temperature gradient should not more than 1..2 Kelvin per meter room height. The sensation of cold feet increases with decreasing room temperature (Skistad, 1994; Chen and Glicksman, 2003). Sodec (2002) denotes a minimum supply air temperature of 20 °C, however referring to German comfort criteria where 22 °C is the minimum room temperature.

The air supply must be positioned as close to the floor as possible. In an adjacent zone the air builds up a "supply air blanket" which spreads over the floor. The air velocity inside the adjacent zone is higher then the face velocity at the diffuser. Skistad has defined the border of the adjacent zone where the air velocity falls below 0.2 m/s. This value is chosen as a comfort criteria related to draught from ISO 7730. Depending on the design air diffuser and the supply airflow rate the adjacent zone can reach out several meters. Most authors on the subject advise a face velocity below 0.2 m/s. (Skistad, 1994)

The face velocity is related to the cross sectional area of the air outlet. The review of data sheets and drawings of FIV units shows that dimensions of grilles meet the requirements to provide the maximum airflow rate at a lower air velocity than 0.2 m/s. Some sheets also state that minimum cross sections are demanded in case of project-specific or on site diffusers.

In the framework of DeAL (2008) one case study with raised floor units was examined where the supply air does not return to the floor. Due to excessive airflow rates the air continues to flow upward along the façade. Displacement ventilation does not occur while draught is induced at the floor.

The supply air temperature of displacement ventilation has to be lower than the room temperature in order to be effective (according to Skistad (1994) at least 0.5 Kelvin). Therefore displacement ventilation is considered ideal for cooling. However, with regard to the above mentioned temperature gradient and the risk of draught maximum 40 to 50 W/m² (depending on author) cooling load can be removed. If the airflow rate related to provide cooling exceeds the minimum airflow rate necessary to supply fresh air then displacement ventilation is not an obvious choice. (Skistad, 1994; Chen and Glicksman, 2003)

Figure 33 shows a comparison of ventilation effectiveness of displacement and mixing ventilation for various temperature differences between room and supply air temperature.

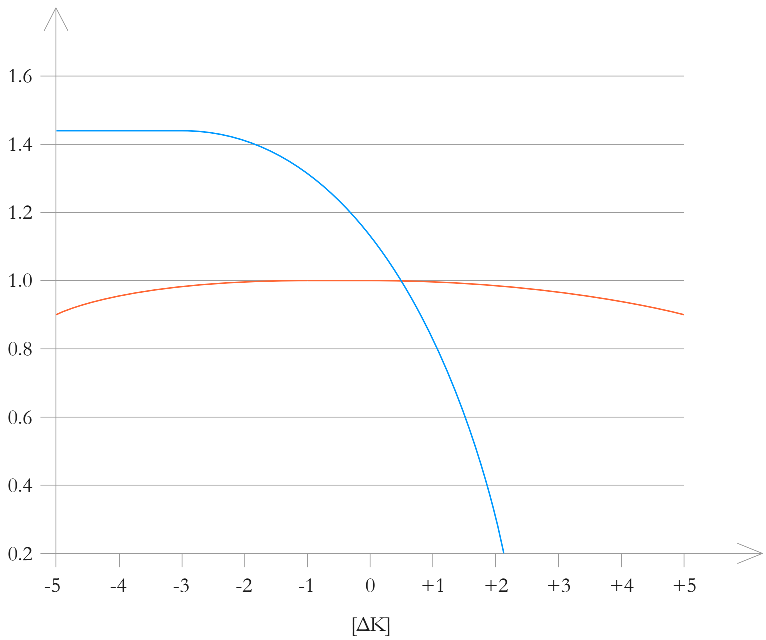

Figure 33: Ventilation effectiveness of displacement ventilation (blue) and mixing ventilation (orange) (after Sodec, 2002)

The ventilation effectiveness of displacement ventilation plunges rapidly if the supply air temperature increases. Mixing and displacement ventilation are even when the supply temperature is 0.5 Kelvin above the room temperature. If the temperature difference increases further then displacement ventilation is less applicable than mixing ventilation. Hence heating with displacement ventilation is not recommended. (Sodec, 2002; Skistad 2003)

In addition, the supply of warm air for heating purposes with FIV entails the risk of short circuiting. The fresh air raises upwards along the unit and re-enters the unit at the extract inlet. (Sefker, 2006)

CO_2 levels

CO_2 concentration can provide an indicator for indoor air quality and is often used as trigger for demand controlled ventilation. Measurements in existing buildings are shown in figure 34. The CO_2 concentration of the outdoor environment was assumed as 500 ppm. (DeAL, 2008)

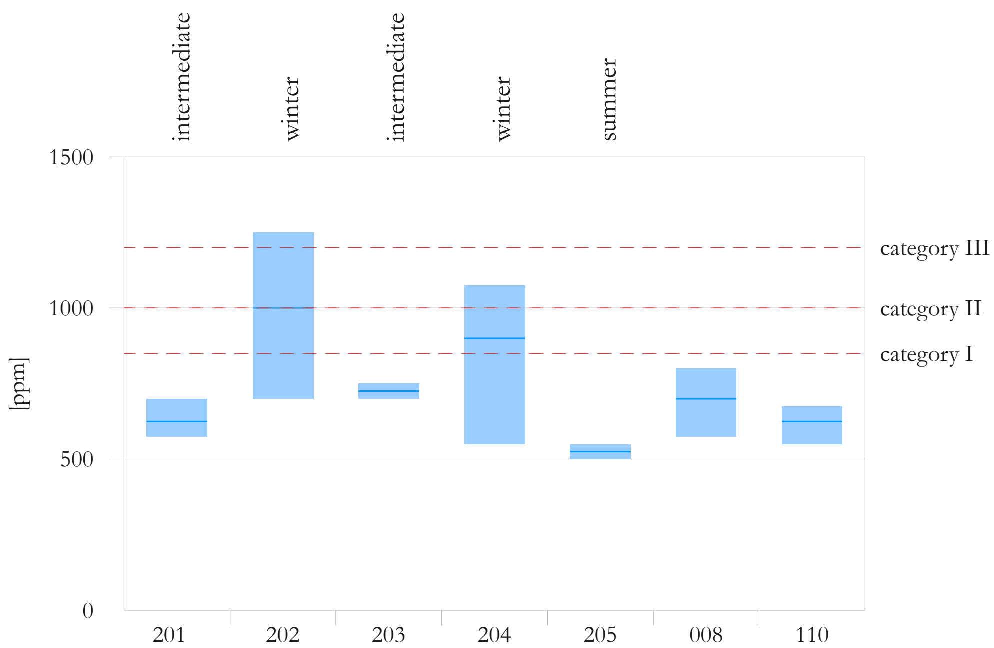

Figure 34: Measured CO_2 concentrations, categories according to EN 15251 (after DeAL, 2008)

In five of seven case studies the range of measured values was below 800 ppm. The average values of the two other buildings were in category II (below 1000 ppm). Reason for the high measured concentrations is high occupancy in both buildings. (DeAL, 2008)

Ventilation rates

Measurements of actual supplied air volumes of FIV units showed that the operative airflow rates were slightly lower than the manufacturer information. In cases of high airflow rates beyond 100 m^3/h deficiencies of up to 10 % were detected. At lower airflow rates the provided amounts corresponded with the declared values of the manufacturers. (DeAL, 2008)

In FIV systems the supply with fresh air can be adopted to the actual presence as the concept of demand-controlled ventilation (DCV) is applied. The full range of sensors to established DCV is available and utilised as the investigations of DeAL (2008) show. The sensors are either integrated in the control units of the FIV units or connected via LON bus. In addition, personal needs can influence the airflow rates e.g. a "silent mode" where the ventilation rate is reduced to a minimum to allow a quiet environment for concentrated work (LTG FVM, 2009). DeAL (2008) concluded that demand control based on presence would demonstrate the biggest advantage of FIV. However, only few of the examined cases studies really employed presence detection. The absence of integrated movement sensors within the units has been faulted.

Centralised systems with demand-controlled ventilation suffer from constraints mainly related to the throttle valves in the ductwork which are necessary to control the air distribution. The dampers in ventilation systems can neither be completely opened nor closed. As a consequence the airflow rate through the damper is limited to a range of 30 to 80 percent of the maximum airflow rate. Furthermore, the control of dampers within the distribution network of a centralised DCV system is very complex as discussed in length by Grini and Wigenstad (2011).

The common practice to deactivate the ventilation outside occupancy is not in compliance with TEK 10 § 13-3. A minimum airflow rate of 0.7 $m^3/(h \cdot m^2)$ is required to remove material and interior related emissions. Grini and Wigenstad (2011) argue that the background for this regulation is not clear. The airflow rate were not related to EN 15251 regarding the procedure described in Annex B.1.2. However, a requirement for an airflow rate of 0.1 to 0.2 $l/(s \cdot m^2)$ can be found in EN 13779 as secondary option to ventilation with two air volumes prior to the occupancy. In the related standard NS 3031 this is taken into account as the operation time for ventilation starts 6:00 to supply fresh air two hours before working hours (Dokka, Berg and Lillelien, 2011). Additionally the minimum airflow rate outside occupancy is required which poses the question why both measures, morning boost and basic ventilation rate are used in NS 3031.

Preventing growth of micro-organisms in the air handling unit is considered a reason for continuous operation. Research reviewed by Grini and Wigenstad (2011) on the alternative operation strategies (continuous vs. night shut-off) shows no clear advantage for one strategy. However, the necessity of regular cleaning and maintenance was highlighted in this context.

These issues do not apply to FIV. Units can run for short periods only outside occupancy which is advised by EN 13779 as third alternative. Running one of the two units for 18 minutes an hour at an airflow rate of 60 m³/h can provide the required 0.7 m³/(h · m²) for the exemplar room, for instance. Furthermore, dampers and ductwork are not present and hence problems linked to them can be neglected.

The influence of the different strategies on the thermal comfort and energy performance can be investigated. The continuous operation is represented in the base case of the "ESP-r" model and will be compared with the alternatives of a morning boost and total deactivation outside occupancy (figure 35).

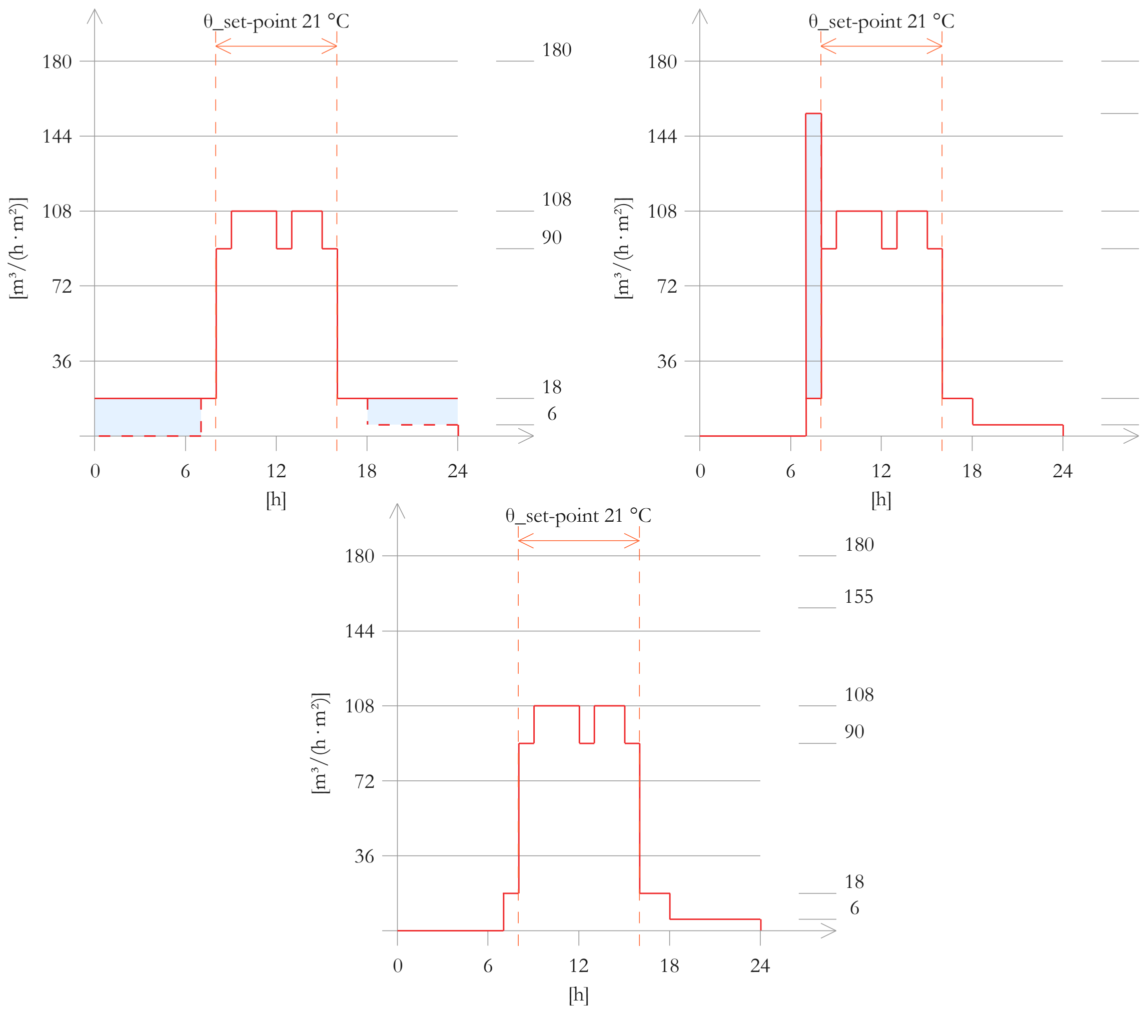

Figure 35: Airflow rate schedules of cases studies

The results of the "ESP-r" simulations for the three cases are presented in figures 36 to 38. The simulations are conducted for the coldest winter day. The summer case does not to be considered as the night cooling strategy would override the settings for airflows outside the occupied period.

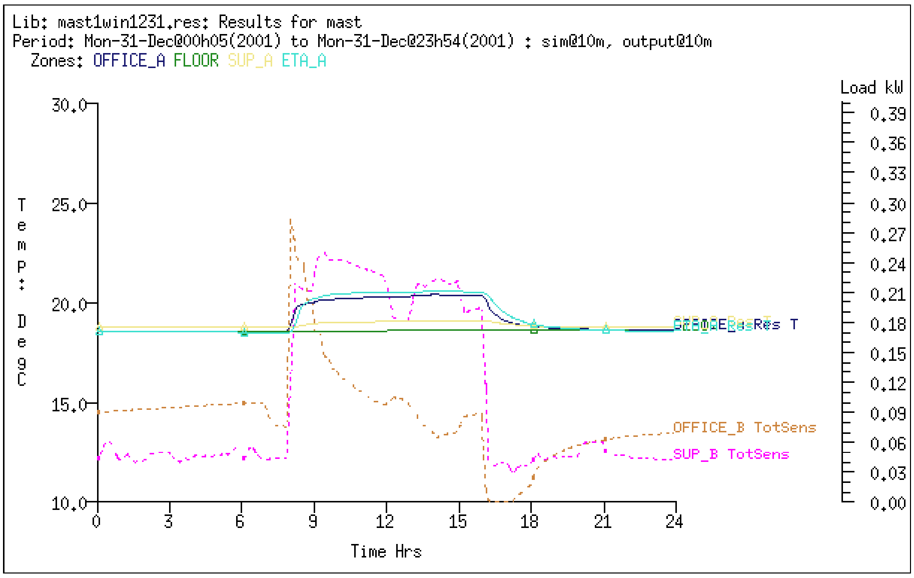

Figure 36: Continuous ventilation (base case), operative temperatures and heating/cooling loads

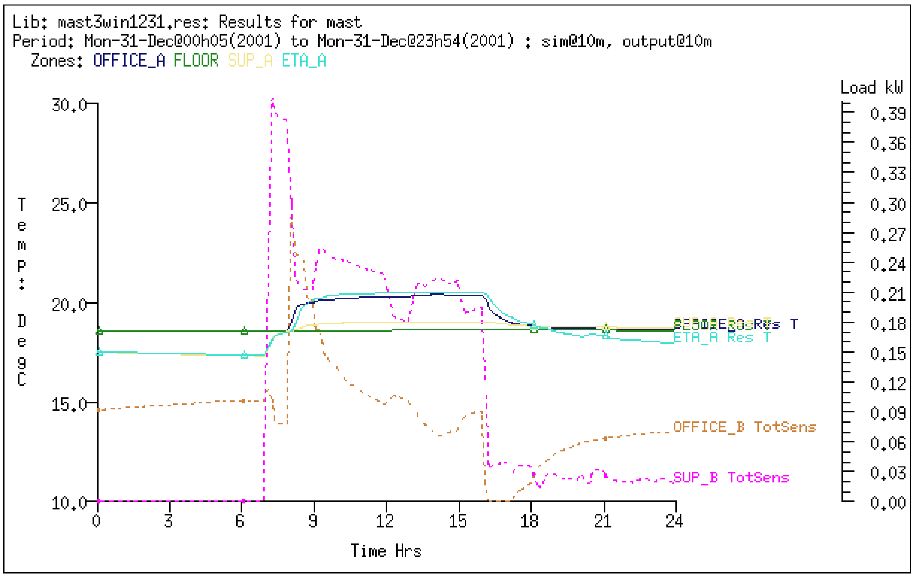

Figure 37: Morning boost, operative temperatures and heating/cooling loads

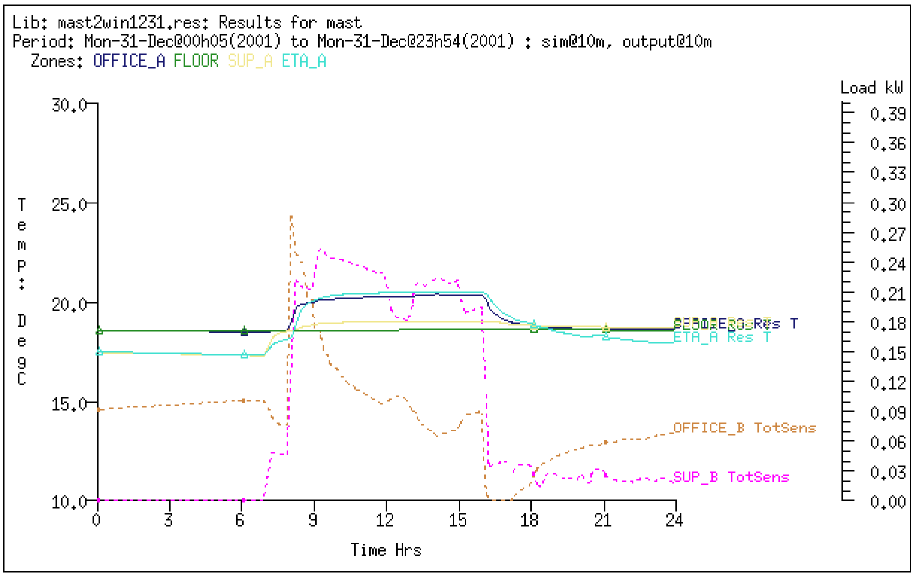

Figure 38: Without ventilation if unoccupied, operative temperatures and heating/cooling loads

The operative temperatures show no significant differences among the cases. All cases display a quick start in the morning and maintain operative temperature above 20 °C during the day. The units cool down during the inactive period in the two latter as noticeable in zones SUP_A and ETA_A (in this case after deactivation during the weekend).

The response to the beginning of heating demand is noticeable in the peak ventilation heating loads and the calculated net-energy demands for the case studies (figure 39). After the morning peak the loads are equal during the day in all cases.

	energy demand for ventilation heating on 31.12. **[Wh/m²]**	**peak ventilation heating load on 31.12.** **[W/unit]**
case 1	190.6	250
case 2	184.4	390
case 3	158.2	250

Figure 39: Daily energy demand and peak load of heating system

Case 3 does not show differences from the base case despite the deactivated ventilation heating during the night. Case 2, on the other hand, requires 1.5 times more capacity to reach the required thermal conditions.

Regarding the required energy for fans no exceptional savings can be assigned to the boost in the morning (figure 40). A reason can be the used schedule where occupancy occurs until midnight.

	supplied air volume per day [m³/d]	**energy demand per day (SFP = 0.6 kW·s/m³) [Wh/d]**
case 1	1098	183
case 2	1037	173
case 3	900	150

Figure 40: Daily air volume and energy demand of fans

In general, the results show that a boost in the morning can reduce the energy demand for heating and fans to the disadvantage of higher heating capacity. The required heating capacities do not pose a problem for FIV units. The similarities between the cases during occupancy shows the possibility of FIV to respond rapidly to scattered local demand of ventilation and heating in combination with a quick heating/cooling system.

System design

EN 13779 characterises requirements linked to the system design such as location of intake and exhaust openings, air filtering, removal of extract air, reuse of extract air, pressure conditions in the system, thermal insulation, airtightness, heat recovery, power consumption, hygienic aspects and acoustic performance. In case of FIV most of them are linked to the integration in the façade and operation. They will be discussed in the respective chapters. Therefore only air filtering, handling of extract air and pressure conditions will be covered here.

Filter classes of reviewed FIV systems show agreement with the requirements in TEK 10 § 13-1 2 d) where at least filter class F 7 is demanded (Appendix B). Usually the filter class can be chosen as required when tendering the units. The design of the air intake is to be solved at the interface between façade design and HVAC design.

The proximity of supply and extract in case of FIV units may pose the risk of short circuits. Therefore the location of the inlet for extract air must be located as far as possible from the supply air outlet. It is recommended practise to place the inlet near the glazing at the window sill in case of sill-mounted units (Sefker, 2006). The responsibility here is strongly linked to cooperation of architectural, HVAC and façade design. However, the removal of extract air as it of concern in EN 13779 refers to leakages in the duct system between the place of extraction and the discharge of the polluted air. This is not a critical issue for FIV technology due to the absence of ductwork. The risk of spreading impurities in the building is minimised by default. On the contrary, stale air is removed and discharged locally which is recommended by the standard.

TEK 10 § 13-1 2 f) allows the use of recirculated air but requires documentation. It is stated that this usually cannot be provided sufficiently which makes application unlikely. It can be assumed that the building code envisages centralised ventilation system where extract of primary areas (offices, etc.) is mixed with the extract of secondary room (bathrooms, etc.) with much higher impurity. FIV units often feature the reuse of extract air for the thermal conditioning of the rooms. Sefker (2006) points out that strictly speaking not recirculating air but secondary air is used. In any case extract air from offices falls in category ETA 1 which is considered suitable for reuse by EN 13779.

Pressure conditions within the system do not create problems since little fan energy is required to distribute the fresh air and there are obviously no pressure losses in the distribution system as there is no ductwork.

Humidity conditions

Users have criticised too dry indoor air in the surveys among tenants (DeAL, 2008). These complaints have been confirmed in measurements where in many cases relative humidity was below 30 percent (required according to the then-valid DIN 1946 part 2) for 3 to 51 percent of the occupied period. In buildings with humidification no problems were noticed. Humidification of supply air has proven to work well in the case studies where it is installed. However, this does not pose a critical issue as the Norwegian building code does not demand criteria for relative humidity.

The “ESP-r” simulations also include latent gains from persons which can be related to the emission of humidity to the indoor air. Hence it can be examined if the recommended range of 25 to 60 % relative humidity (EN 15251) can be reached. In accordance with design instructions in EN 13779 50 W are assigned per person and no other humidity sources than humans and the ventilation air are considered. Figure 41 shows the relative humidity profile for the coldest week from 15th to 21st January as winter conditions are usually the worst case scenario.

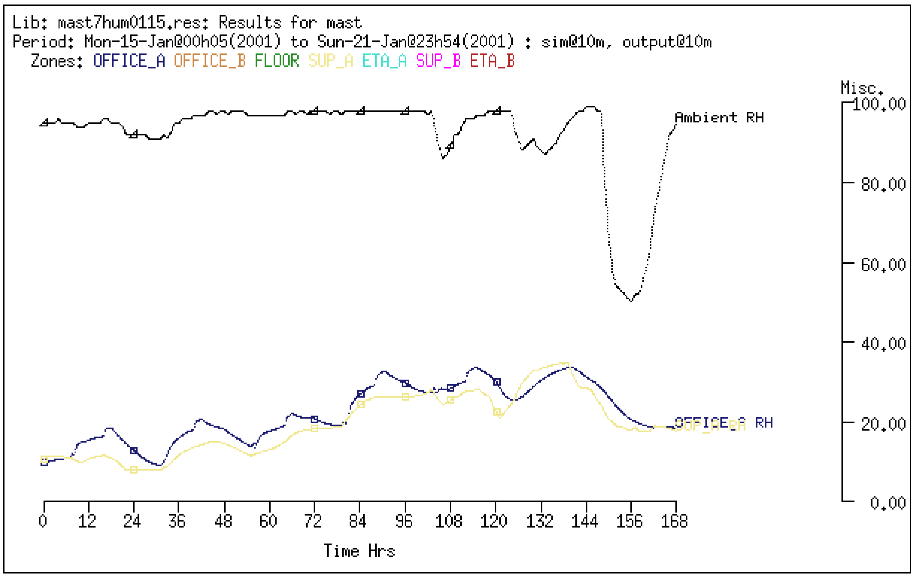

Figure 41: Relative humidity of office and supply air

The daily average relative humidity during occupancy (8:00 – 18:00) has its lowest on Monday with 15.5 % and its highest on Friday with 29.4 %. At the end of the week maximum values above 30 % relative humidity can be reached. However, the recommended value of 25 % in EN 13779 is not accomplished. In a simulation of the coldest winter day (31.12.) the relative humidity ranges does not even exceed 16 % at the end of the occupancy.

It can be concluded that humidification would be desirable to achieve recommended comfort conditions. However, the installation of humidification in FIV units is linked to high effort and usually omitted.

Acoustic environment

TEK 10 § 13-8 allows a maximum sound pressure level $L_{P,AFmax}$ in workplaces of 35 dB(A) setting the absorption factor to 0.2 and the reverberation time T_{max} to 0.2 times the room height.

Data sheets present two values – sound power level L_W and sound pressure level L_P. The sound power level L_W in dB(A) describes the sonic energy of a sound source (in this case the FIV units). The sound pressure level L_P in dB(A), on the other hand, is dependent on the room and the distance from which it is measured. In most data sheets 8 dB noise reduction of the room are assumed. (Emco FLH, 2010) contains a diagram to estimate the difference between sound power level and sound pressure level. For the test room under TEK 10 conditions and a distance of 1.5 meter to the FIV units a difference ΔL of 6 dB is more realistic. In case of only 1 meter distance the difference would decrease to 4 dB. The average L_W of all available data for FIV units has been determined (see Appendix B) as approximately 38 dB(A) for an average airflow rate of approximately 90 m³/h of all examined units. This result refers to the use of outdoor air as supply air. The use of secondary air involves usually higher airflow rates and results in a higher average L_W of approximately 39.5 dB(A). Consequently, theoretical sound pressure levels L_P of 32 and 34 dB(A) can be achieved for ODA-only operation in 1.5 and 1 meter distance which is within the limits of the Norwegian building code.

Measurements under real conditions in building were conducted in DeAL (2008) (figure 42).

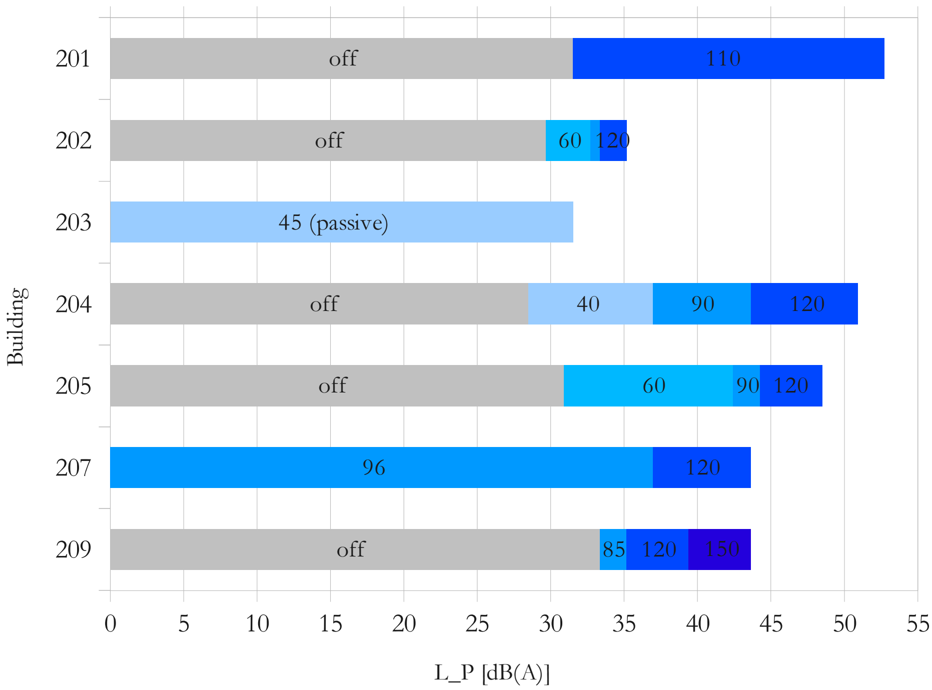

Figure 42: Measured sound pressure levels at different airflow rates (values inside bars) (after DeAL, 2008)

The measured values show very different results from the calculated values. The results show that only three of seven case studies would fulfil the Norwegian requirements where one case studies uses passively driven FIV units without fans. Especially the higher airflow rates beyond 90..100 m³/h exceed the permissible maximum by far. It seems that the location of installation does not allow to draw conclusions on the acoustic performance as, for instance, the buildings 201, 202 and 207 use raised floor units but show very different results.

It has been concluded that the compact size of the units does not allow much space for the installation of sound dampers. Sefker (2006) notes that advantageous conduction of air inside the unit, correct choice of fans, and casings with dampening lining must substitute the absence of dampers. In building 205 sound dampers were not installed at all due to the limited space that was planned for the units in the façade elements. Other reasons for high noise levels are undersized sound dampers or erroneously set airflow rates. However, the measurements in building 202 show that it is possible to stay within the required sound pressure level with an airflow rate of 120 $m^3/(h \cdot m^2)$. In this particular case study the FIV units were installed in the raised floor. Noise problems were detected during the initial usage phase but later eliminated. In general, standard FIV units showed better performance then those which were designed specifically for the project. (DeAL, 2008)

Environmental Impact

Energy efficiency

A FIV system with two different schedules (FIV_a and FIV_b) based on detailed design input parameters is compared with an ideal TEK 10 compliant centralised system. Two reference settings are based on standard values in the corresponding sources NS 3031 and PR 42 (TEK10_DCV, PR42_0). They are adjusted for DCV and primary area. Others reference cases (PR 42_a and PR 42_b) use design values equivalent to the FIV case study for a direct comparison of performance.

Cases FIV_a and PR 42_a will be in the main focus as both allow a direct comparison.

Heat loss

Figure 43 shows the heat loss budgets for the case studies FIV_a and PR 42_a.

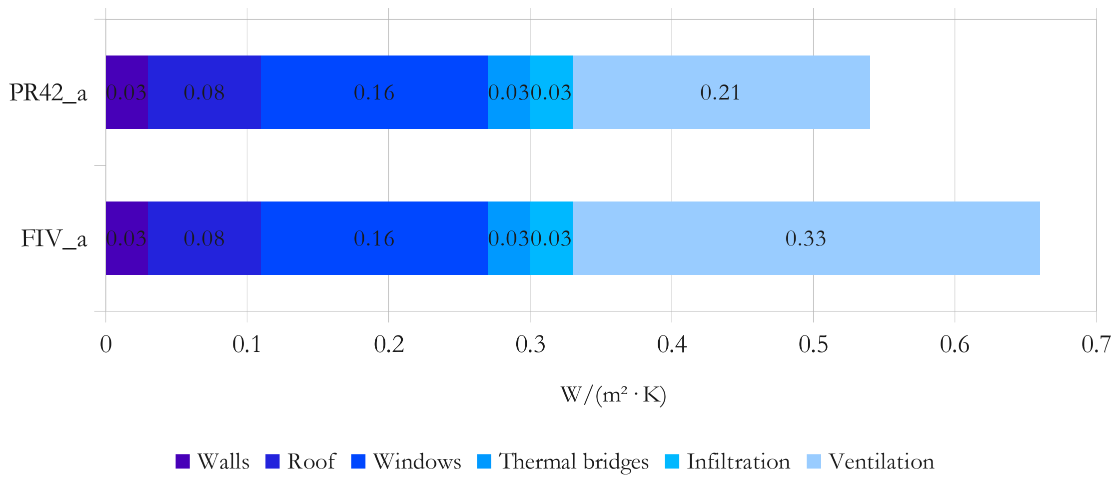

Figure 43: Partitioning of heat losses

A heat loss factor of 0.5 $W/(m^2 \cdot K)$ as specified by PR 42 for office buildings cannot be achieved with either system due to the location of the exemplar room and the resulting heat loss to the roof. Case study FIV_a reaches a heat loss factor of 0.65 $W/(m^2 \cdot K)$ while the reference scenario PR42_a reaches 0.53 $W/(m^2 \cdot K)$. The ventilation heat loss of the FIV system accounts for half of all losses.

Heating / cooling demand

An overview of the simulation results is given in figure 44.

	specific energy demand			specific installed capacity		
	space heating [kWh/m²]	ventilation heating [kWh/m²]	ventilation cooling [kWh/m²]	frost protection [W/m²]	ventilation heating [W/m²]	ventilation cooling [W/m²]
TEK10_DCV	7.8	11.4	3.8	–	19.5	23.0
PR42_0	9.1	9.1	4.0	–	21.8	24.2
PR42_a	6.2	10.5	3.4	–	17.5	19.5
PR42_b	9.4	11.7	3.1	–	17.0	19.5
FIV_a	5.3	24.3	2.4	12.6	36.2	18.7
FIV_b	9.4	20.3	2.7	9.8	36.2	18.7

Figure 44: Specific energy demand and required heating/cooling capacities

Figure 45 shows the annual specific heating demand and installed capacity for ventilation heating at maximum airflow rate of the examined rooms. (In case of the FIV units the ventilation heating capacity is the total of ventilation heating and the capacity of the frost protection)

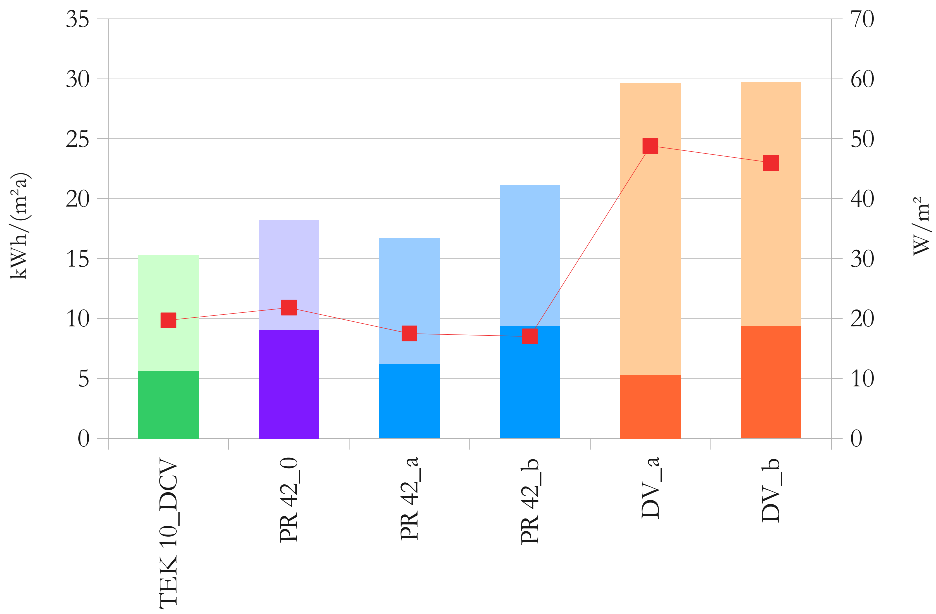

Figure 45: Specific energy demands for space (dark) and ventilation (light) heating

The heating demand of FIV units is significantly higher than centralised systems. The space heating demand is approximately equal for both systems. However, the ventilation heating demand is up to 2.5 times higher due the low efficiency of the heat recovery. This also affects the power requirements. The FIV case studies require 46 to 49 W/m² installed ventilation heating capacity where 36 W/m² account for the heat exchanger compared to the less than 18 W/m² of the comparable centralised system. However, even assumed 50 W/m² installed heating capacity result in approximately 650 W per unit which does not affect the applicability of FIV as the nominal heating capacities for FIV units begin at ca. 800 W (Appendix B). Furthermore, it has to be considered that these values do not include frost protection.

The results for the cooling demand are shown in figure 46.

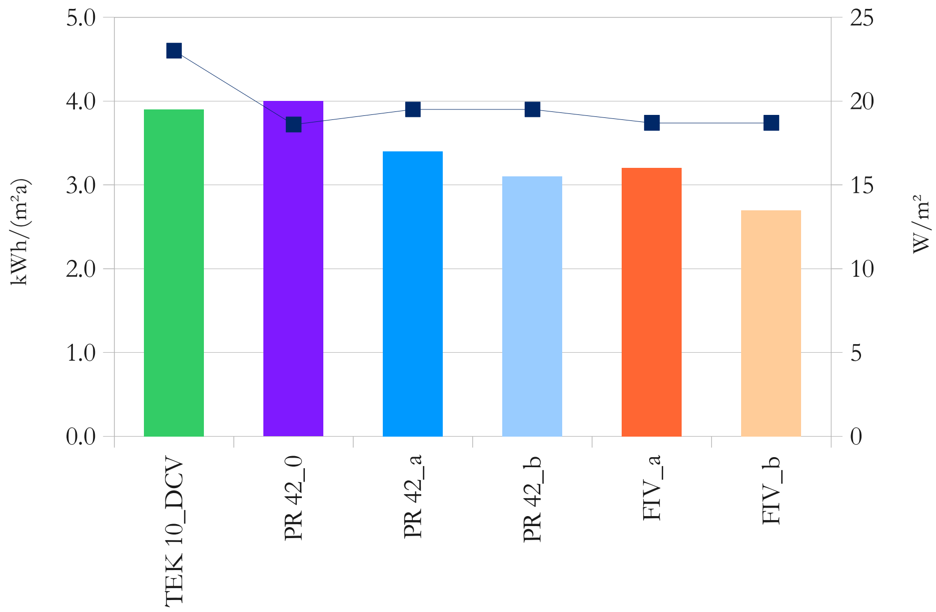

Figure 46: Specific energy demands and capacities for cooling

In general, the cooling demand is very low compared to the heating demand due to the night cooling strategy. The necessary capacities for corresponding centralised and decentralised systems are close to 20 W/m². The dependency of the cooling demand on the transported air volume is apparent comparing the results for FIV_a and FIV_b. If the schedules are optimised to the occupancy as in FIV_b then both, the energy demand for cooling and the installed power of centralised systems is lower than for the comparable case studies PR42_a and PR42_b.

Electricity demand for ventilation

The difference of the specific fan power between centralised and decentralised ventilation system is evident in the results for the electricity consumption of the fans (figure 47).

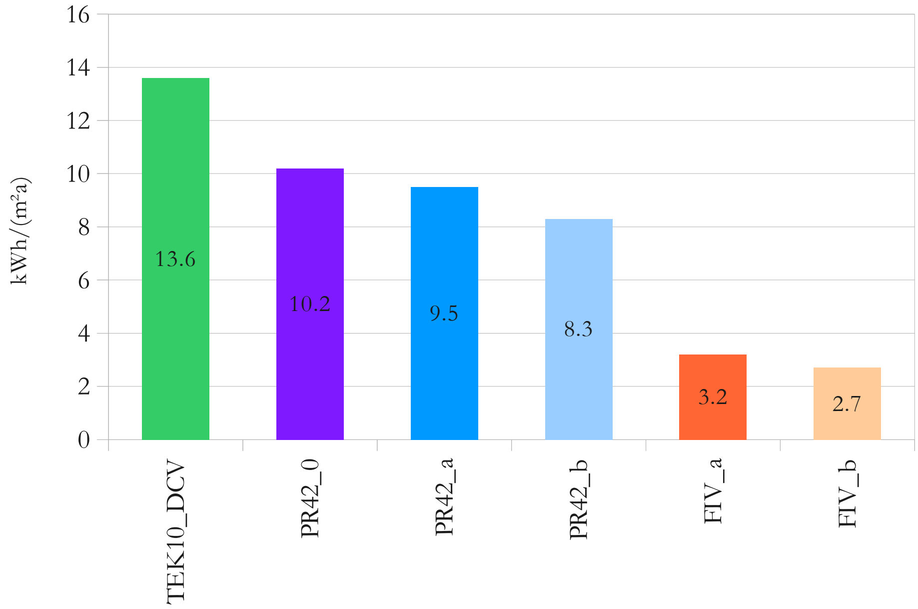

Figure 47: Specific energy demands for fans

Decentralised systems consume on average only one third of the respective centralised system. Advantageous for the FIV systems are the low SFP of 0.6 kW·s/m³ due to the use of energy-saving EC fan motors.

Also noticeable are the quite different results for the case studies of the centralised ventilation system depending on the used input parameters. Using design values shows in general lower demands than the default values. Separate calculations for clearly defined primary and secondary areas instead of assumptions for the whole building (60 % primary area in PR 42 or 65 percent in Dokka, Berg and Lillelien, 2011) might show more refined results.

Net-energy demand

Figure 48 shows the total energy budgets for the investigated case studies.

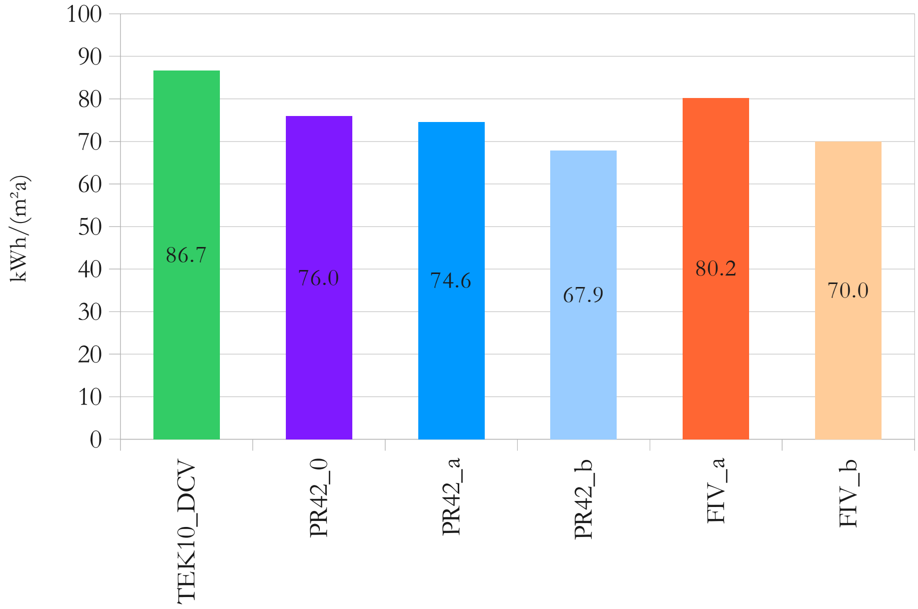

Figure 48: Specific annual net-energy demands

Within the framework of this study the examined FIV case studies show higher net-energy demands than the centralised ventilation systems. The energy demand of FIV_a is 8 percent higher than the equivalent case study PR42_a. However, considering the quick response which is typical for FIV the comparison between PR42_a and FIV_b is possibly more realistic. The reduced occupancy from 12 to 9 hours leads to reductions of 11 to 13 percent energy demand.

The detailed energy budgets for PR42_a and FIV_a are presented in figure 49.

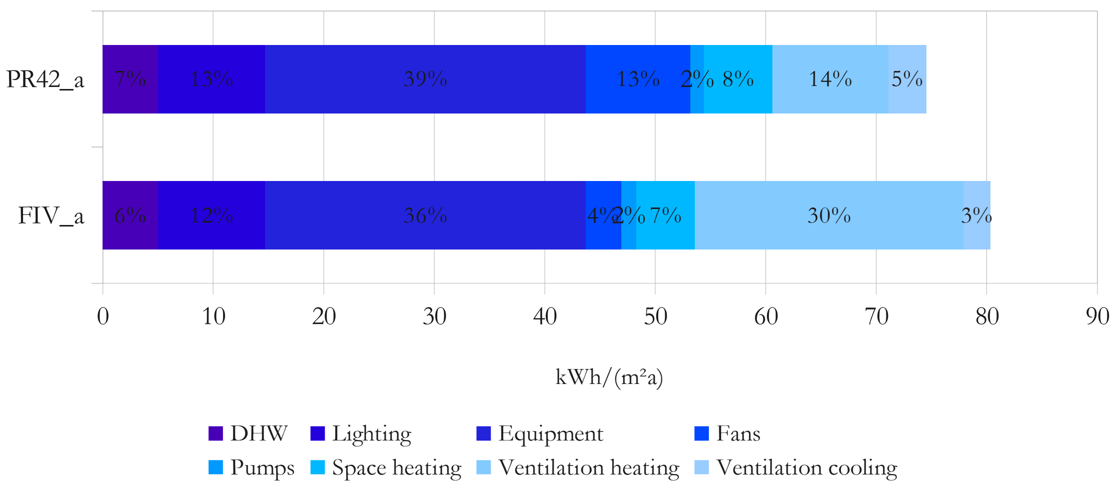

Figure 49: Breakdown of specific net-energy demand

In both cases almost 60 percent of the energy budget is related to domestic hot water, lighting, and equipment where lighting is already adjusted to every single month. In case of the FIV scenario the positive impact of the little energy demand demand vanishes considering the high demand for heating which accounts for 37 percent of the total energy budget. In contrast, the centralised system has both, a low energy demand for fans and heating due to demand-controlled ventilation. Apparent is the little demand for cooling. Reasons may be found in the impact of adjusted input parameters for schedules and internal loads due to demand control.

Heat recovery

Key issue to an improved energy performance of FIV systems is heat recovery. Most FIV units have heat recovery with low efficiencies due to the use of plate heat exchanger. Higher efficiencies have not been requested in previous applications. If a rotary heat recovery would be used then efficiencies can be further increased and frost protection is not required. The integration of rotary heat recovery in FIV is technologically possible as one system uses already rotary heat recovery with up to 62 % efficiency in a 160 mm thin FIV unit (LTG FVM, 2009).

This has been investigated further with case study FIV_a using PR42_a as reference. Figure 50 shows the net-energy demand for FIV_a equipped with either plate heat exchangers or rotary heat recovery with various heat recovery efficiencies. The maximum limit of the plate heat exchanger is assumed as 70 % efficiency.

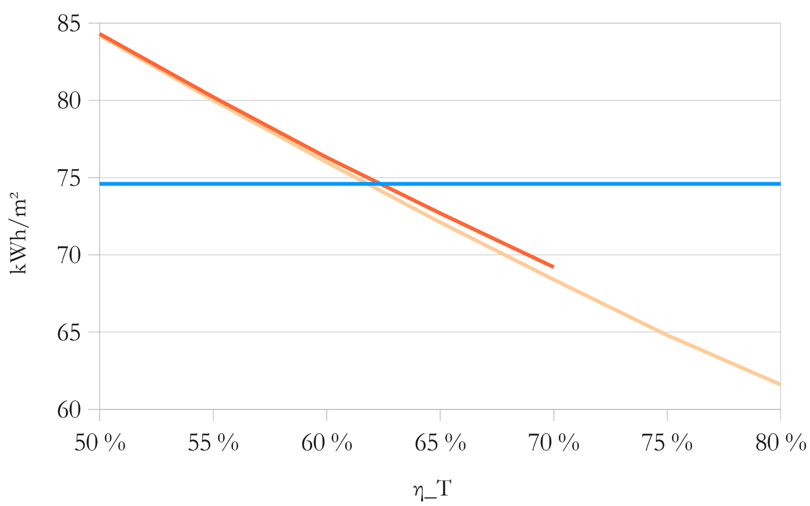

Figure 50: FIV_a with plate heat exchanger (orange) and FIV_a with rotary heat recovery (light orange) compared with PR42_a as benchmark (blue)

If the heat recovery of the FIV case study would reach an efficiency of approximately 62 % then the net-energy demand equals the reference case with 80 % efficiency. A rotary heat recovery unit with 70 % efficiency results in an annual energy demand of 92 % of the corresponding centralised system. With the same efficiency heat recovery as the reference the net-energy demand of a FIV system would be only 83 percent of the reference.

Delivered energy

Figure 51 shows a comparison of delivered energy of the case studies PR42_a and FIV_a using a mixed energy supply of grid electricity and heat pump for water-borne systems.

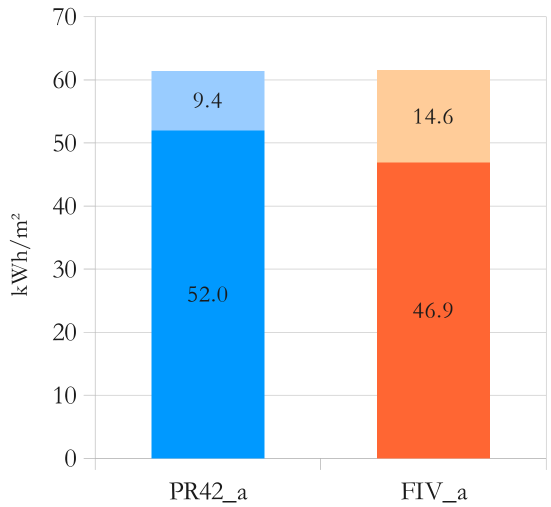

Figure 51: Annual delivered energy

The total specific demand for delivered energy of both systems, 61.4 and 61.5 kWh/m², are approximately equal. The distribution of delivered energy in case of the FIV system shows more demand for the heat pump due to increased water-borne ventilation heating demand. However, the reduced fan energy demand counterbalances the heating demand.

Emissivity of materials

Materials characteristics can influence the energy performance. The use of certified low emitting materials can reduce the ventilation rates significantly from 3.6 $m^3/(h \cdot m^2)$ to the permissible minimum of 2.5 $m^3/(h \cdot m^2)$. For case study FIV_a the required airflow rates for both, normal emitting and low-emitting materials are presented in figure 52. To include the alteration of airflow rates in the "Simien" simulations the approach of mean airflow rates has been substituted by running the simulations demand-controlled based on keeping the CO_2 level below 800 ppm. The annual results of both approaches are in agreement with each other.

	design airflow rates			simulation airflow rates		
	materials [$m^3/(h \cdot m^2)$]	persons [$m^3/(h \cdot m^2)$]	total [$m^3/(h \cdot m^2)$]	chosen maximum [$m^3/(h \cdot m^2)$]	minimum [$m^3/(h \cdot m^2)$]	outside operation [$m^3/(h \cdot m^2)$]
normal emitting materials	3.6	3.01	6.61 (171 m^3/h)	6.94 (180 m^3/h)	0.7	0.7
very low-emitting materials	2.5	3.01	5.51 (143 m^3/h)	5.79 (150 m^3/h)	0.7	0.7

Figure 52: Airflow rates depending on emissions from building materials

It is noteworthy that in the first case the requirements for materials are higher than for persons. The choice of better materials reduces the total airflow rates by 16 percent.

Also a higher heat storage capacity of materials is considered to have an influence on the performance. Therefore the energy demands and the room temperatures of four case studies have been examined with "Simien" following the matrix in figure 53.

FIV_mat,1 normal emitting materials light-weight construction	**FIV_mat,2** normal emitting materials heavy-weight construction
FIV_mat,3 low emitting materials light-weight construction	**FIV_mat,4** low emitting materials heavy-weight construction

Figure 53: Matrix of material-related case studies

The two different heat storage capacities refer to the construction of the internal partitions ("lett vegg" in contrast to "tung tegl"). FIV_mat,1 equals FIV_a as base case representing the normal emitting, light-weight construction.

The resulting heating, cooling and net-energy demands are shown in figure 54.

	specific energy demand		
	heating (space + ventilation) [kWh/m²]	cooling [kWh/m²]	net-energy [kWh/m²]
FIV_mat,1	4.0 + 17.4	3.7	73.0
FIV_mat,2	3.5 + 17.2	3.7	72.3
FIV_mat,3	4.0 + 17.1	3.1	71.7
FIV_mat,4	3.5 + 16.9	3.1	70.9

Figure 54: Specific annual energy demands

The heavier construction does not reduce the heating and cooling demand significantly. The reduced airflow rates lead to reductions of energy demands by 2 %.

The results of the operative temperatures during occupancy are shown in figure 55.

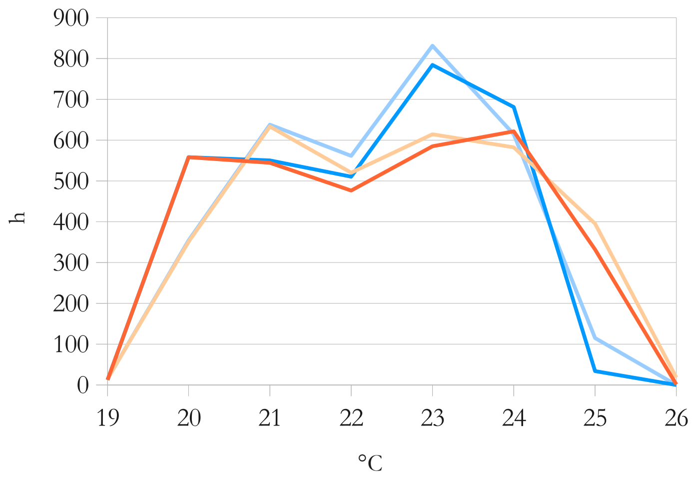

Figure 55: Temperature distribution during occupancy for FIV_mat,1 (light blue), FIV_mat,2 (blue), FIV_mat,3 (light orange), FIV_mat,4 (orange)

The monthly mean, minimum and maximum temperature in room are shown in figure 56. They also include temperature outside the operation hours.

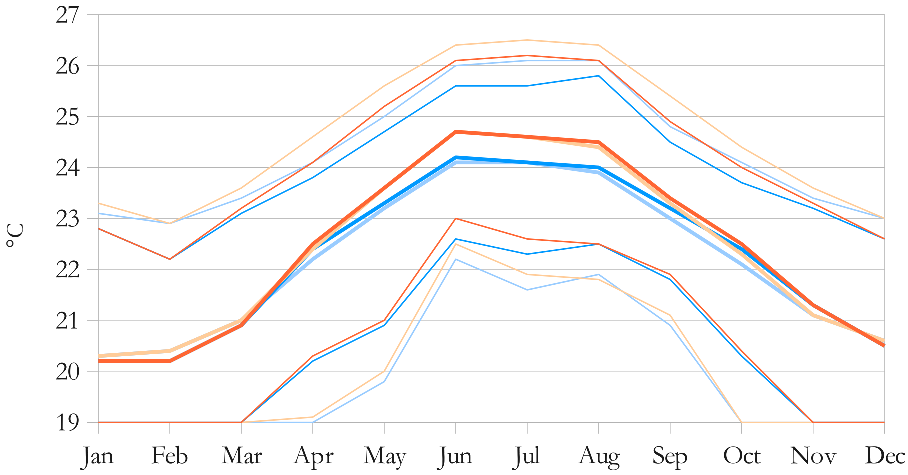

Figure 56: Annual mean, minimum, maximum temperatures for FIV_mat,1 (light blue), FIV_mat,2 (blue), FIV_mat,3 (light orange), FIV_mat,4 (orange)

The graphs show that the variation of minimum and maximum temperatures from the mean temperature is determined by the heat storage capacity. As expected, heavy-weight construction results in higher minimum and lower maximum room temperatures than the corresponding light-weight case studies. Higher mean room air temperatures in summer are the consequence of lower mean airflow rates. During occupancy both case studies with reduced airflow rate reach temperatures above 26 °C for a few hours but less than 50 hours.

Similar results for ventilation aiming at achieving comfortable indoor air quality (as CO_2 was in case of this investigation) are already recorded by Skistad (1994) and Chen and Glicksman (2003) for displacement ventilation. Solely mechanically demand controlled ventilation based on pollutant load is not practicable if the cooling load exceeds the requirements for indoor air quality. If space cooling is to be avoided as in the Norwegian building code TEK 10 then additional measures like natural ventilation with windows or vents or ´reversed´ heating systems (e.g. TABS, radiant floor cooling) may be used.

Emission efficiency

Emissions can be divided in emissions related to operational energy consumption and emission linked to the production of the building materials.

Operational emissions

The annual emissions related to operation per room and per square meter floor area for the case studies PR42_a and FIV_a are shown in figure 57.

	total operational emissions [kg CO_2equiv.]	**specific operational emissions [kg CO_2equiv./m²]**
PR42_a	629	24.27
FIV_a	630	24.30

Figure 57: Specific annual operational emissions

Emission connected to energy demand are linked to the energy supply system. Therefore it refers to the evaluation of the delivered energy. As both case studies are eventually based on electricity which is brought to the building (or produced on site) the difference between the two ventilation systems is neglectable.

Embodied emissions

FIV systems eliminate the need for ductwork, dampers, etc. and lead to reduced sizes of technical rooms, shafts and plena for ventilation (Hirn, 2009). The lesser material use can have a positive impact on the emission balance. As mentioned already a full inventory cannot provided but data for selected parts (ventilation unit and supply network) can be investigated.

Data regarding Global Warming Potential (GWP 100) is available for ventilation systems with heat recovery from the "Ökobau.dat 2011" database (figure 58). Lifetime of the FIV unit and the AHU is stated as 20 years. It must be considered that the central AHU in this dataset is not equipped with any air-conditioning components while the FIV units are full-functioning entities.

	air volume capacity [m³/h]	**GWP [kg CO_2equiv./unit]**	**GWP/(m³/h) [kg CO_2equiv./(m³/h)]**
central AHU	1000	352	0.352
	5000	1408	0.282
	10000	2723	0.272
decentralised	60	29.6	0.493

Figure 58: Specific embodied emissions

Sizing of the supply network will be based on the assumption that the AHU with a capacity of 10000 m^3/h supplies air to 1440 m^2 floor area (120 x 12 meters) using the maximum airflow rate 6.94 $m^3/(h \cdot m^2)$ of the exemplar room. 167 FIV units would provide the same air volume. An average duct size of 315 x 315 mm (12.2 kg/m) and a length of the ductwork of 2 x 120 = 240 meters made of galvanised steel is assumed for the centralised system. For the decentralised systems a 4-pipe system with 4 steel water pipes each with assumed 33.7 mm outer diameter (2.52 kg/m) running along two 120 long façades. Ducts and pipes are not insulated. Alternatively, 26.9 mm (1.63 kg/m) is used as outer diameter. The sizes of ducts and pipes as well as the GWP are taken from the "Ökobau.dat" data sheets (Figure 59).

	unit	GWP / unit [kg CO_2equiv.]	GWP [kg CO_2equiv.]
central AHU	1 AHU with 10000 m^3/h	2723 (per unit)	2723
	$2 \cdot 120 = 240$ m total duct length; 240 m $\cdot$ 12.2 kg/m = 2928 kg total weight of ducts	2.76 (per kg)	8081
		total:	**10804**
FIV units case 1	167 FIV units with each 60 m^3/h	29.6 (per unit)	4943
	$2 \cdot 4 \cdot 120 = 960$ m total length of water pipes: 960 m $\cdot$ 2.52 kg/m = 2419 kg total weight of pipes	2.23 (per kg)	5394
		total:	**10337**
FIV units case 2	167 FIV units with each 60 m^3/h	29.6 (per unit)	4943
	$2 \cdot 4 \cdot 120 = 960$ m total length of water pipes: 960 m $\cdot$ 1.63 kg/m = 1565 kg total weight of pipes	2.23 (per kg)	3490
		total:	**8433**

Figure 59: Global warming potential for supply and distribution options

The comparison of total emissions shows advantages on the part of the FIV technology. The first case study of a FIV system accounts for 4 percent less embodied emission than the centralised counterpart. The second case shows an significant reduction in total GWP and saves 22 percent emission related to materials. Taking into account the absence or reduction of building elements like shafts, ductwork, plena, suspended ceilings etc. the difference between centralised systems and FIV might increase if the inventory would be followed up. However, the inexact character of this estimation must be considered. Further investigation is necessary.

Operation & Life Cycle

Usability

Heiselberg et al. (2002) provide a framework for the evaluation of user interaction between user and ventilation system. It is indicated that direct influence on the indoor environment is highly appreciated by users, increases user satisfaction and consequently also may have a positive impact on the productivity. Critical characteristics are the provision of facilities for interaction, if the interfaces are easy to understand, respond and support user interventions, and have possibilities of overriding in case of a centrally controlled system.

The user interfaces of FIV units for user interaction are mounted in the room at walls or directly integrated in the unit. They usually feature a panel with only a few buttons, knobs and visual displays as colour-LED or LCD. It is possible to take readings of the maintenance status and malfunctioning which also can be sent to a central monitoring system. Some interfaces can be installed in standard flush sockets or have integrated sensors for demand control. A review of typical digital user interfaces shows no differences between the user interfaces of FIV and temperature controls regarding ergonomics. Figure 60 shows an selection of user interfaces which are partly neither specific for FIV nor for the particular unit. It is stated in most cases that any LON bus-capable interface can be used. (Emco FLH, 2010; LTG FVM, 2009; Trox, 2009)

Figure 60: User interfaces and a CO_2 sensor (low right corner) found in data sheets (not proportional) (from Emco FLH, 2010; LTG FVM, 2009; Trox, 2009)

A survey among building operators in DeAL (2008) investigated the usability of FIV units by tenants. The users expressed high satisfaction. In none of the examined 15 buildings operation was described as "too complicated". On the other hand, in nine buildings the operation was "easy to understand" and in five buildings satisfactory "after introduction".

Individual manual control is highlighted as advantage of FIV and implemented as in many cases the room temperature is controlled centrally but can be adjusted locally. However, manual override can cause comfort problems in the rooms as it depends on the competence of the user (Grini and Wigenstad, 2011). DeAL (2008) reports on a case where users instinctively opened the window during hot period. Since a switch to the windows was installed the FIV system was deactivated. This resulted in room temperature above the limit of thermal comfort.

Serviceability

Robustness and redundancy are important features during operation. Redundancy of FIV is provided as the failure of one unit impairs the indoor environment only locally but does not lead to the break down of the ventilation system in the entire building (Sefker, 2006). According to Chen and Glicksman (2003) displacement ventilation presents some challenges regarding robustness. A high level of control is necessary to avoid uncomfortable conditions. In the particular case of FIV, faults are reported linked to wrong arrangement of interior and furniture. Desks were positioned too close to the units respectively the façade. In case of a museum an exhibition concept was agreed upon during the planning phase. The implemented exhibition arrangement disregards the planned conception. However, no shortcomings have been observed. (DeAL, 2008)

DeAL (2008) also compiled problems during operation in buildings with FIV technology. The reported problems linked to a malfunctioning ventilation system do not in all cases address the FIV units. Most issues are related to incorrect control and settings. Inadequate control strategies or the non-compliance with the planned control strategy lead to insufficient comfort and deficient performance. Erroneously set supply air rates and temperatures and badly calibrated room temperature sensors have been registered. Also the interaction with central facilities causes issues such as the provision of supply water with inappropriate temperatures for the heat exchanger. Furthermore, erroneous or missing communication with the central BES did not provide informations about the status of the units. (DeAL, 2008)

Maintenance

EN 13779 provides guidelines hygienic and technical aspects of maintenance and operation of ventilation systems. Resistance to corrosion, cleanability, accessibility, hygienic innocuousness and discouragement of growth of micro-organisms of all ventilation components and parts of the ductwork are highlighted. It is elaborated further, that all components shall be located to be removed for service and cleaning. Service openings shall be provided if this is not possible.

In many cases of raised floor units the system can be lifted out the casing for better access. Hereby is also the possibility given to replace defect units. Furthermore there is also the prospect to install units later on if empty casings and façade air intakes were installed in the first place. (Emco UZA, 2011; Kampmann 2009)

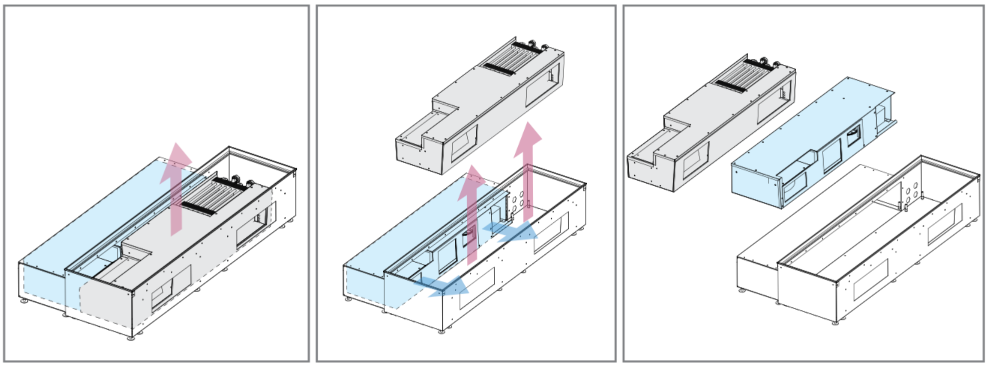

Figure 61: Disassembly of fan unit (blue) or heat exchanger (grey) for service, replacement or upgrading (Emco UZA, 2011)

In the framework of DeAL (2008) the maintenance activities in the studied buildings were examined. They comprise the opening of the unit, releasing and replacement of the air filter and in some cases vacuum-cleaning of the heat exchanger. It has been observed that the regular cleaning not only satisfies the requirements for hygiene but also reduces the energy consumption of the fans significantly. In contrast to centralised units, the work always takes place in the tenant's space. Notwithstanding, spacial requirements for technical rooms or constraints due to access to ductwork in corridors or shafts can be eliminated. In most cases the maintenance work was considered “normal” requiring unskilled personnel and simple tools (screw driver). The operation takes 2 to 10 minutes per unit and is conducted annually or biannually. However, for a building with approximately 800 FIV units the expenditure of time related to maintenance is considered two to three times higher than for a comparable centralised system. (DeAL, 2008)

The interview with Jakobsen (2012) provides a spot-check insight in maintenance procedures in Norway. The revision of the 26 AHUs of the centralised system in “Realfagbygget” with a range of capacities from 16000 to 30000 m³ is conducted circa annually if the pressure conditions require it and takes on average 5 hours per AHU. Filters are replaced and fans controlled and the gratings of the air intakes checked for clogging. A cleaning of air extracts is rarely necessary. The service is done by an external contractor, however, does not require special skills. The personnel in the building is instructed to pay attention and report unusual noises. A cleaning of ducts is not considered necessary since at least two (a coarse particulate filter and a fine particulate filter F 8) respectively up to three air filters succeed each other.

Assuming the use of FIV technology in the "Realfagbygget" 6667 FIV units with each 120 $m^3/(h \cdot m^2)$ maximum airflow rate would be necessary to achieve the same air capacity. If cleaning these would require 5 minutes per unit then 556 working hours (69.5 working days) would be necessary. Compared with the 130 hours of the actual ventilation system this means a 4.3 times higher time expenditure in case of FIV technology.

Costs

Regarding the economic aspects of ventilation systems EN 13779 in Annex B advises to use "the best functioning equipment at the most reasonable costs". Costs are considered here as life cycle costs (LCC) and include investment costs, operational costs and maintenance costs.

Investment costs

Due to lack of response from the manufacturers no latest costs for the units are available. Instead will be drawn on results of DeAL (2008). List prices for 50 FIV units for the year 2006 vary from 1140 to 2400 Euro depending on the integrated functionalities. Prices for units with heat recovery begin at 1700 Euro. Building costs of the façade also may be considered partly since FIV units require special effort for the shaping and locating of air intakes and discharge.

To interpolate to the current year 2012 the price index for office buildings (DESTATIS, 2012) can be used for an estimation. Reference is February 2005 with an index of 100. The price index was given as 102.1 for 2006 and has increased to 118.2 in 2011. Hence costs for FIV units with heat recovery can be estimated as 1970 to 2780 Euro (roughly 15000 to 21000 NOK) for today.

It must be considered however that these prices can decrease significantly. A discount with increased volume of the order can be expected (DeAL, 2008). Also factors like common business conducts in the building branch and interest on the part of the system provider (e.g. pilot projects in new markets) can have an enormous influence on the price.

The investment cost for constructed buildings were investigated and compared with typical building categories in the framework of the DeAL study. No significantly higher cost were detected. The costs for building services were in the range of 307 to 360 Euro (2300 to 2700 NOK) per square meter gross floor area and therefore between "medium" and "high standard" (305 respectively 450 Euro). (DeAL, 2008)

Operational costs

Savings compared to a centralised ventilation system because of a reduced energy demand cannot be directly attributed to FIV technology (see section "Energy efficiency"). However, the costs related to energy consumption are linked to the energy supply options. Here the comparison of the examined supply scenario shows FIV in level with the centralised system.

Furthermore, more energy-saving potential can be assigned to FIV. FIV technology is water-borne where a variety of renewable technologies are available and on the other hand requires little direct electricity for fans. Hence it is in little competition with other operation-linked electricity consumption for equipment and lighting. Also the influence of a straightforward application of DCV can lead to less costs due to the short time delay for response.

Maintenance costs

Franzke et al. (2003) provide an hypothetical overview of maintenance for German conditions in the year 2003 comparing centralised and decentralised system. Both system have shown similar expenditure. In contrast, DeAL (2008) demonstrated an effort two to three times higher than for conventional systems.

A calculation based on data from DeAL (2008) and the interview with Jakobsen (2012) can be conducted to estimate roughly costs in Norwegian conditions. The "Realfagbygget" is equipped with 26 air handling units (AHU) with a maximum total air volume of 800000 m^3/h. The maintenance of one AHU takes on average 5 hours and costs approximately 3000 NOK (Jakobsen, 2012). 6667 FIV units with a maximum airflow rate of 120 m^3/h are necessary to supply the same amount of air. 2, 3, 4, 5 and 10 minutes time needed for maintenance per FIV are compared with the centralised system. (Figure 62)

	total time expenditure [h]	total cost expenditure [NOK]	additional costs [NOK]	relative effort [%]
Centralised ventilation system (26 AHUs)	26 · 5 h = 130 h	600 NOK/h · 130 h = 78000 NOK	–	100 %
Decentralised ventilation system (6667 FIV units)	6667 · 2 min ~ 222 h	600 NOK/h · 222 h = 133200 NOK	55200	171 %
	6667 · 3 min ~ 333 h	600 NOK/h · 333 h = 199800 NOK	121800	256 %
	6667 · 4 min ~ 444 h	600 NOK/h · 444 h = 266400 NOK	188400	342 %
	6667 · 5 min ~ 556 h	600 NOK/h · 556 h = 333350 NOK	255350	427 %
	6667 · 10 min ~ 1111 h	600 NOK/h · 1111 h = 666700 NOK	588700	855 %

Figure 62: Comparison of costs for centralised and decentralised ventilation

Expenditures for maintenance of FIV systems are significantly higher compared to the corresponding centralised ventilation. Even the shortest time needed is still almost twice as expensive as the reference system.

Costs constitute a serious issue. Investment costs in the range of the “upper medium standard” can be justified by other advantages of FIV systems. Maintenance costs, on the other hand, are unarguably high. Especially in a Norwegian context with very high labour costs this represents an insurmountable obstacle.

Lower operational costs because of higher energy efficiency due improved heat recovery could balance the costs in the future. To estimate the influence roughly it can be extrapolated from the test room assuming installed maximum air volume of 240 m^3/h and using the annual energy performance and cost results from “Simien” with 0.80 NOK/kWh direct electricity. Three scenarios are compared: the base case with 55 % and two scenarios with improved heat recovery efficiencies of 65 and 70 %. Five minutes are assumed as time to clean one FIV unit (figure 63).

	energy costs for test room (240 m³/h) [NOK]	energy costs for entire building (800000 m³/h) [NOK]	costs savings [NOK]
Base case $FIV_{2,a}$ (55 % η_T)	1276	4253333	–
Heat recovery efficiency 65 % η_T	1194	3980000	273333
Heat recovery efficiency 70 % η_T	1156	3853333	400000

Figure 63: Energy costs for FIV options with improved energy performance

If comparing these results with the additional costs from maintenance then the energy cost savings can possibly counterbalance the high maintenance costs. However, the energy cost savings can also apply to a centralised unit which would eliminate the advantage ion the part of FIV. A more detailed cost analysis is necessary.

Building Integration

Building design

Space requirements

EN 13779 defines an occupied zone in every room where requirements regarding the thermal environment, the indoor air quality, the indoor air humidity and the acoustic environment are to be met. Occupied zone refers to an area with 0.50 m distance to internal walls, 1.00 m distance to windows and between a vertical range of 0.05 and 1.80 m distance to the floor. The distance to HVAC appliances is defined as 1.00 m. Outside the occupied zone comfort does not have to be guaranteed.

The clearance to windows and HVAC overlaps in case of FIV units as they are located at the façade. However, FIV units project approximately additional 30 cm into the room which also applies to units in the raised floor. Figure 64 shows the resulting occupied zone for the test room.

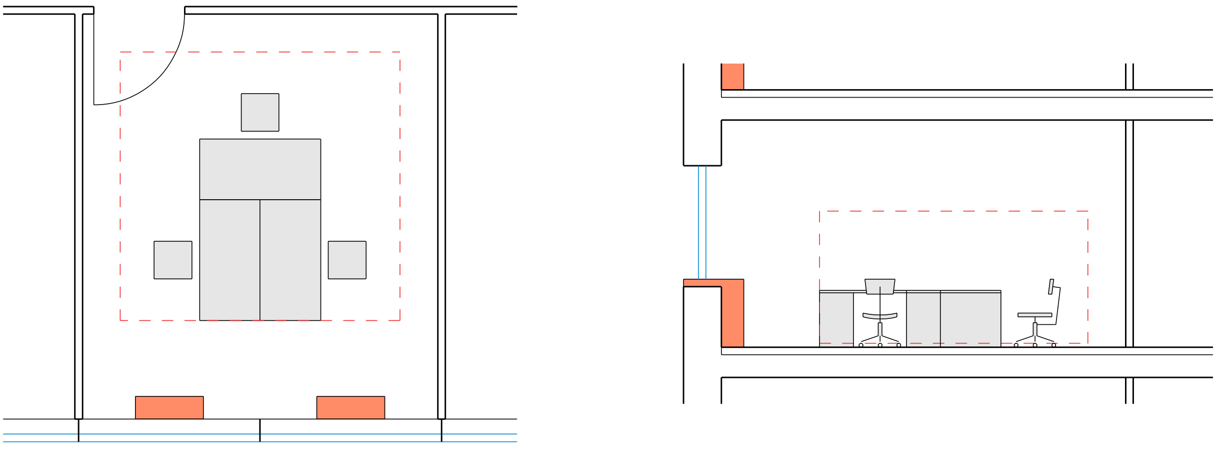

Figure 64: Occupied zone of exemplar room

The space in front of the diffusers must be kept free to allow a correct formation of the supply air blanket for displacement ventilation (For further discussion see Skistad., 1994). Also system providers such as Trox (2009) stress that 1.0 to 1.5 metres clearance is obligatory. This is in particular necessary for the units located in the raised floor where air jets vertically and space is required to allow the air to return to the floor.

While the occupied zone sets only spacial limitations of achievable comfort the clearance challenges an efficient use of space as well as daylighting conditions. The two persons sitting closest to the façade in figure 64 are 2 metres away from the window. According to a simulation in “Radiance” this represents the limit where a daylight factor of 5 can be reached. A person sitting further away would require artificial lighting.

Other spacial limitations refer to room depth and room height. The depth of the room is limited to 6 metres to facilitate a return of the impurer air to the FIV units (Trox, 2009). Therefore the building depth is limited to approximately 12 metres or additional ventilation system have to be installed in the core area.

The clear height is defined by the principle of displacement ventilation. Authors agree that the higher the room the better is the functionality of displacement ventilation but differ regarding the minimum height. Skistad (1994) states 2.30 m as minimum and 3.00 m as optimum, Chen and Glicksman (2003) refer to a eight feet minimum (2.44 m), Sodec recommends a height of approximately 3.00 m which is also demanded by Trox (2009).

TEK 10 § 12-7 requires a minimum height of 2.70 m for Norwegian offices which leads to a minimum floor height of 3.60 m according to Byggforsk (2004) to provide enough space for installations in the suspended ceiling. FIV does not require a suspended ceiling which can compensate for the higher room height. (For benefits of high ceilings in Norwegian context see also Dokka et al., 2003 and Høseggen, 2008). That FIV in general safes floor height and may allow an extra storey has been revised by DeAL (2008) given that the compared buildings with centralised ventilation systems use a ´slender´ HVAC concept without suspended ceilings, yet with TABS.

Location of installation

A review of built examples in DeAL (2008) showed that façades with all types of fenestration (single windows, strip windows, fully glazed, double skin façade) whether built on site or prefabricated as façade elements are apt for FIV units. FIV has been used also in the context of a listed building (German Historical Museum in Berlin) and for revitalisation to adapt to correspond to a contemporary level of comfort and appearance.

Figure 65 shows the façade sections of two buildings with retrofitted façades. The building were erected in the 1960's and 70's. The façades were stripped off so that only the structural system remained. The interior and the previous installations were removed.

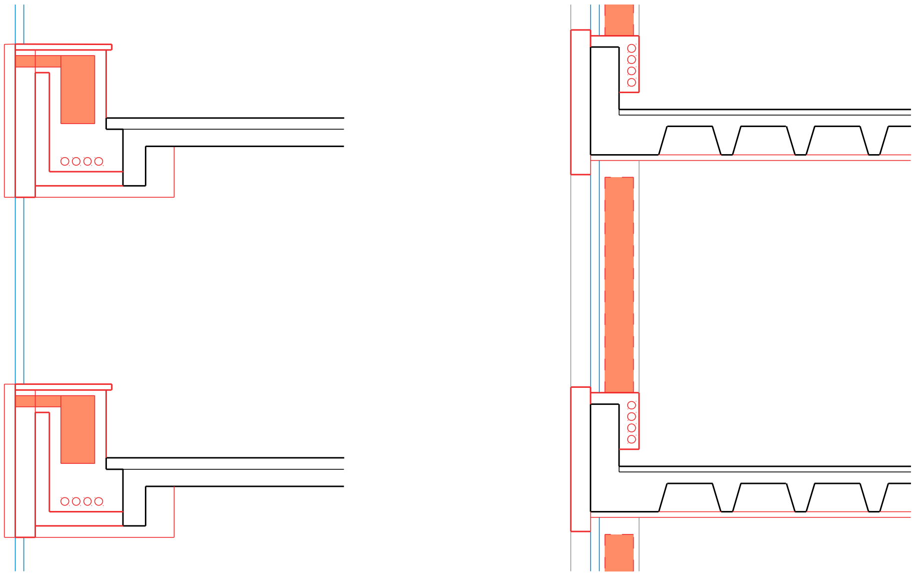

Figure 65: Façade sections of two refurbishment projects (new structures in red) (after Trox Lighttower, 2006 and Trox Feldbergstraße, 2009)

In the left case L-shaped spandrel elements were attached to the old façade. These prefabricated elements comprise the pre-installed casings of the FIV units and the new technical infrastructure. In the right case the massive sill allowed no intervention. The FIV units are designed as part of the strip window façade and received casings with a perforated pattern on the inside and outside to highlight the special functionality. In both buildings the floor heights are approximately 3.00 and 3.15 m which did not permit the use of a raised floor or a suspended ceiling in order to guarantee a sufficient room height. Therefore the façade was the only feasible location for the installation of building services to fulfil contemporary demands. (Trox Lighttower, 2006; Trox Feldbergstraße, 2009)

In several examples FIV units are integrated in fully glazed double skin façades. The advantages are the lesser influence of outdoor conditions on façade and the possibility of preheating the incoming air in the cavity. Measurements of the air intake temperature in the case studies conducted by DeAL (2008) showed average over-temperatures between the intake temperature and the outdoor temperature of 2 Kelvin during the heating period and 3 Kelvin during the cooling period.

EN 13779 gives recommendations for system requirements which are addressed in TEK 10. The location of intake and exhaust, thermal insulation and airtightness are those related to building integration. TEK 10 § 13-1 2 d) advises to locate intake and exhaust in such a manner that impurities from the discharged air cannot re-enter through the fresh air intake. The limited size of the units may pose issues with respect to the proximity of intake and discharge. Table A.2 in EN 13779 annex A provides recommendations for minimum distances between intake and exhaust. The formula is as follows for the case that intake and exhaust are located within the same façade where the intake is located below:

$$2 \cdot l + \Delta h \; > \; 0.308 \cdot \sqrt{q_V}$$

where:

l … distance between the centres of vents,

h height difference between the centres of the vents,

q_V discharge airflow rate in l/s.

If using 120 m³/h respectively 33.3 l/s as q_V and neglecting the height difference then the allowed distance between the opening can be approximately 90 cm. Measurements in drawings of the units at hand show agreement with this figure. It can be seen that small distances between the openings are possible due the low airflow rates. (see data sheets Emco, Kampmann, LTG, Trox)

Sufficient air volume under the cover and a closed casing on front and sides shall provide enough protection from intruding rain and snow according to TEK 10 § 13-1 2 d). Figure 66 shows an overview of schematic air intake (and exhaust) situations found in literature (scale approximately 1:100).

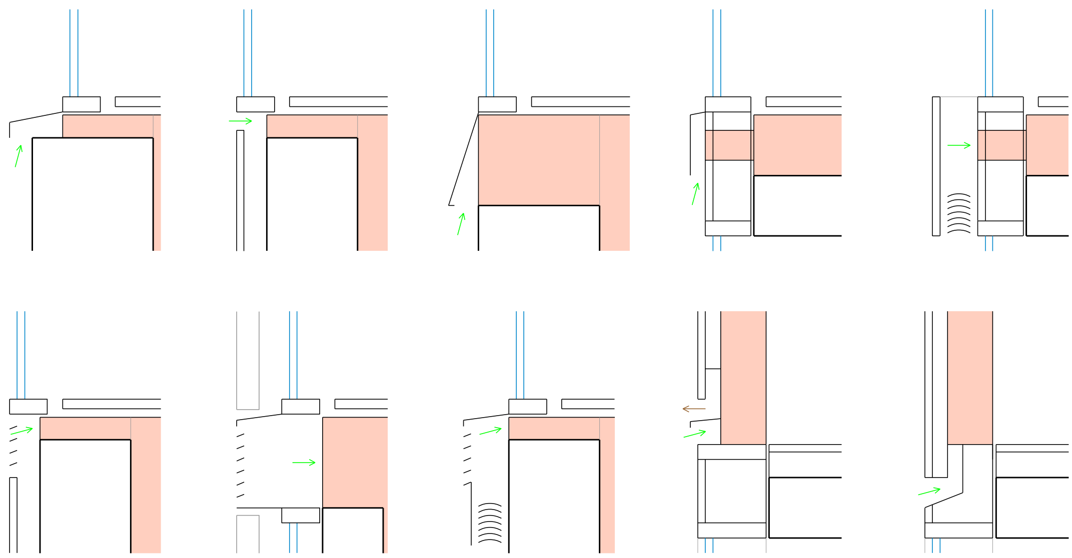

Figure 66: Examples of air intakes (after DeAL, 2008; Emco, 2009; LTG FVM, 2009; various Trox brochures, 2002-9)

The examples demonstrate a diversity of intake designs. The designs are not only adapted to the chosen type of FIV unit yet also an integral part of the full façade design (see the integrated design in conjunction with shading devices in three examples). However, not all designs would comply with the requirements of TEK 10. Especially the demand for a front which is 30 cm prolonged below the opening is not fulfilled in most cases. Such intervention would considerably intervene with the elaborate scale of intakes and protective elements.

The design of ventilation systems and their openings can be a source of considerable heat loss in otherwise super-insulated buildings. Problems with airtightness have been reported in DeAL (2008). In one case the installation in the façade was not executed correctly. Höptner (2001) investigated leakages also between the path of supply and extract air within residential FIV units and discussed the consequences for the quality of the supply air. Amendments to units were demanded.

Thermal insulation depends on the location of the FIV unit. Units mounted against the sill or located in the raised floor usually penetrate the façade only for the air intake and exhaust (total area approximately 0.2 m^3). On the other hand, units mounted in the jamb or in the wall require insulating measures as they expose more surface to the outside. In case of wall-integrated units two solutions are used. Either the systems are located on the inside of an insulating curtain wall system or vacuum insulation panels (VIP) are used. However, curtain walls usually do not have insulating levels as opaque exterior walls. Regarding the use of VIP must be considered that a considerable thickness is required in case of an super-insulated envelope like for the exemplar room in this study. 7 to 8 centimetres thick VIP are necessary for the U-values of the exterior of 0.08 $W/(m^2 \cdot K)$ considering 0.007 $W/(m \cdot K)$ as the certified thermal conductivity of VIP (Porextherm, 2011).

Building programme

Adaptability

As the usage of the spaces might change over the building's lifetime also the ventilation system has to adapt to various occupancies. Byggforsk (2004) identifies three main principles of adaptability. Elasticity allows shrinking and expansion of units (e.g. dividing and merging of tenants spaces) within an existing structure, generality allows various activities and usages within an unchanged room, and flexibility is referred to as the possibility of effortless alterations by use of plan modules, for example. It is states that 60 % of costs and measures related to alterations are related to building services. Therefore two flexibility strategies are proposed in Byggforsk (2004) – moveable installations with fixed capacity and fixed installation with variable capacity. The associated oversizing of technical systems is considered a recommended concept.

Decentralised ventilation, on the other hand, offers the possibility to subsequently upgrade or replace the FIV units according to the demand if the empty casings are installed during the construction (Sefker, 2006) (see also figure 61). Elasticity is provided since unleased spaces are only ventilated and conditioned to a necessary minimum and the billing can take place for each single tenant.

Generality can be investigated for the exemplar room based on the requirements on the ventilation system in EN 15251 and TEK 10. The base case assumes three persons in the room which required roughly 180 m^3/h airflow rate. Other possible occupancies are the same office with less occupancy (two persons), two combi-offices with one person each, an highly occupied office with four persons, an office with very low occupancy, a meeting room with 10 persons (figure 67).

Figure 67: Variations of occupancy in exemplar room

Figure 68 summarised the resulting airflow rates for the configurations.

	occupancy [persons]	required ODA volume [m²/h]	unit supply [m²/h]
occupancy 1 (base case)	3	$3 \cdot 26 + 25.92 \cdot 3.6 = 171.31$	2 x 90
occupancy 2	2	$2 \cdot 26 + 25.92 \cdot 3.6 = 145.31$	2 x 75 (or 60 + 90)
occupancy 3	1 + 1	$2 \cdot 26 + 25.92 \cdot 3.6 = 2 \times 72.66$	2 x 75
occupancy 4	4	$4 \cdot 26 + 25.92 \cdot 3.6 = 197.31$	2 x 100
occupancy 5	1	$1 \cdot 26 + 25.92 \cdot 3.6 = 119.31$	2 x 60
occupancy 6	10	$10 \cdot 26 + 25.92 \cdot 3.6 = 353.31$	4 x 90
unoccupied	–	$25.92 \cdot 0.7 = 18.14$	1 x 60 for 18 min/h

Figure 68: Occupancies and air supply per FIV unit

Most of the different occupancies can be covered with a small set of airflow rates which correspond to the common pre-defined factory settings. The quick responding character of FIV facilitates that these results apply to both, permanent changes of occupancies and sudden changes of occupancies e.g. because of visitors in the office or people leaving the room.

Typology

Changes in the organisation of work led to the requirement for a different physical built environment. Because of the desire for flexible, process-oriented workplaces and economical considerations the open plan is an often used office typology in Norway (Byggforsk, 2001).

According to EN 15251 the open-plan office has different requirements then a cellular office. Ventilation rates and acoustic criteria differ meaning that a lower level is considered as sufficient. However, TEK 10 does not differentiate between the typologies and sets the same demands for both.

The study above shows that higher airflow settings are required for the open plan with high occupancy. Considering the acoustic issues of FIV units detected in DeAL (2008) this may lead to non-compliance with TEK 10 as the same acoustic requirements apply to open plan. On the other hand, EN 15251 recommends different permissible sound pressure levels for cellular offices, L_P = 35 dB(A), and open-plan offices, L_P = 40 dB(A).

The thermal comfort for the hottest day was simulated in "ESP-r". Therefore the inter walls were substituted by a fictitious material representing air and the internal gains and airflow rates adjusted to an occupancy of four persons (6.48 m²/person). (Figure 69)

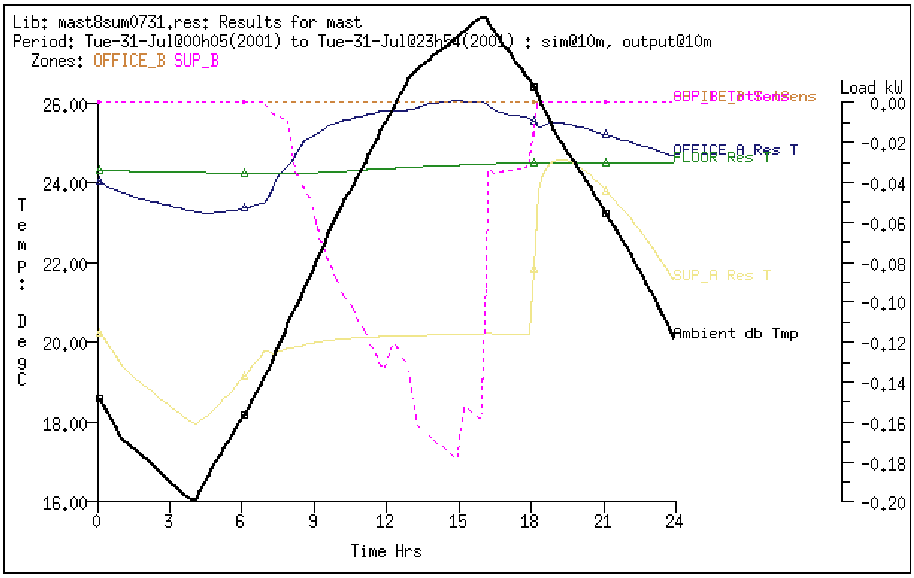

Figure 69: Exemplar room as open-plan office, operative temperatures and cooling loads

The operative temperature in the office reaches 26.1 °C in the afternoon. The room temperature is 0.5 Kelvin higher compared to the base case with three persons. The night cooling strategy ensures that no differences regarding the temperatures outside occupancy can be detected. In general, no decreasing of comfort can be identified for the simulated FIV.

Figures 70 and 71 show the convective heat fluxes on opaque interior partitions (ochre) to other zones and surfaces facing the exterior (blue).

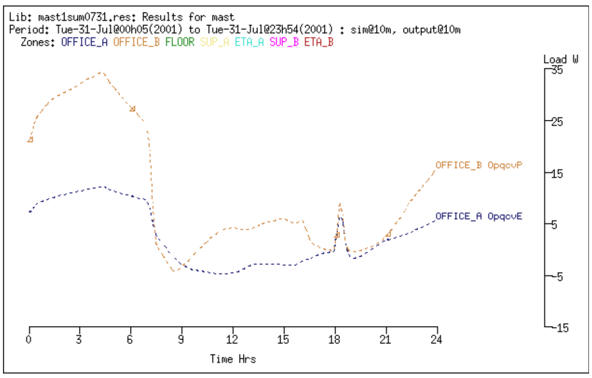

Figure 70: Base case, convective heat fluxes on opaque surfaces

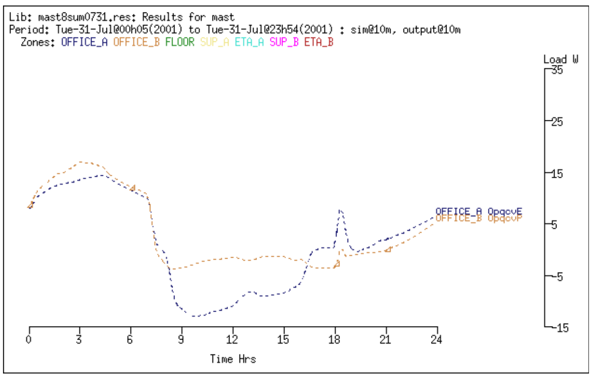

Figure 71: Open plan, convective heat fluxes on opaque surfaces

In the base case can be seen that heat is released from the surfaces to the room. It is evident that more heat is discharged outside than during occupancy. On the other hand in case of the open plan, only very little heat is emitted to the room during occupancy. Instead the heat balance to the exterior surfaces changes as the increased internal gains force the envelope surfaces to charge heat.

Individual control in open-plan office is usually considered difficult as many people have many sensations of thermal comfort. Therefore an automatic control ministering to the occupants is advised overseeing room temperature, airflow rates and scheduled operating hours (Heiselberg et al., 2002). More recent investigations on possibilities of adaptive comfort in a open-plan office with high occupancy in London by Barlow and Fiala (2007) show that the disposition to variations of thermal comfort was more critical than the indices calculated according to ISO 7730. Possibilities of intervening in the centrally controlled heating/cooling system was higher evaluated than individual measures. On the other hand, there is a significantly high support to open windows manually. The tolerance of the users towards draught was higher than the respective calculated draught rates. A decentralised ventilation system may provide individual control of ventilation rates but can also respond to the desire of openable windows. If linked to a window switch then energy savings can be expected as the affiliated FIV unit would be deactivated.

This conceivable control strategy differs from case studies in DeAL (2008) with larger office spaces as those mainly use centrally controlled ventilation rates and a central or a floor-wise control of room temperature. Yet the case studies show a comparatively good energy consumption. In these two cases the concept of centralised extract is utilised. The absence of partitions allows a proper airflow to the extract vents.

4. Discussion

The reported satisfaction of users with FIV is generally good and the indoor air quality is mostly perceived as satisfactory. This may be related to the advantages of displacement ventilation used. Yet, the benefits of displacement ventilation depend on a small range of variances of air supply and distribution parameters. Heating and cooling of the ventilation air in the same day is necessary in the intermediate season and in summer to keep the supply air temperature as stable as required for displacement ventilation. Hence only a 4-pipe system for the heat exchanger of FIV units can be recommended. Local discomfort in form of draught may be sensed if the supply air temperature is not within the narrow band of 0.5 to 2 K below room temperature. The application of displacement ventilation is limited to cases where loads from persons and building materials outweigh loads from heavy contamination or thermal conditions. Heating with displacement ventilation is commonly not recommended.

Thermal comfort can only be achieved with a supplementary heating/cooling system. Ventilation preheating during the night which is often suggested in connection with FIV shows insufficient results for Norwegian conditions in this study. On the other hand, planar systems like TABS and radiant floor heating show very good results with regard to comfort and required capacities. Both can also be used for cooling in summer with little effort. Convective heating can provide a quick responding system. However, the interaction with the airflow of the displacement ventilation needs further research.

Key concept of FIV is to deactivate ventilation in the room during absence of persons even only for short periods if the respective presence sensors are installed. This is in accordance with the former German standard which allocated ventilation rates solely to persons but not with the Norwegian building code where a minimum airflow rate outside the occupied period is demanded. In centralised ventilation systems the layout of dampers limits the possibilities of reducing the airflow rates to the minimum which leads to either inefficient operation at too high airflow rates or the common habit of not code-compliant night shut-off. In contrast, FIV can be code-compliant by utilising the possibility to activate the air supply for short periods in accordance with EN 13779.

Noise problems are a serious issue as the small size of FIV units does not permit sufficient acoustic shielding of occupants from the sound created by the fans. Acoustic conditions in only two of seven measured buildings would be acceptable according to the Norwegian building code. In one of these two cases only a passive unit without fan was used. An assessment of expected acoustic performance based on data sheets or assumptions related to the location of the units does not correlate with the measurements which show rather project-dependent results.

The tendency of buildings with FIV being more energy-efficient than buildings with centralised ventilation systems as stated for Central Europe cannot be confirmed with simulations in Norwegian conditions. Critical issue is the higher ventilation heat loss due to an insufficient efficiency of the heat recovery. Consequently neither a net-energy demand competitive with centralised systems nor compliance with TEK 10 § 14-3 regarding the demanded efficiency of 80 % can be accomplished. So far FIV units have been developed for Central European climate where the focus is more on coverage of cooling demand in office buildings. However, if the efficiency of heat recovery in FIV units would improve by approximately 10 percent to 65 percent then the same net-energy demand as a centralised system with 80 percent efficiency could be achieved. If both have 80 percent efficiency then the net-energy demand of a decentralised system would be 17 percent lower than the centralised system.

Regarding the delivered energy the façade-integrated ventilation can be comparable to a centralised system if an energy supply strategy is chosen which focusses on renewable sources for the water-borne supply of the FIV system. This also applies accordingly to the emissions related to operation. In case of building materials there is the tendency to comparably lower embodied emissions for FIV despite the extensive supply network. The omission or reduction of elements which are features related to centralised systems may reinforce this tendency. However, an expanded inventory is necessary for conclusive results.

Individual control of the indoor environmental is possible with FIV if the control facilities allow. The complexity of the user interfaces is reported low enough to enable non-expert users to adjust parameters. Also the demand for energy-efficient hybrid ventilation in combination with individual window ventilation can be handled exemplarily with FIV as an installation of switches to the window would deactivate the respective FIV units. However, local discomfort may be expected if users are not familiar with the handling of environmental controls and the consequences of their interventions.

Problems in decentralised ventilation systems are rarely linked to malfunctioning of the FIV units themselves. Reported reasons for deficient performance are mainly inadequate implementation of control, erroneous settings as well as lack of integration in the central building management. Unlike centralised systems failures do not affect the entire ventilation system as issues with single units can be fixed locally.

Displacement ventilation performs effectively only if the formation of the specific airflow patterns can be ensured. It must be presupposed that the arrangement of interior is in accordance to a planned layout or corresponds to agreed and known rules. Problems with erroneously placed furniture and too high occupancy can inhibit a regular airflow in the rooms.

Maintenance is faced with the challenge of the large number of units. Although only simple tools and unskilled personnel are needed the cleaning and service procedures of the units take 3 to 5 times longer than a comparable centralised ventilation system.

This has serious implications for the life cycle costs. Investment costs for building services in total in new buildings are in the range of upper medium standard for the German situation. Due to the immense time expenditure also the maintenance costs can be assumed as indisputably high. In Norway this may be more apparent due to the high labour costs. The total cost balance may constitute a serious drawback as no savings of operational costs can be assigned to FIV units with the current technology.

Spatial requirements for FIV derive mainly from the demands of displacement ventilation which entails limited room depth and minimum room heights and clearances to the air supply vents. The resulting usable area is mostly congruent to the occupied zone according to EN 13779. However, floor area must remain unused and the distance from façade may cause issues with daylighting due to the required clearance in front of the FIV units.

Floor heights in buildings with FIV are similar to advised floor heights in Norway, while buildings with FIV have higher ceilings. Higher room heights especially in conjunction to a discarded suspended ceiling can lead to enhancements of thermodynamic properties of the room, more possibilities of daylighting and no dust accumulation in the ceiling space and at the ductwork. A more spacious feeling of the room may also stimulate user satisfaction.

Design of ventilation system and façade are strongly interlinked. The location of FIV units within the façade can be chosen freely depending on project specifications or design intentions. The visible parts of the FIV systems can be integrated as architectural element into the overall façade design. This means that a proper functionality of the FIV system is not only part of the HVAC design but also depends on details in other system e.g. the design of air intakes, insulation, airtightness, etc. The FIV unit alone does not provide all solutions for its functionality. Cases have been reported where inaccurate planning leads to inferior performance. Close cooperation between client, architect, HVAC planner, system providers and contractors from an early stage on and through out the entire building process is necessary. A system of checks and balances is essential to assure quality.

The flexibility of design and small scale of decentralised ventilation systems can be beneficial if space for building services is constrained. A typical situation are low ceiling heights in retrofitting projects which do not allow the installation of technical infrastructure in suspended ceilings. In such cases FIV may be the only means to facilitate energy-efficient ventilation and state-of-the-art indoor environment. Shallow radiant floor systems which are available in Norway may be a suitable supplementary heating system.

The capability to adapt to changing use of indoor spaces is a prominent feature of FIV. Most occupancies can be covered with few standard settings for airflow rates. This applies to long-term usage of the building as well as to transient occupancy. The technology can respond to requirements for elasticity, generality and flexibility without oversizing of facilities or tedious alterations. FIV units can be manually replaced, upgraded, installed, removed or deactivated. However, planning ahead is required by installing empty casings and the supply distribution network already from the start. Such strategy is often used during construction to avoid the risk of damage during building.

Open plan as office typology has different demands than the cellular office as a higher occupancy is to be expected. Higher airflow rates are required which might cause acoustic issues and conflict with the Norwegian building code where the office typologies are not differentiated regarding permissible indoor environment parameters such as sound levels. The application of the advantages of FIV is limited as the implementation of individual control in open-plan offices is difficult. Support for individual intervention in the control of indoor environmental parameters has been stated but refers rather to the heating/cooling system and the wish to open windows than to the control of the mechanical ventilation. Simulations indicate that open-plan offices require different control strategies resulting in a higher energy demand. Further investigations of office typologies and their influence on energy efficiency might be insightful.

5. Conclusions

Aim of the work was to evaluate the applicability of façade-integrated ventilation in a Norwegian context. It can be seen that currently the reviewed FIV systems designed for Central European conditions do not comply with all requirements of the Norwegian building code. Adaptations to Nordic conditions are necessary leading to custom-made systems which are possible if a demand arises. For some aspects a good performance can be expected also in Norway while other topic require upgrades.

Energy efficiency with respect to the net-energy demand is critical in Nordic conditions. Competitiveness with centralised systems cannot be achieved in this respect at the moment. If also the energy supply and related emission factors are considered then both systems can reach similar performance. Expanding the boundary and including emissions from embodied energy then decentralised systems show a trend to be more sustainable than centralised ventilation systems.

High indoor comfort and indoor air quality can be expected as state-of-the-art concepts and mature high-performance components are utilised. Acoustic and humidity conditions are critical where the first may be solved by proper planning. The possibilities of individual control and usability will results in high user satisfaction. On the other hand, cost and maintenance can be an issues especially in Norwegian conditions. If must be decided in the individual case if feasibility is given when the expenditures are traded off against user satisfaction.

However, the concept of façade-integrated ventilation has an enormous potential as the technology has not reached its limits yet. More research and development regarding design of components and operational strategies of the technology is necessary. Also the potentials of application have to be explored further. FIV can be a competitive alternative when conventional systems fail or are not applicable, in retrofitting for example.

This work gives an insight into the potential and is addressed to planners, system providers and researchers to progress in the development of the technology to explore the prospects provided by the concept.

In this context further research and development may also focus on investigation on better solutions for local decentralised heat recovery with high efficiencies. In general, high-performance concepts for heat recovery are desirable where the reclaiming of energy is spatially separated from the actuator of the recovered heat.

The airflows need further investigations regarding the interaction with heating appliances. Especially in case of FIV the circulation of the ventilation air in the room in combination with simultaneous airflows of convector heaters needs to analysed. This seems even more important as the supply of fresh air, the supply of heated air and the return of stale air to the extract are in very close proximity at the façade.

Comprehensive investigations with respect to the entire life cycle are necessary. This applies on one hand to the assessment of the emissions over the system or buildings lifetime where the tendency of lower embodied energy must be investigated further. On the other hand, costs have to be examined further in detail to provide more conclusive statements about the feasibility.

Input parameters influence the evaluation of energy performance evaluation and the actual system design. In particular for demand controlled systems variable schedules lead to different results than static schedules as they can reflect variable presence. This can be seen very evidently for FIV with its tight fittings and possibilities of quick response. Further research on the representation of such possibilities in regulations is necessary.

State-of-the-art technology such as FIV is not envisaged by current Norwegian regulations or standards. For instance FIV may reach general requirements (delivered energy, possibly net-energy demand, emission balance) but not individual requirements such a minimum of 80 percent temperature efficiency of heat recovery. Hereby potential may be locked. Regulations may be adapted for more flexibility and generality to handle and encourage advanced technologies.

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Appendix

A. Input Parameters for Simulations

General settings

The building is an low-energy office building with ambitions for lowest possible energy demand.

Oslo as standard reference location was selected as geographical position. For the "Simien" simulation could be drawn on the standard climate condition for Oslo/Blindern. This climate data is not available for "ESP-r" where the climate file for Oslo/Fornebu is used which has been retrieved from the US DoE website. Here seasons and typical weeks have been determined in agreement with the procedure described in Hand (2011). (Figure 72)

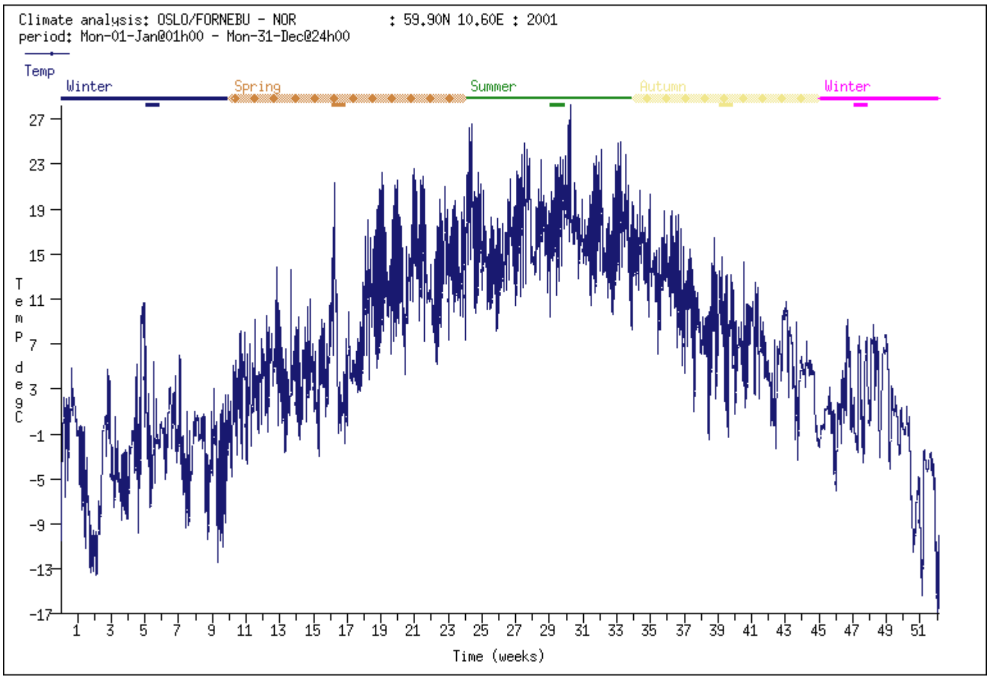

Figure 72: Climate data for Oslo Fornebu for year 1991

Usage and internal loads

Various schedules and loads are used for different simulations due to their significant impact on the performance. Initial input values of PR 42 for the energy performance simulations with "Simien" have been reviewed since the test room does not include secondary spaces like corridors, storage, etc.

"ESP-r" allows very dynamic schedules and loads. This will be taken into account for the detailed simulation focussing on transient conditions. Design values for human loads and equipment from EN 13779 will be used. The possibility to express the delay between between presence and actuation (See also further discussion of the time delay TD-OFF in Halvarsson, 2012) was discarded to reduce the complexity but also because instant response form FIV system can be expected.

Schedule & presence

The occupied period in NS 3031 is scheduled as 12 hours a day five days a week in 52 weeks per year. Default settings in "Simien" are set from 6:00 until 18:00 and are used for the simulations there. Since normal working hours are from 8:00 until 16:00 this results in two hours overhang before and after the working hours. It is assumed that the ventilation system starts two hours before the occupancy begins at 8:00 and remains active one hour after the occupancy has ended at 17:00 (Dokka, Berg and Lillelien, 2011). For an advanced representation of DCV a scenario was developed with an occupied period from 8:00 until 17:00. This schedule is assigned to the case studies with index "b". ´Unforeseen´ occupancy is assigned then in the hour between 16:00 and 17:00. This is in good agreement with with measured data for operation hours in offices by Halvarsson (2012).

The presence during the occupied period is estimated in NS 3031 by reducing the standard values by 20 %. PR 42 assumes 60 % actual presence to establish input values for lighting loads and ventilation rates. The results of Halvarsson (2011) reinforce this assumption. Based on his profile for high occupancy on page 164 a slightly modified and simplified occupancy pattern was developed for the "ESP-r" simulations. The assumption of variable occupancy equivalent to 2 minutes per hour in the evening was kept. (Figure 73)

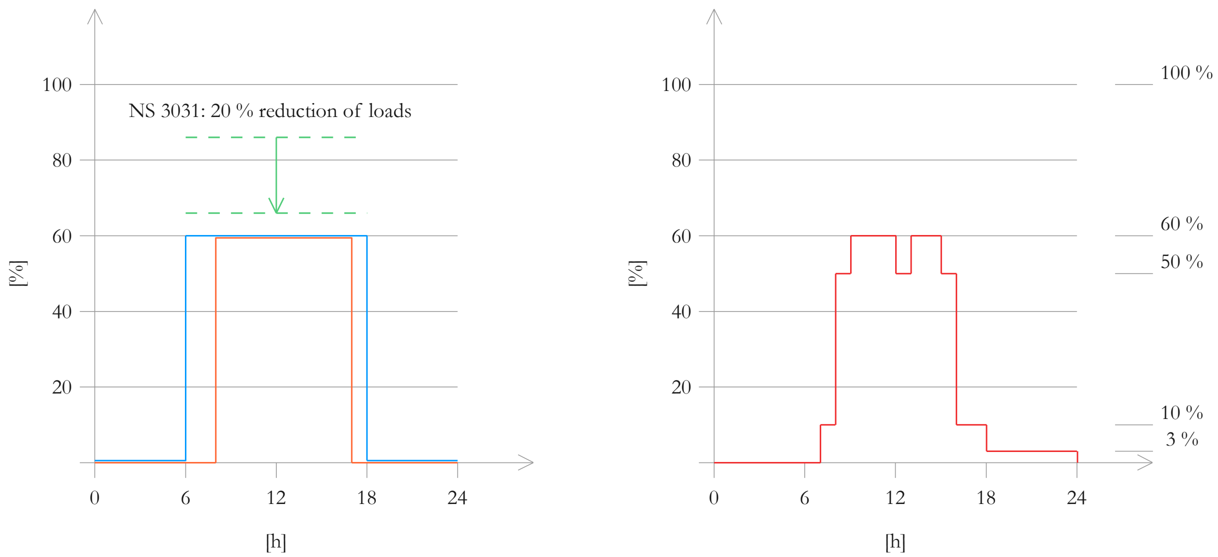

Figure 73: Schedules for simulations, "Simien" (left) and "ESP-r" (right)

Equipment

PR 42 assumes the use of equipment (a total of 100 W / person, same as used in EN 13779 for design load) for 80 % of the occupied period in the primary area while in the secondary area the load is assumed as 2 W/m² during the occupancy. Therefore the gains in the scenarios PR 42 and FIV of the energy simulations are also decreased to 80 W per person.

In the detailed simulations the loads follow the percentages of the schedule schedule assuming 100 W/person as 100 %.

Lighting

PR 42 refers to "Lyskultur" and uses 6.4 W/m² as initial value for the primary area without control and 100 % occupancy. In the case study PR42_0 this values is used adjusted for 60 % presence. For case studies with higher precision (PR42_a, PR42_b, FIV_a, FIV_b) loads for the single months are respecified using an "Ecotect"/"Radiance" daylight simulation of the test room in combination with EN 15193. In a second step the values have been altered depending on presence (60 %). The last column is the reference value from PR 42 with static values for each month. (Figure 74)

For "ESP-r" 6 W/m² are assumed in case of 100 % presence throughout the year. The hourly values depend on the dynamic schedule.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	mean	default
source based on 100 % presence	6.1	5.6	5.0	4.5	4.5	4.5	4.5	4.5	5.0	5.5	6.0	6.2	5.2	6.4
adjusted for 60 % presence	3.7	3.4	3.0	2.8	2.7	2.7	2.7	2.7	3.0	3.3	3.6	3.7	3.1	3.8

Figure 74: Monthly internal gains from lighting according to EN 15193

Persons

Both NS 3031 and PR 42 use 4 W/m² for internal gains related to persons. Halvarsson, 2011 deduces that this value already includes reduction factors. A lucid explanation, however, is not available. Hence the design values of EN 13779 are used in the case studies with higher exactness. Then 125 W/person for total heat gain and thereof 75 W/person for sensible heat gain are assigned representing a sedentary activity with 1.2 met metabolic rate.

"ESP-r" considers also latent heat gains. Therefore 50 W/person are assigned. The loads from humans are also adjusted by the schedule.

Summary

Hereinafter the overview of schedules and loads for the different scenarios of the energy simulation and the parametric study. The second case study eliminates the share of secondary space in the default settings. The third case used FIV specific values if applicable. (Figure 75)

	occupancy schedule [h]	presence [%]	lighting load [W/m²]	equipment load [W/m²]	persons load [W/m²]
TEK 10_DCV	6 – 18	–	80 % · 11.0 = 8.8	80 % · 8.0 = 6.4	4.0
PR42_0	6 – 18	60	60 % · 6.4 = 3.84	80 % · 10.0 = 8.0	4.0
PR42_a & FIV_a	6 – 18	60	60 % · monthly values	80 % · 100 W/pers.	60 % · 75 W/pers.
PR42_b, FIV_b	8 – 17	60	60 % · monthly values	80 % · 100 W/pers.	60 % · 75 W/pers.
"ESP-r"	…	…	hourly values	hourly values	hourly values

Figure 75: Summary of internal gains

Construction

Buildings elements facing outside are super-insulated. Walls and the roof feature U-values of 0.08 W/(m²K) and the U-value of the window glazing and framing is 0.8 W/(m²K). As typical in Norway the wall construction is a light-weight timber construction with low heat storage capacity. The size of windows is guided by TEK 10 § 14-3 limiting the window area to 20 % of the usable area BRA. The triple-pane glazing has a g-value of 0.45 and external shading with 80 mm Venetian blinds and automatic control triggered 200 W/m² insolation limits unwanted solar gains. The concrete slab at the ceiling is fully exposed. A raised floor with parquet flooring is assumed for the energy calculations and results in a light-weight floor construction (in the detailed simulation the floor is a separate zone). Internal walls are light-weight partitions with gypsum plasterboard finish. Thermal bridges are considered with 0.03 W/(m²K). All building materials are considered normal-emitting.

The value for airtightness n_{50} is 0.4 h^{-1} which is considered easily achievable with proper quality of the construction. For "ESP-r" the n_{50}-value must be converted to infiltration in normal pressure conditions. Therefore 0.4 h^{-1} is multiplied with the screening factor e = 0.07 in agreement with NS 3031. The resulting value for n_4 is approximately 0.03 h^{-1} and applied to the zone of the raised floor only.

Space geometry

The façade of room is 4.80 m wide representing four common Norwegian façade raster of 1.20 m (Byggforsk, 2004). To investigate different usage scenarios the exemplar room spans over two single office room modules with 2.40 width. The partition walls and the connecting central wall depends on the usage. The height of the room is 3.00 m.

The determination of the room depth and the eventual total space volume is a result of considerations related to the FIV system to find an optimum starting point for a parametric study. The room could be designed according to heating, cooling or ventilation capacity of the FIV units. However, heating and cooling are not crucial parameters because firstly, the units are powerful enough to cover the loads and secondly, compensation with auxiliary systems is possible. In contrast, designing a system according to indoor air quality dictates high benchmarks when using TEK 10 and EN 15251.

Most commercial systems have three-stage pre-sets. Common settings are 60, 90, 120 m³/h outdoor air supply. A first literature review in DeAL (2008) shows that airflow rates higher than

90 m³/h may cause acoustic issues. Therefore this value is considered as design airflow rate during occupancy. Up to 120 m³/h airflow rate may be used outside operation hours e.g. for night cooling or higher occupancies.

The dimensioning airflow rates follow TEK 10 § 13-3 – during occupancy 26 m³/h per person and 3.6 m³/h per square meter floor area for materials and outside occupancy 0.7 m³/h per square meter floor area. The size of the room was chosen to find reasonable matches between the required airflow rates and the air volumes of the FIV units. In the base case which will be used in the energy calculations two units with each 90 m³/h and a total of 180 m³/h supply air to an office with an occupancy of three persons representing a demanding setting with 8.64 m²/person.

In summary, the dimensions of the exemplar room are 4.80 x 5.40 x 3.00 metres, the floor area is 25.92 m², and the heated air volume 77.76 m³. Windows are treated differently in "Simien" and "ESP-r". In "Simien" each window measures 2.20 x 1.20 m. In "ESP-r" a window measures 2.40 x 1.20 m whereof 10 cm frame all around result in a glazing area of 2.20 x 1.00 m. The exterior surface/volume ratio A/V equals 0.52 m^{-1}, which makes the room an ambitious environment for studies.

System settings

Type of FIV unit

A generic approach is chosen. Therefore no unit-specific parameters are used. However local conditions are addressed by assuming a system design that would be optimised to be employed all year round in Norwegian climate. A pre-heater for frost protection of the plate heat recuperator would be integrated, for instance.

Also the location of the unit is unspecified for the energy performance simulations. In case of the detailed simulations with "ESP-r" the unit is mounted on the exterior wall for modelling purposes. However, in an airflow network of this scale the location has only a minor impact for the investigated aspects.

The centralised ventilation system of the reference case uses standard input (supply water temperatures etc.) from "Simien".

Heat recovery

A local heat recovery inside the FIV unit is foreseen. The efficiency of the plate heat exchanger used in FIV units is approximately 55 % according to manufacturer information. To avoid icing of the heat exchanger a frost protection temperature of 0 °C is considered in accordance with NS 3031 annex H.4 for office buildings. This will be automatically taken into account in "Simien".

The reference cases with centralised ventilation have rotary heat recovery according to the requirements in TEK 10. Hence the temperature efficiency is 80 % and there is no need for frost protection.

Environmental control strategies

System type

Displacement ventilation is assumed. Hence the common practice of constant supply air temperature is utilised which requires additional measures and systems to cover the heating and cooling loads. For simplicity reasons a uniform supply air temperature throughout the year of 19 °C is assumed.

Heating / cooling system

The set-point temperatures for space heating follow NS 3031. In "Simien" energy simulations a merely convective and a radiant surface heating system are postulated. The latter as the less efficient of both is used with a convective portion of 80 % (Programbyggerne, 2011) and 38/32 °C supply and return temperature. An auxiliary space cooling system is not necessary.

Heating in the detailed simulations is limited to the main occupied period from 8:00 to 16:00 despite the schedule includes occupancy also before and after this period.

Ventilation

In "Simien" simulations the ventilation system in all cases is assumed as demand controlled VAV. NS 3031 suggests a lump reduction of the average airflow rate from 8.0 $m^3/(h \cdot m^2)$ by 20 % to 5.6 $m^3/(h \cdot m^2)$ to simulate VAV. Also PR 42 proposes the use of a mean airflow which is derived from assumptions regarding occupancy, presence and the relation of primary and secondary area. Eventually, the resulting airflow rate 5.41 $m^3/(h \cdot m^2)$ is only slightly different from the NS 3031 value for DCV.

Another more dynamic approach for simulations operates within a range of a maximum and minimum airflow rate during the occupied period. In “Simien” this approach attempts to keep the CO_2 level below 800 ppm as required in TEK 10 § 13-1 2b. The discharge of CO_2 is coupled to occupancy and linked to the internal gains from persons (Programbyggerne, 2012). The exact linkage, however, could not be determined. There is a risk of distortion of results since the input parameters in the simulations deviate from the standard input of 4 W/m^2 for human loads. Hence the robust approach of mean airflow rates is used in the energy performance calculations for all case studies although leading to higher net-energy demands. (See net-energy demands of the basically identical case studies FIV_a and FIV_mat,1: 80.2 kWh/m^2 versus 73.0 kWh/m^2)

Case study TEK10_DCV operates with 20 % reduced standard mean airflow rate. Hence 5.6 $m^3/(h \cdot m^2)$ during and 2.0 $m^3/(h \cdot m^2)$ outside occupancy are used. In case of PR42_0 the calculation steps in PR 42 were reconstructed, however only for primary area. PR42_a, PR42_b and FIV operate with the same specific maximum airflow rate of 6.94 $m^2/(h \cdot m^2)$ (equalling 180 m^3/h) and with system-specific settings for the minimum airflow rate. Outside the occupied period the minimum airflow rates are used. An overview of the used airflow rates is presented in figure 76.

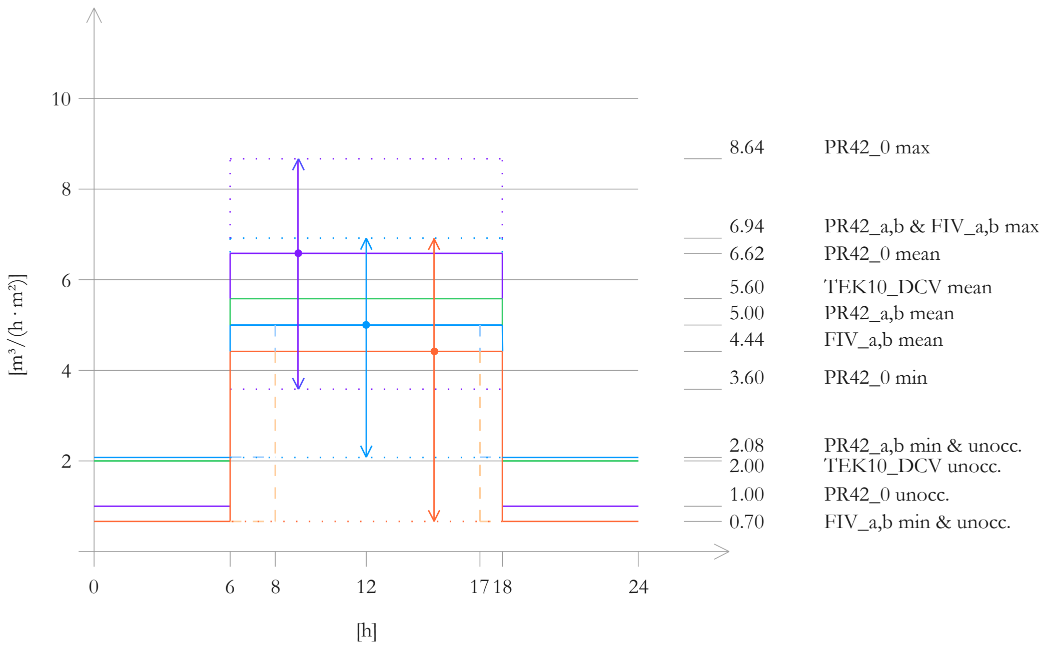

Figure 76: Airflow rates in "Simien" simulations

For PR42_a and PR42_b it is assumed that the minimum airflow rate cannot be less than 30 % of the maximum as it is typical for centralised ventilation systems (Grini and Wigenstad, 2011). Data sheets of FIV units indicate that 60 $m^3/h \pm 10$ % is the minimum airflow rate of most units (Iczek, 2012; LTG FDVplus, 2008; Trox, 2009). However, it is assumed that FIV units don not have to run continuously. Hereby the required airflow rate outside occupancy of 0.7 $m^3/(h \cdot m^2)$ in accordance with TEK 10 can be achieved in contrast to centralised systems running compliant with regulations.

During summer nights ´free´ cooling is used to cool down the building. Minimum airflow rates of 2.08 $m^3/(h \cdot m^2)$ equalling 54 m^3/h are utilised in the case studies PR42_a and PR42_b while 60 m^3/h are used in cases FIV_a and FIV_b. The night purge is activated according to the standard input of "Simien".

Figure 77 shows an numerical overview of all case studies of the energy performance simulations.

	schedule		airflow rate				
	schedule [h]	presence [%]	maximum [$m^3/(h \cdot m^2)$]	minimum [$m^3/(h \cdot m^2)$]	mean [$m^3/(h \cdot m^2)$]	outside operation [$m^3/(h \cdot m^2)$]	night cooling [$m^3/(h \cdot m^2)$]
TEK10_DCV	6 – 18	–	–	–	5.6	2	2
PR 42_0	6 – 18	60	8.64	3.6	6.62	1	1
PR 42_a	6 – 18	60	6.94	2.08	5.00	2.08	2.08
PR 42_b	8 – 17	60	6.94	2.08	5.00	2.08	2.08
FIV_a	6 – 18	60	6.94	0.7	4.44	0.7	2.31
FIV_b	8 – 17	60	6.94	0.7	4.44	0.7	2.31
"ESP-r"	…*	…*	6.94	0.7	–	0.7	2.31

Figure 77: Summary of airflow rates in simulations

The specific fan power SFP of the FIV unit is set to 0.6 $kW \cdot s/m^3$ which is considered achievable (see Appendix B). Plugge (2012b) mentions even an SFP of 0.3 $kW \cdot s/m^3$. However, it is not clear if it applies for the the single supply and extract fans or for the total of both. The water-borne heat exchanger of the FIV unit operates with $\Delta\theta = 70 - 50 = 20$ K in case of heating and with $\Delta\theta = 18 - 12 = 6$ K in case of cooling. The same settings also applies to the water-borne heating and cooling coil of the centralised ventilation system where the SFP is 1.5 $kW \cdot s/m^3$.

In “ESP-r” simulations the ventilation rates follow the schedule relative to the maximum airflow rate (figure 78).

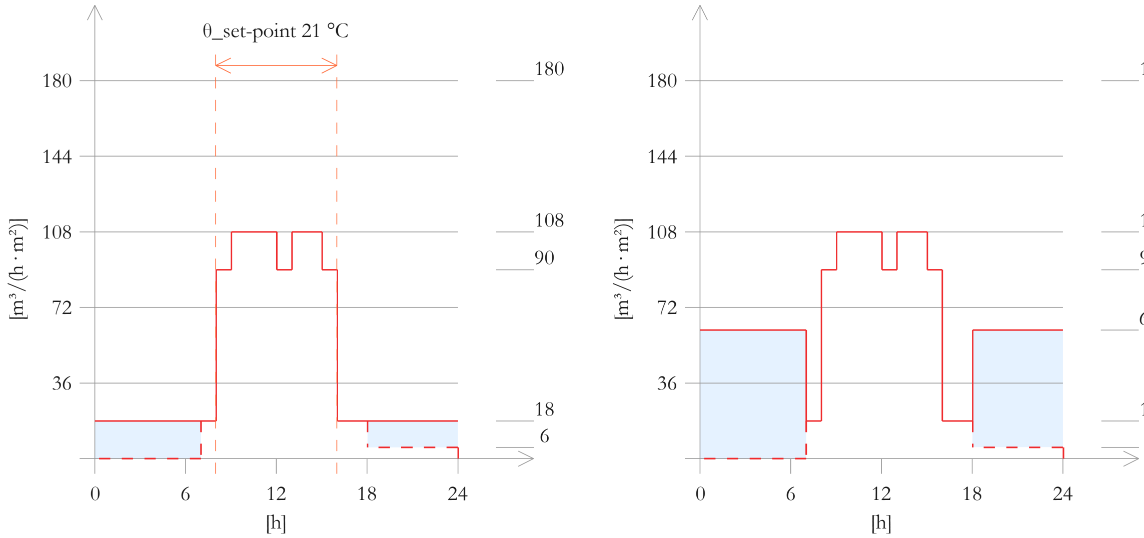

Figure 78: Heating and cooling scenario in "ESP-r" base case

For the base case the ventilation rate outside occupancy is in accordance with TEK 10 § 13-3. Thus the airflow rate is 0.7 $m^3/(h \cdot m^2)$. ´Free´ cooling with outdoor air in summer nights is also utilised in case of the detailed simulations. The airflow rate is increased to 60 m^3/h then.

Energy supply

For the calculation of the delivered energy and the emissions related to operation a simple energy concept was chosen. A heat pump covers 100 % space heating, 100 % DHW and 90 % of the ventilation heating. The other 10 % are covered by electricity for peak loads in case of centralised and for the frost protection in case of FIV. The rest is direct electricity from the grid.

B. Specifications of FIV Units

The informations are collected from data sheets which are available at the websites of the manufacturers. The accuracy of statements cannot be guaranteed.

		EMCO				**KAMPMANN**						**KRANTZ**			**LTG**					
		emcovent UZA			emcovent FLH	Kavent BA					Kavent FW	LG-ZA-M-SB			FVD		FVDplus	FVM		
general data																				
object																				
year		2011			2010	2009					2009	2008			2009		2009	2009		
building location		–			–	–					–	–			–		–	–		
unit location		raised floor			window sill	raised floor					window sill	sill			raised floor		raised floor	wall, vertical		
					or lintel															
supply air		yes			yes	yes					yes	yes			yes		yes	yes		
exhaust air		yes			yes	no					yes	yes			yes		yes	yes		
mixing ODA + SEC		no			no	yes					no	yes			no		yes	no		
heat recovery		plate			plate	no					plate	plate			no		no	rotary		
efficiency		< 60 %			?						?	ca. 62 %						max. 62 %		
bypass		supply			?						supply	supply						?		
secondary air only		no			no	yes					no	yes			no		y ?	no		
static heating		?			no	no					no	no			no		no	no		
dimensions												space requirement		unit						
length	mm	1250			1200	1150					900	1200		1000	1250		1250	450		
height	mm	230			190	230					550 / 370	900		650	220		220	170		
width	mm	600			375	650					490 / 160	300		300	610		610	2700		
weight	kg											60						75		
ventilation capacity																				
ODA airflow rates	m^3/h	60 – 120			0 – 180	50 – 180 (60, 90, 120)						30 % of SUP			60,90,120 (– 180)		60,90,120 (–180)	0 – 140		
SEC airflow rates	m^3/h	–			–	80 – 140						–			–		196 – 506	–		
ODA filter		F 7			F 7	F 7					F 7	F 5 (F 7)			F 7		F 7	F 7		
ETA filter		G 3			G 3	G 1					F 4	F 5 (F 7)			G 3		G 3	F 5		
													(assuming 30 % ODA)							
ODA airflow rate	m^3/h	60	90	120		60	90	120	60	120		33	45	57	120		120	120	120	60
SEC airflow rate		–	–	–		–	–	–	140	80		77	105	133	–		288	stufe II	stufe II	(20)
												110	*150*	*190 (max)*						
heating capacity		w/ HR	w/ HR	w/ HR											"nenn."	"recomm."	"nenn."	"nenn."	"recomm."	62 % HR
outdoor air temperature	°C	-12	-12	-12		-12	-12	-12	-12	-12			-15		-12	-12	-12	-12	-12	-12
room air temperature	°C	20	20	20		22	22	22	22	22			22		22	22	22	22	22	22
total heating power	W	1020	1467	1884		1000	1280	1530	2460	2980			1200		4620	2090	2744	1860	1405	1355
power heat recovery	W	356	481	596																
outdoor air heating power (T_SUP – T_ODA)	W	288	484	691		?	?	?	?	?			280		1360	1360	944	740	790	345
internal heating power (T_SUP – T_Raum)	W	376	502	597		?	?	?	?	?			920		3260	730	1800	1120	615	1010
intake air temperature	°C	5.7	3.9	2.8		-12	-12	-12	11.8	1.6										
intake air relative humidity	%												90							
supply air temperature	°C	38.7	36.6	34.8		59.6	56.7	54.6	47.3	43.1			36							
supply air relative humidity	%																			
supply water flux	kg/h	57	85	110		88	112	134	215	261			90		390	100	100	160	120	
supply water temperature	°C	55	55	55		75	75	75	75	75			50		75	50	75	60	50	75
return water temperature	°C	45	45	45		65	65	65	65	65			40		65	32	51.5	50	40	
return water pressure loss	kPa	1.4	2.9	4.8		1.4	2.9	4.8	5.5	7.8			40..50 mbarmax 10 bar		94	6	3.4	9	5	
cooling capacity		w/ HR	w/ HR	w/ HR											"nenn."			"nenn."	"recomm."	62 % HR
outdoor air temperature	°C	32	32	32		32	32	32	32	32			32		32		32	32	32	32
room air temperature	°C	26	26	26		26	26	26	26	26			26 (50% RH)		26		26	26	26	26
total cooling power	W	417	593	752		280	411	532	587	694			720		630		770	580	480	235
power heat recovery	W	67	90	112																
outdoor air cooling power (T_SUP – T_ODA)	W	129	189	242		0	0	0	0	0			200		240		240	90	100	55
internal cooling power (T_SUP – T_Raum)	W	221	314	398		280	411	532	587	694			520		390		530	490	380	180
intake air temperature	°C	28.7	29	29.2		32	32	32	27.8	29.6										
intake air relative humidity	%	40	40	40		40	40	40	40	40			40							
supply air temperature	°C	15	15.6	16.1		17.6	17.9	18.3	18.9	19			18							
supply air relative humidity	%					66	65	64	69	76										
supply water flux	kg/h	60	86	110		93	124	153	253	298			200		265		200	250	140	
supply water temperature	°C	10	10	10		17	17	17	17	17			14		17		17	16	17	17
return water temperature	°C	15	15	15		19	19	19	19	19			17		19		20.3	18	20	
return water pressure loss	kPa	1.7	3.3	5.2									200..250 mbar		28		6.2	34	12	
acoustic performance																				
sound power level L_W [schalleistungspegel]	dB(A)	30	38	45	30 – 53	30	34	36							32,35,40		ODA: 32,35,40	43		37
sound pressure level L_P [schalldruckpegel]	dB(A)												38				SEC: 29,33,40,45,51	39 ohne WRG		
energy performance																				
fan electric potential	V																			
electric current	A																			
complex (apparent) power [scheinleistung]	VA												100							
real (active) power [leistungsaufnahme]	W					10	14	18					max. 65		5,7,9		ODA: 5,7,9	27		14
																	SEC: 16,17.5,19.5,22,27			

												TROX									
												FSL-B-190-**ZAB**				FSL-B-ZAB		FSL-U-ZUM			FSL-U-ZAB
general data																					
object																"IBC"		"Posttower"			
year												2003				2003		2003			2009
building location												– (e.g. Neckersulm)				Frankfurt / M.		– (e.g. Bonn)			–
unit location												window sill				window sill		raised floor			raised floor
supply air												yes				yes		yes			yes
exhaust air												yes				yes		no			yes
mixing ODA + SEC												no				no		yes			?
heat recovery												plate				plate		no			plate
efficiency												?				?					?
bypass												supply				exhaust					?
secondary air only												no				no		yes			no
static heating												yes				yes		no			yes
dimensions																					
length	mm											1200				800 ?		1200			1200
height	mm											600 / 190 (below window)				480 / ?		180			200
width	mm											450 / 230 (in front of sill)				510 / ?		550			500
weight	kg																				
ventilation capacity																					
ODA airflow rates	m³/h											60 – 120				90 – 120		50, 75, 90, 120			60 – 120
SEC airflow rates	m³/h											–				–		0, 55, 110			–
ODA filter												F 6				G 3		F 6			
ETA filter												G 3				G 4		G 3			
																			(examples)		
ODA airflow rate	m³/h	90	120	–	–	60	60	90	90	120	120	60	80	90	120	90	120	90	–	80	
SEC airflow rate		(25)	(30)	75	100	80	100	90	105	110	130	–	–	–	–	–	–	–	120	120	
heating capacity		58 % HR	54 % HR			62 % HR	62 % HR	58 % HR	58 % HR	54 % HR	54 % HR										
outdoor air temperature	°C	-12	-12	-12	-12	-12	-12	-12	-12	-12	-12										
room air temperature	°C	22	22	22	22	22	22	22	22	22	22	20	20	20	20			20	20	20	
total heating power	W	1670	1940	850	1100	1690	1840	1930	2040	2145	2220	962	1257	1349	1735	800	1010	1459	805	1987	400 – 800
power heat recovery	W																				
outdoor air heating power (T_{SUP} – T_{ODA})	W	510	695	0	0	345	340	515	510	695	695	762	991	1049	1335			1459	0	?	
internal heating power (T_{SUP} – T_{Raum})	W	1160	1245	850	1100	1345	1500	1415	1530	1450	1525	200	266	300	400			0	805	?	
intake air temperature	°C											-12	-12	-12	-12	4	4	-12	20	7.2	
intake air relative humidity	%																				
supply air temperature	°C											30	30	30	30	30.3	29	36.3	40	36.9	
supply air relative humidity	%																				
supply water flux	kg/h											60	120	120	360	36	54	60	30	138	
supply water temperature	°C	75	75	75	75	75	75	75	75	75	75	55	55	55	55	70	70	75	75	75	
return water temperature	°C											41	46	45	51	50.5	53.6	54	51	62.3	
return water pressure loss	kPa											1.2	3.9	3.9	28.7	0.3	0.6	0.9	0.3	3.9	
cooling capacity		58 % HR	54 % HR			62 % HR	62 % HR	58 % HR	58 % HR	54 % HR	54 % HR										
outdoor air temperature	°C	32	32	32	32	32	32	32	32	32	32										
room air temperature	°C	26	26	26	26	26	26	26	26	26	26	26	26	26	26			26	26	26	
total cooling power	W	290	375	200	250	330	350	370	400	430	465	286	384	431	559	560	690	432	335	715	280 – 560
power heat recovery	W																				
outdoor air cooling power (T_{SUP} – T_{ODA})	W	85	95	0	0	55	55	80	80	95	95	126	174	191	239			432	0	?	
internal cooling power (T_{SUP} – T_{Raum})	W	205	280	200	250	275	295	290	320	335	370	160	210	240	320			0	335	?	
intake air temperature	°C											32	32	32	32	32	32	32	26	28.4	
intake air relative humidity	%											40	40	40	40			40	40	40	
supply air temperature	°C											18	18	18	18	18.3	18.7	18	18	18	
supply air relative humidity	%																				
supply water flux	kg/h											48	96	120	156	100	100	48	48	144	
supply water temperature	°C	17	17	17	17	17	17	17	17	17	17	16	16	16	16	12	12	16	16	16	
return water temperature	°C											19	19	19	19	17	18	23.9	22.1	20.3	
return water pressure loss	kPa											0.3	0.9	1.2	2.1	9.4	9.4	0.3	0.3	3	
acoustic performance																					
sound power level L_W [schalleistungspegel]	dB(A)	38	41	32	36	38	40	39	41	41	43	39	42	44.5	49.5	41	45.5	35	43	48	39 – 49
sound pressure level L_P [schalldruckpegel]	dB(A)											31	34	36.5	41.5	33	37.5	27	35	40	
energy performance																					
fan electric potential	V											76	90	100	130	94	120				
electric current	A											0.3	0.34	0.37	0.44	0.3	0.34				
complex (apparent) power [scheinleistung]	VA																				
real (active) power [leistungsaufnahme]	W	18	27	14	17	23	23	27	28	37	37							5	11	19	

		FSL-U-ZUS	FSL-B-ZAU						FSL-B-ZAU	FSL-B-ZUS			FSL-B-ZAB			FSL-B-ZAB			SCHOOLAIR-B		
general data																					
object			"Lighttower"						"Capricorn"	"Feldbergstrße"			"Laimer Würfel"			"Traungasse"					
year		2009	2004-5						2005	2006-8			2007			2009			2010		
building location		–	Frankfurt / M.						Düsseldorf	Frankfurt / M.			München			Vienna			–		
unit location		raised floor	sill						wall	vertical			window sill			sill			sill		
supply air		yes	yes						yes	yes			yes			yes			yes		
exhaust air		no	yes						yes	no			yes			yes			yes		
mixing ODA + SEC		y ?	yes						yes – 3 stages	yes			no			no			no		
heat recovery		no	plate						plate	no			plate			plate			plate		
efficiency			?						?				?			?			~ 55%		
bypass			exhaust						exhaust				supply			supply			supply		
secondary air only		y ?	yes						yes – 3 stages	yes			no			no			no		
static heating		no	no						y (frost protection)	no			yes			no			no		
dimensions																					
length	mm	1200	980						1050	350			950			1100			1600		
height	mm	180 – 230	400						1050	1900			590 / 400			630			640		
width	mm	550 – 640	640						200	300			490 / 180			320			420		
weight	kg																		70		
ventilation capacity																					
ODA airflow rates	m^3/h	80	50						60, 90, 120	75			90 – 180			80 – 120			150, 200, 250, 325		
SEC airflow rates	m^3/h	120	50 – 150						30, 60	0 – 135			–			–			–		
ODA filter			F 6						F 6	F 7			F 7			F 7			F 7		
ETA filter			G 3						G 3	–			G 3		for night	G 3			G 3		
ODA airflow rate	m^3/h		50	50	50	–	–	–		75	75	75	90	120	180	80	100	120	150	200	250
SEC airflow rate			–	50	100	50	100	150		–	75	135	–	–	–	–	–	–	–	–	–
heating capacity									w/o HR				open bypass			w/o HR			w/o HR	w/o HR	w/o HR
outdoor air temperature	°C												-16	-16	–	-12	-12	-12	-12	-12	-12
room air temperature	°C		22	22	22	22	22	22		22	22	22	21	21	–	22	22	22	22	22	22
total heating power	W	530 – 1330	827	956	1107	261	489	650	800	1346	1657	2153	1800	2300	–	1397	1604	2300	2490	3349	4095
power heat recovery	W																				
outdoor air heating power (T_{SUP} – T_{ODA})	W		577	573	607	0	0	0		809	817	823	1270	1640	–	916	1143	1888	1618	2160	2692
internal heating power (T_{SUP} – T_{Raum})	W		250	383	500	261	489	650		537	840	1330	530	660	–	481	461	412	872	1189	1403
intake air temperature	°C		-12	5	10.5	22	22	22		-10	6	10.6			–						
intake air relative humidity	%														–						
supply air temperature	°C		37	33.5	32	37.6	36.6	35	30	43.5	38.8	41	38.5	37.5	–	40	35.8	32.4	37.4	37.8	36.8
supply air relative humidity	%														–						
supply water flux	kg/h									42	72	132	60	120	–	42	48	54	84	120	150
supply water temperature	°C		70	70	70	70	70	70	35	70	70	70	70	70	–	80	80	80	60	60	60
return water temperature	°C									42	50	56			–	51	50	51	33.8	35.4	35.9
return water pressure loss	kPa														–				< 3	< 3	< 3
cooling capacity									w/o HR				w/ HR			w/o HR					
outdoor air temperature	°C												32	32	–	32	32	32			
room air temperature	°C		26	26	26	26	26	26		26	26	26	26	26	–	26	26	26			
total cooling power	W	370 – 930	479	649	786	243	446	662	460	367	632	836	500	600	–	542	664	781			
power heat recovery	W																				
outdoor air cooling power (T_{SUP} – T_{ODA})	W		316	312	286	70	110	162		117	132	136	240	300	–	264	327	356			
internal cooling power (T_{SUP} – T_{Raum})	W		163	337	500	173	336	500		250	500	700	260	300	–	278	337	425			
intake air temperature	°C		32	29	28	26	26	26		32	30	29.4			–						
intake air relative humidity	%														–						
supply air temperature	°C		16.2	15.9	16	15.6	15.9	16	18	18	18	18	17.2	18.4	–	15.6	15.9	15.4			
supply air relative humidity	%														–						
supply water flux	kg/h									39	93	192	84	84	–	78	96	114			
supply water temperature	°C		10	10	10	10	10	10	16	14	14	14	12	12	–	7	7	7			
return water temperature	°C									22	20	18			–	13	13	13			
return water pressure loss	kPa														–						
acoustic performance																					
sound power level L_W [schalleistungspegel]	dB(A)	35 – 40	35	39	42	29	34	39	31 – 46	32.4	37.2	39.8	36	42	51	31	35	39	30	34	39
sound pressure level L_P [schalldruckpegel]	dB(A)		27	31	34	21	26	31	23 – 38	24.4	29.2	31.8	28	34	43	23	27	31	22	26	31
energy performance																					
fan electric potential	V																				
electric current	A																				
complex (apparent) power [scheinleistung]	VA																				
real (active) power [leistungsaufnahme]	W		27	31	36	16	19	26	9 – 20	25	41	57	15	21	61	15	17	19		17	

			SCHOOLAIR-V			
general data						
object						
year			2010			
building location			–			
unit location			sill			
supply air			yes			
exhaust air			yes			
mixing ODA + SEC			no			
heat recovery			plate			
efficiency			~ 55%			
bypass			supply			
secondary air only			no			
static heating			no			
dimensions						
length	mm		440			
height	mm		2400			
width	mm		370			
weight	kg		80			
ventilation capacity						
ODA airflow rates	m³/h		150, 200, 250, 325			
SEC airflow rates	m³/h		–			
ODA filter			F 7			
ETA filter			G 3			
ODA airflow rate	m³/h	325	150	200	250	325
SEC airflow rate		–	–	–	–	–
heating capacity		w/o HR	w/o HR	w/o HR	w/o HR	w/o HR
outdoor air temperature	°C	-12	-12	-12	-12	-12
room air temperature	°C	22	22	22	22	22
total heating power	W		2760	3710	4560	
power heat recovery	W					
outdoor air heating power (T_SUP – T_ODA)	W		2024	2701	3366	
internal heating power (T_SUP – T_Raum)	W		736	1009	1194	
intake air temperature	°C					
intake air relative humidity	%					
supply air temperature	°C	40 ?	36.7	37.1	36.3	40 ?
supply air relative humidity	%					
supply water flux	kg/h		70	100	130	
supply water temperature	°C		60	60	60	
return water temperature	°C		25.6	27.6	29.3	
return water pressure loss	kPa		< 8	< 8	< 8	
cooling capacity						
outdoor air temperature	°C					
room air temperature	°C					
total cooling power	W					
power heat recovery	W					
outdoor air cooling power (T_SUP – T_ODA)	W					
internal cooling power (T_SUP – T_Raum)	W					
intake air temperature	°C					
intake air relative humidity	%					
supply air temperature	°C					
supply air relative humidity	%					
supply water flux	kg/h					
supply water temperature	°C					
return water temperature	°C					
return water pressure loss	kPa					
acoustic performance						
sound power level L_W [schalleistungspegel]	dB(A)	44	35	39	42	48
sound pressure level L_P [schalldruckpegel]	dB(A)	36	27	31	34	40
energy performance						
fan electric potential	V					
electric current	A					
complex (apparent) power [scheinleistung]	VA					
real (active) power [leistungsaufnahme]	W	45		17		45