Energy and Indoor Environment in New Buildings with Low-Temperature Heating System

Licentiate Thesis

Arefeh Hesaraki

Stockholm-Sweden

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Royal Institute of Technology KTH
School of Architecture and the Built Environment
Division of Fluid and Climate Technology

“Increasing energy efficiency is the largest, least expensive, most benign, most quickly deployable, least visible, least understood and most neglected way to meet future energy demand”

Amory Lovins, Energy expert, head of the Rocky Mountain Institute, USA

Abstract

The aim of this thesis was to evaluate new buildings with low-temperature heating systems in terms of energy consumption and thermal comfort, and to pay some attention to energy savings and indoor air quality. To reach this aim, on-site measurements as well as building energy simulations using IDA Indoor Climate and Energy (ICE) 4 were performed. Results show that the investigated buildings with low-temperature heating system could meet the energy requirements of Swedish regulations in BBR (Boverkets byggregler), as well as provide a good level of thermal comfort. By implementing variable air volume ventilation instead of constant flow, i.e. decreasing the ventilation air from 0.35 to 0.10 $l \cdot s^{-1} \cdot m^{-2}$ during 10 hours of unoccupancy, gave up to 23 % energy savings for heating the ventilation air. However, the indoor air quality was not acceptable because VOC (volatile organic compound) concentration was slightly above the acceptable range for one hour after occupants arrive home. So, in order to create acceptable indoor air quality a return back to the normal ventilation requirements was suggested to take place two hours before the home was occupied. This gave 20 % savings for ventilation heating. The results of this study are in line with the European Union 20-20-20 goal to increase the efficiency of buildings by 20 % to the year 2020.

Sammanfattning (licentiatavhandling Arefeh Hesaraki, Strömnings- och klimatteknik, KTH)

Syftet med denna avhandling var att utvärdera nya byggnader med lågtempererade värmesystem utifrån energiförbrukning och termisk komfort samt att diskutera energibesparingar och inomhusluftens kvalitet. För att få fram resultaten har på plats mätningar i nya fastigheter samt energisimuleringar med IDA Indoor Climate and Energy (ICE) 4 utförts. Resultaten visar att undersökta byggnader med låg framledningstemperatur till rumsvärmare klarar svenska energikrav enligt BBR (Boverkets byggregler) och ger god termisk komfort. Genom att införa variabelt ventilationsflöde istället för konstant flöde, dvs minska ventilationsluften från 0,35 l/s till 0,10 l/s per kvadratmeter under 10 timmar, erhölls upp till 23 % energibesparing för uppvärmning av ventilationsluft . Däremot var inomhusluftens kvalitet då inte helt godtagbar då koncentrationen av VOC (flyktiga organiska ämnen) var något över acceptabel nivå under cirka en timme. För att skapa acceptabel luftkvalitet inomhus under dygnets alla timmar minskades därför lågflödesperioden (0,10 l/s per kvadratmeter) till 8 timmar och lades till den period de boende i huvudsak befann sig utanför bostaden (kontorstid). Detta gav 20 % i energibesparing för uppvärmning av ventilationsluft. Resultaten från denna studie ligger i linje med Europeiska unionens 20 20-20 mål att öka energieffektiviteten i byggnader med 20 % till år 2020.

Acknowledgment

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Contents

The work presented in this licentiate thesis started in September 2011 under supervision of Professor Sture Holmberg.

List of papers

This licentiate thesis includes the following papers, which are referred to by their Roman numerals in the text.

Paper I	Energy performance of low temperature heating systems in five new-built Swedish dwellings: a case study using simulations and on-site measurements, Journal of Building and Environment 64, 85-91, June 2013
Paper II	Demand controlled ventilation in new residential buildings: consequences on indoor air quality and energy savings, Submitted to Journal of Indoor and Built Environment, May 2013
Paper III	Energy Performance Evaluation of New Residential Buildings with a Low-Temperature Heating System: Results from Site Measurements and Building Energy Simulations, In: Proceedings of The Second International Conference on Building Energy and Environment, COBEE 12, Colorado, USA, August 2012.
Paper IV	Demand Controlled Ventilation in a Combined Ventilation and Radiator System, To be published in: Proceedings of International Conference CLIMA 13. The 11th REHVA World Congress and 8th International Conference on IAQVEC, Prague, Czech Republic, June 2013.
Paper V	An investigation of energy efficient and sustainable heating systems for buildings: Combining photovoltaic with heat pump, In: Proceedings of The Fourth International Conference on Sustainability in Energy and Buildings, SEB 12, Stockholm, Sweden, September 2012.

1. Introduction

The building sector accounts for approximately one third of the worldwide total energy consumption [1]. A tendency toward rising energy demand in the housing sector is due to population growth, higher comfort requirements, and increasing indoor activities. However, in this sector there are good opportunities for reducing the energy demand through improvements in design, operation and renovation technology.

In the building sector about 10 – 15 % of the total consumption including hot water, heating and electricity is used during the short-term phase, i.e. during construction and later during demolition, and the rest, 85 - 90 %, is used during the long-term phase when the building is being used [2]. So, to make energy-efficient buildings there should be special considerations to the long-term phase.

The energy consumption in the building is affected by three factors, i.e. technology, people, and outdoor climate.

Technology: refers to the technical properties of the building, e.g. size, shape, heat transfer coefficient (U value) of materials, heat capacity and airtightness of the building envelope.
People: the role of people shows the different living habits such as domestic hot water (DHW) usage, or airing by opening windows and etc.
Outdoor climate: refers to the outdoor temperature, solar radiation intensity and wind velocity depending on location of the building.

These three factors determine the total amount of energy usage of a building. Currently, the first factor, i.e. technology, receives the main attention in the construction sector. Minimizing transmission and ventilation heat losses, e.g. by using efficient insulation in the building envelope with high air tightness, or by using energy saving equipment such as a heat recovery system, variable air volume ventilation system, or a heat pump all result in improving the technical properties of the building. However, compared to the technology there has not been much consideration given to the role of people and their living habits in energy savings.

By studying the diagram of energy consumption in the households of northern Europe, shown in Fig. 1, it is obvious that the heating demand for space and ventilation is responsible for the main part of the total consumption, i.e. 73 % [3]. Therefore, an energy-efficient heating and ventilation system such as the combination of a heat pump and a low-temperature heating system (supply temperature below 45 °C), or a demand-controlled ventilation system (DCV) has high potential for energy savings. The lower the supply water temperature to the radiator, the higher the coefficient of performance (COP) the heat pump has. The thermal efficiency of a heat pump improves by 1-2 % for every degree of reduction in supply water temperature [4].

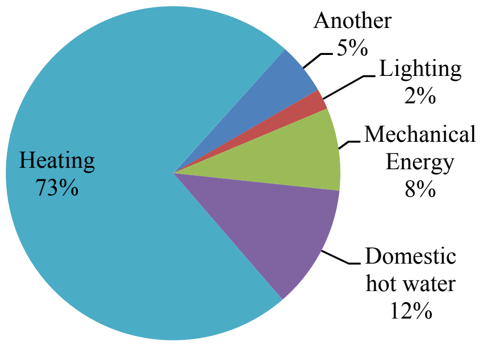

Fig. 1. Structure of energy consumption in north EU households [3]

In 2007, the European Union set up an energy efficiency policy called "20-20-20". The target of this policy was to reduce energy consumption in buildings by 20 %, and to make buildings more sustainable by a 20 % share of renewable energies, and also a 20 % reduction in greenhouse gas emissions by the year 2020.

To meet the first criterion of this policy, i.e. reducing the energy demand in buildings by 20 %, one suggestion could be implementing efficient building services in both new buildings and in retrofitting of existing buildings. New-built buildings are usually airtight with low mean U value compared to the old buildings, giving reduced transmission loss and increasing the share of ventilation loss. So, an energy-efficient ventilation system such as heat recovery by heat exchanger, or using a DCV system has high potential in reducing the total heating demand. The Swedish building regulations, BBR [5], states that an occupied residential building must have a minimum ventilation rate (per floor area) of 0.35 $l \cdot s^{-1} \cdot m^{-2}$, and if the house is unoccupied, this value is allowed to drop to 0.10 $l \cdot s^{-1} \cdot m^{-2}$. However, in many residential buildings where people leave the house in the morning and return in the afternoon, a constant air volume (CAV) ventilation system is dominant. This results in wasting energy on heating unnecessary airflow. Hence, in order to achieve energy-efficient buildings the ventilation rate should be varied according to the demand, depending on occupancy level, pollutant emission and indoor air quality requirements.

In addition, to satisfy the second and third criteria of "20-20-20" and to make buildings more sustainable and to reduce their CO_2 emission, a heating source such as a heat pump is recommended to supply hot water to the hydronic heating system or domestic usage. The heat pump uses electricity to produce heat. In order to make the system more sustainable this electricity can be produced by renewable sources such as photovoltaic, PV. PV is usable year round performing even at low solar intensity and slight light during the cold and mostly dark winter time since it works on light not heat.

1.1 Objectives

This thesis presents the project results in three main parts.

1 The first part was to evaluate buildings with low-temperature heating systems in terms of energy consumption and thermal comfort. The purpose was to investigate whether buildings with low-temperature heating systems are following the Swedish rules for

energy consumption in the housing sector, BBR. For this purpose five new buildings with low-temperature heating systems including under floor heating and ventilation radiators were chosen (the main parts of a ventilation radiator are shown in Fig. 2). The results were presented in papers I and III.

2 The aim of the second part was to increase the energy efficiency of the buildings and to reduce the primary energy demand. For this purpose the potential for energy savings and its consequences on indoor air quality when using a variable air volume system was evaluated. The results were presented in papers II and IV.

3 The third part deals with improving the sustainability of the heating system by combining it with a renewable energy source. The building was equipped with an exhaust air heat pump. Photovoltaic (PV) as a renewable source was chosen to provide electricity for the heat pump. The results were presented in paper V.

1.2. Description of the building

The five investigated buildings were new-built semi-detached houses equipped with low-temperature heating system. All buildings were identical in terms of construction materials and geometry but different in compass orientation. The dwellings were numbered 27, 30, 31, 32 and 33. The number of inhabitants in dwellings 27 and 30 was four persons, and five persons were living in dwelling 31. Dwellings 32 and 33 both had three persons as occupants. Each building had three stories including hallway, living room, kitchen, bedrooms, toilet, and bathroom.

On the first floor, the heating system used was under-floor heating, and the ventilation supply devices were placed above the windows, so the cold supply air came directly inside without preheating. On the second and third floors the heating and ventilation systems were combined, this system is called "ventilation radiator". In a ventilation radiator the vent device is placed on the external wall behind the radiator, and it is connected to the radiator via a channel [6], see Fig. 2. So, the ventilation air was preheated between the radiator's panels before entering the building.

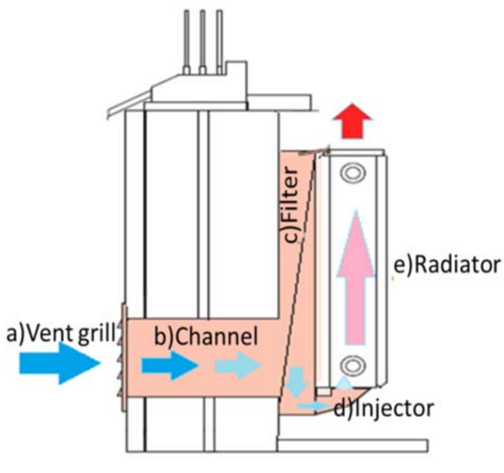

Fig. 2. Main parts of a ventilation radiator: a) Vent grill on the external wall, b) Channel through wall, c) Filter, d) Injector or inlet (with or without mixing of cold supply air with room air) e) Traditional radiator

The plan of the building and a schematic of airflow as well as heating systems are shown in Fig. 3. The exhaust devices were placed in the kitchen (10 $l \cdot s^{-1}$), bathroom (15 $l \cdot s^{-1}$), toilet (11 $l \cdot s^{-1}$), wardrobe (6 $l \cdot s^{-1}$), and third floor (18 $l \cdot s^{-1}$). The total airflow rate was 60 $l \cdot s^{-1}$, i.e. 0.55 h^{-1}, fulfilling the minimum requirements of BBR, i.e. 0.50 h^{-1} [5].

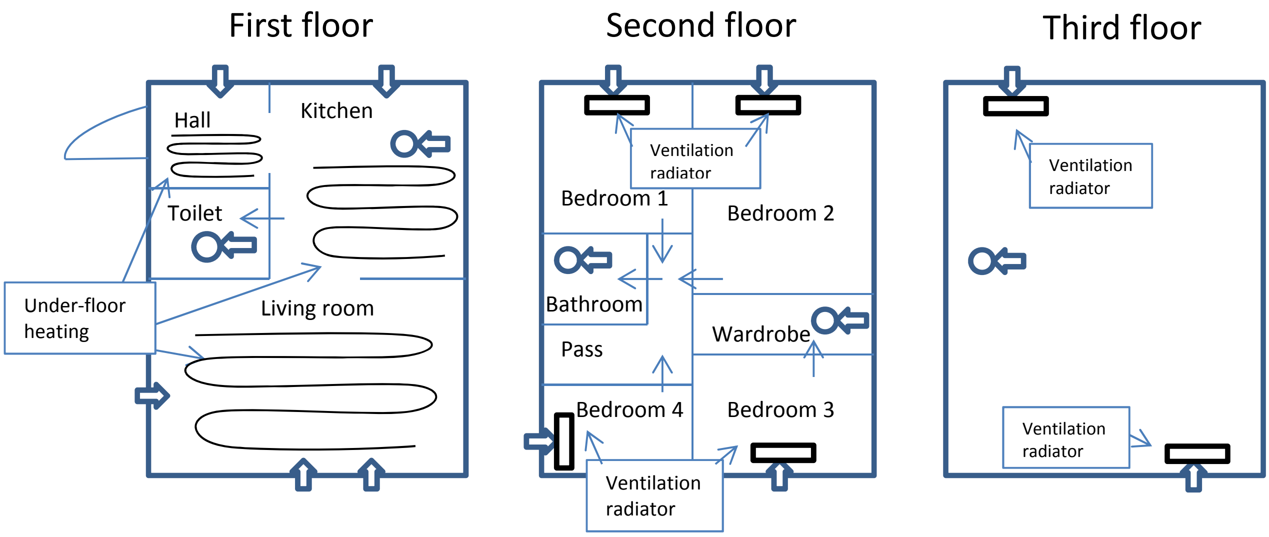

Fig. 3. Building plan and schematic of airflow and heating systems

The building envelope was very airtight with low mean U value, i.e. U_m was 0.28 $W \cdot m^{-2} \cdot K^{-1}$, see Table 1. So, not much energy could be saved by decreasing transmission loss through building materials and space heating demand. Hence, to increase the building efficiency it was suggested to reduce the ventilation loss by implementing a DCV system instead of a constant ventilation system. In the investigated buildings, as well as in buildings in general, during weekdays occupants go to work or school in the morning and return home in the afternoon. However, most ventilation systems in residential buildings are working with a constant air volume (CAV), which results in heating an unnecessary airflow. Therefore, in order to reduce the total energy consumption in buildings, a ventilation control system is required based on the presence of occupants and indoor air quality requirements. This arrangement is in line with Swedish building regulations, BBR [5]. However, the indoor air quality should be considered when decreasing the ventilation rate.

Table. 1. U values for building envelope

Structure	Roof	Floor	Exterior wall	Windows	Doors	U_m
U value, $W \cdot m^{-2} \cdot K^{-1}$	0.13	0.15	0.15	1.10	1.50	0.28
Area, m^2	65.0	58.0	116.0	20.2	4.0	

The heating source in the building was an exhaust air heat pump, which extracted the heat from outgoing air to produce hot supply water for the heating system and domestic hot water usage, see Fig. 4. The heat pump consumed direct electricity, so in order to increase the sustainability and efficiency of the system, the electricity needed can partly or totally be supplied by renewable energy sources, e.g. solar photovoltaic, PV.

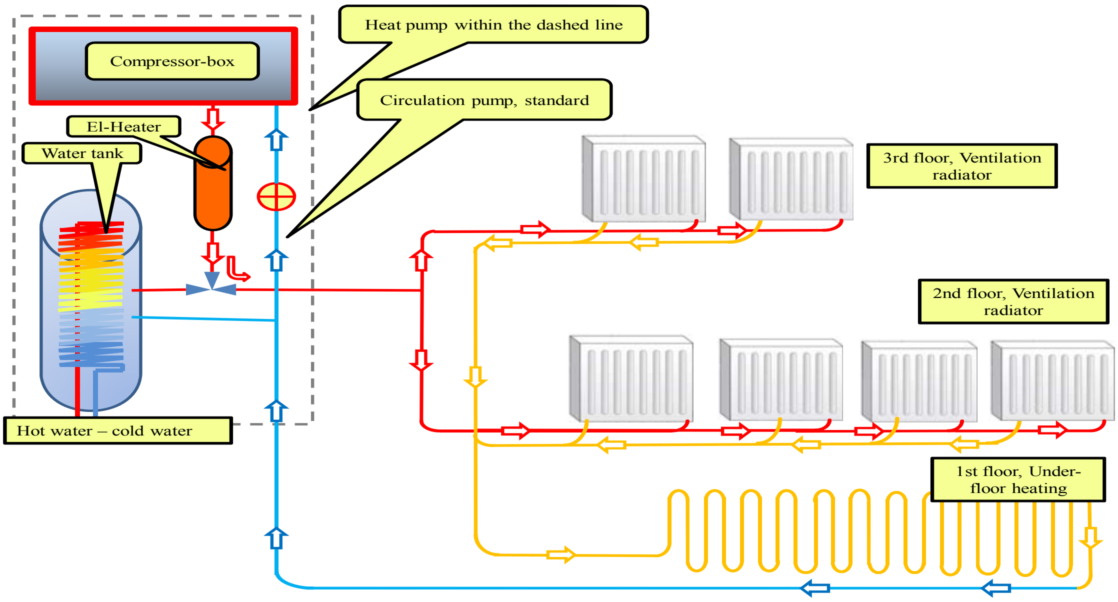

Fig. 4. Low-temperature hydronic heating system connected to the heat pump and water tank for producing domestic usage

2. Method

2.1. Simulation program

The main method used in all appended papers was the IDA Indoor Climate and Energy (ICE) 4 simulation program. IDA ICE 4 [7] is a dynamic multi-zone simulation tool to study the thermal indoor climate, indoor air quality as well as the energy consumption of the modeled building. IDA ICE 4 has a user-friendly interface, which makes it easy to build and simulate different cases. The model library of IDA ICE 4 was written in neutral model format (NMF) [8]. NMF is a program-independent language using differential algebraic equations for modeling the dynamic systems. The validation of IDA ICE 4 has been done in several studies [9-13]. In IDA ICE 4, the mathematical library is modeled with the equations from ISO 7730 [14]. Equations 1, 4 and 5 show convective (Q_{cv}) and radiative ($Q_{rad,occ}$) heat loads, and moisture (HumOcc) load from occupants.

$$Q_{cv} = f_{cl} \cdot h_{cl} \cdot 1.8 \cdot (t_{cl} - t_{air}) + 1.8 \cdot 0.014 \cdot M \cdot (34 - t_{air}) \tag{1}$$

$$h_{cl} = 2.38 \cdot (t_{cl} - t_{air})^{0.25} \qquad \text{for } 2.38 \cdot (t_{cl} - t_{air})^{0.25} > 12.1\sqrt{v_{air}} \tag{2}$$

$$h_{cl} = 12.1 \cdot \sqrt{v_{air}} \qquad \text{for } 2.38 \cdot (t_{cl} - t_{air})^{0.25} < 12.1\sqrt{v_{air}} \tag{3}$$

$$Q_{rad,occ} = 1.8 \cdot 3.96 \cdot 10^{-8} \cdot f_{cl} \cdot (t_{cl}^{\ 4} - t_{mrt}^{\ 4}) \tag{4}$$

$$HumOcc = 1.8 \cdot (3.05 \cdot 10^{-3} \cdot (5733 - 6.99(M \cdot 58 - W) - P_{vap}) + 0.42\,((M \cdot 58 - W) - 58.15) + 1.7 \cdot 10^{-5} \cdot M \cdot 58 \cdot (5867 - P_{vap}))/2501000 \tag{5}$$

where f_{cl} is the ratio of a man's surface area while clothed to a man's surface area while unclothed; h_{cl} is convective heat transfer coefficient, $W \cdot m^{-2} \cdot K^{-1}$; t_{cl} is surface temperature of clothing, °C; M is metabolic rate, MET; t_{air} is air temperature, °C; v_{air} is air velocity relative to human body, $m \cdot s^{-1}$; t_{mrt} is mean radiant temperature, °C; W is external work, $W \cdot m^{-2}$; P_{vap} is partial water vapor pressure, Pa.

In all appended papers, IDA ICE 4 was mainly used to evaluate the buildings in terms of energy consumption, energy savings and indoor air quality. In papers I and III the simulation program was used to predict the energy demand and thermal comfort. The simulation results were validated against on-site measurements. In papers II and IV IDA ICE 4 was used to predict the energy savings and indoor air quality after implementing a variable air volume (VAV) ventilation system. In IDA ICE 4 the VAV model can be controlled by humidity, CO_2, temperature or pressure in each zone.

In paper V, another simulation program called WINSUN was used to design and evaluate photovoltaic, PV. WINSUN [15] is a system simulation program for designing solar collectors and PV. This program developed at Lund University is an abbreviation of the Windows version of the MINSUN program usable in DOS. WINSUN is based on the TRNSYS program [16], and thermal energy equations are solved based on a modular approach depending on the input data. This program gives the output of PV or solar collector system in $kWh \cdot m^{-2}$ based on the efficiency, location, tilt of the system, and weather data including diffuse and beam radiation [17]. The validation of WINSUN was conducted by comparing the simulation results with site measurements [18]. In paper V, WINSUN simulations were used to evaluate the performance and output of fixed PV with 15 % efficiency at 23° tilt towards the south, according to the plan of the chosen building.

2.1.1. Description of the IDA ICE 4 model

The investigated building type was modeled in IDA ICE 4 similar to the actual geometry, see Fig. 5. The model was divided into 12 zones in each dwelling, and for each zone a different exhaust and supply air flow rate was defined. For ventilation system, a mechanical exhaust system was defined, and the supply devices were placed on external walls as openings. In the simulations cooling energy was not considered since there was no cooling device supplied in the buildings. All dwellings were modeled with different orientations and different internal heat gains. A detail description of the model is given in papers I and III.

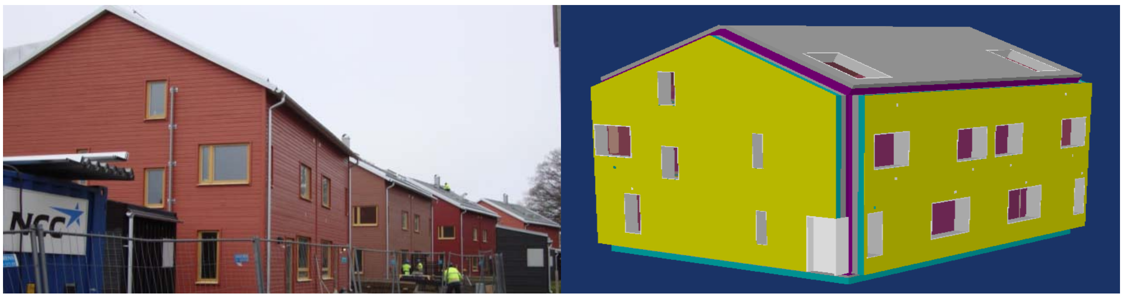

Fig. 5. Left: photo of four identical semi-detached houses in Stockholm; Right: simulation model frame of the houses in the IDA ICE 4 software

In order to investigate the potential for energy savings in the investigated buildings, the CAV ventilation system was changed to a VAV system. The clock-controlled VAV system was based on step increased/ decreased pressure (speed) by ventilation fan. Start time to decrease the ventilation rate was at 8:00 when occupants left the house. According to the defined schedule, the ventilation fan speed was decreased to 27 % of its initial speed during unoccupancy from 8:00 till 18:00, shown in Fig. 6. The value of 0.27 shows the ratio of 0.100 (minimum ventilation rate) to 0.375 $l \cdot s^{-1} \cdot m^{-2}$ (normal ventilation rate in the building studied). The reduction in fan power (P) and thus energy used is a third power of flow (q) variation, shown in Eq. (6).

$$P_{0.1} = P_{0.375} \cdot \left(\frac{q_{0.1}}{q_{0.375}}\right)^3 \tag{6}$$

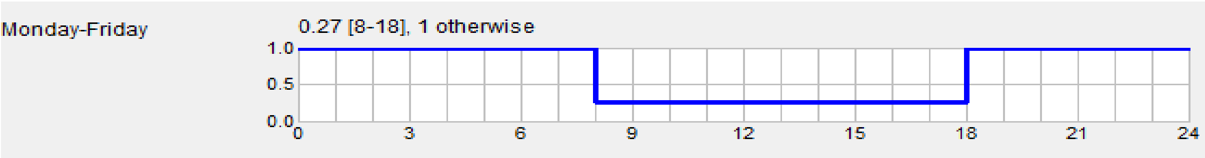

Fig. 6. Profile for the ventilation fan speed for weekdays, 1.0 for working at full power during occupancy from 18:00 till 08:00, and 0.27 for working during unoccupancy from 8:00 till 18:00 with 27 % of full speed.

In order to investigate how many hours the ventilation air is allowed to be decreased in terms of acceptable IAQ, four different cases with different mean air change rates were considered, see Table 2. LVR in the table is the abbreviation of low ventilation rate, the number after refers to the duration of decreased ventilation rate, e.g. LVR10h means that ventilation rate was decreased for 10 hours. Each VAV control was modeled in IDA ICE 4 taking into account energy savings and indoor air quality. A detail description of the model is found in papers II and IV.

Table 2. Start and stop time and mean air change for each proposed VAV control

	LVR4h	LVR6h	LVR8h	LVR10h
Start time	8:00	8:00	8:00	8:00
Stop time	12:00	14:00	16:00	18:00
Mean air change averaged over a day, h^{-1}	0.48	0.45	0.42	0.38

2.2. On-site measurements and questionnaire

To conduct site measurements, occupants were asked to read and report heat pump electricity consumption month by month. This consumption was used for space heating, hot water and ventilation fan. The starting date for these measurements was on 14th of December 2011, the date when the buildings were occupied. The last value was reported on 13th of December 2012; and thus their annual energy consumption was measured.

To survey the perceived thermal comfort and indoor air quality a questionnaire was distributed to occupants. In the questionnaire they were asked how satisfied they were with the mean temperature and air quality in different rooms and whether they had felt any discomfort with respect to draught. The questionnaire is found in the Appendix.

3. Results and discussions

Results of the study were presented in detail in each appended paper, here follows a short review of the results and also the discussions are presented.

The results of papers I and III showed that the buildings with low-temperature heating systems not only met the energy requirements of Swedish building rules but also provided a good level of thermal comfort. The total measured annual energy consumption in all building studied varied between 48 – 54 kWh·m^{-2}, i.e. lower than 55 kWh·m^{-2} as limiting value assigned by BBR for climate zone 3, including Stockholm. This consumption included the energy demand for domestic hot water, heating and ventilation systems. The measured and simulated results of total energy consumption for the five houses studied are given in Fig. 7. The share of each usage is shown separately in Fig. 8. DHW consumption was estimated from heat pump electricity consumption during the summer months, since the only reason for the heat pump to provide heating is because of DHW consumption during this period. A variation in DHW consumption between buildings was observed, probably depending on different living habits. The consumption of the air handling unit (AHU) was achieved by knowing the specific fan power (SFP), which was 0.9 kJ·m^{-3} and working 24 h per day.

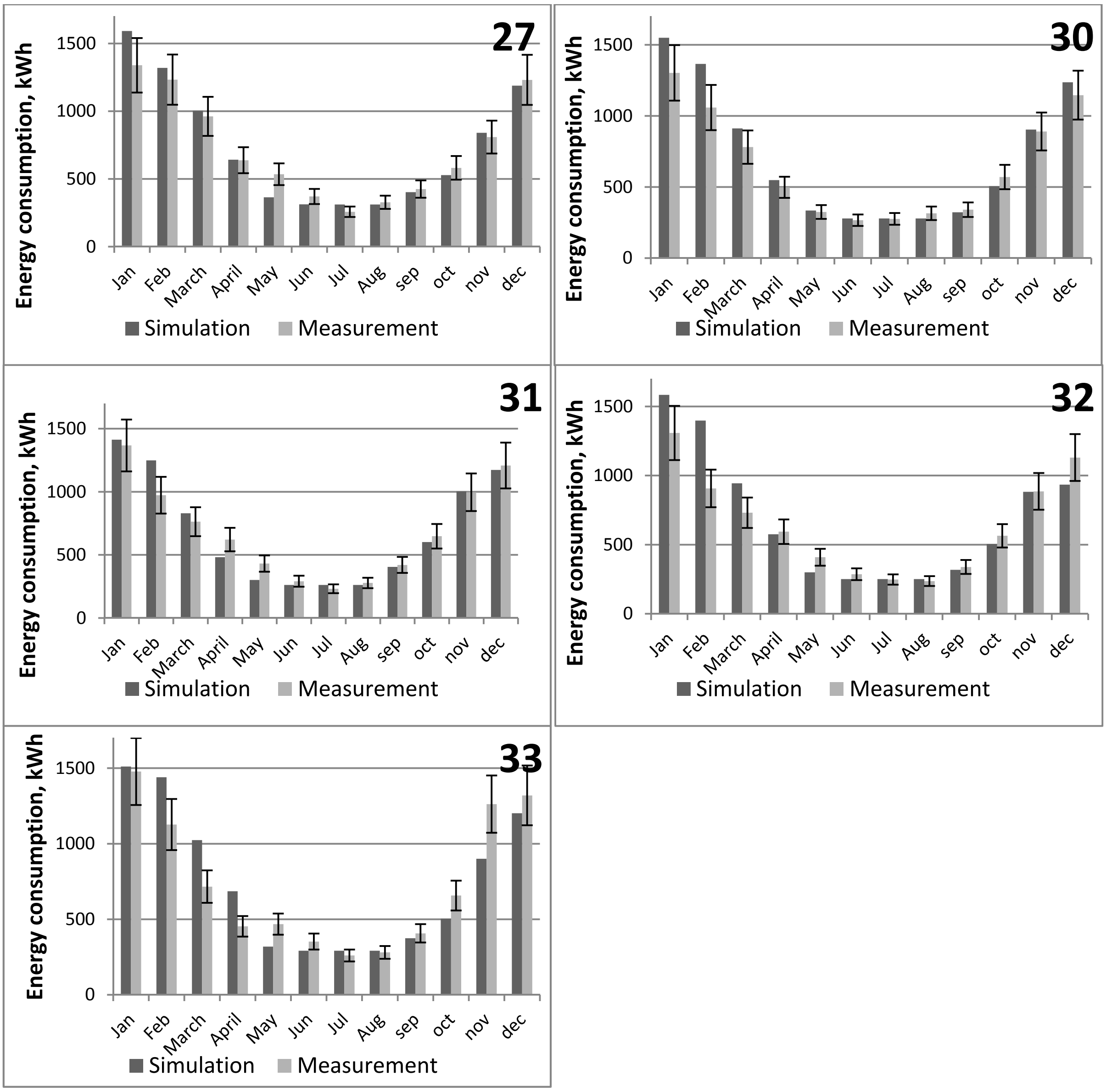

Fig. 7. Comparison between measured and calculated energy consumption for the five houses studied

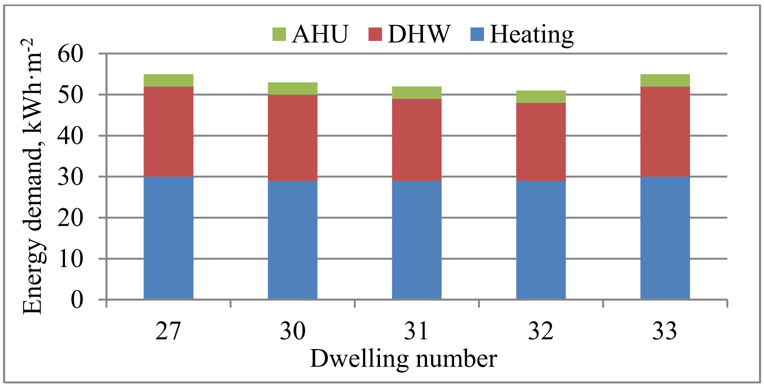

Fig. 8. Measured energy consumption for each dwelling for air handling unit (AHU), domestic hot water (DHW) consumption and heating system

All these buildings were identical in terms of construction materials and ventilation systems, so the heat losses through ventilation and materials were approximately the same. Results showed, however, that the total energy consumption differed depending on the occupants' behavior and internal (by occupants, lights, and equipment)/ external (by solar radiation) heat gains. The simulated internal and external heat gains for each dwelling is presented in Fig. 9. The consumption by house equipment and light was assumed to be the same, so heat gains from those are equal in all dwellings. However, due to different number of inhabitants and building orientation, the heat gains from humans and solar radiation are different.

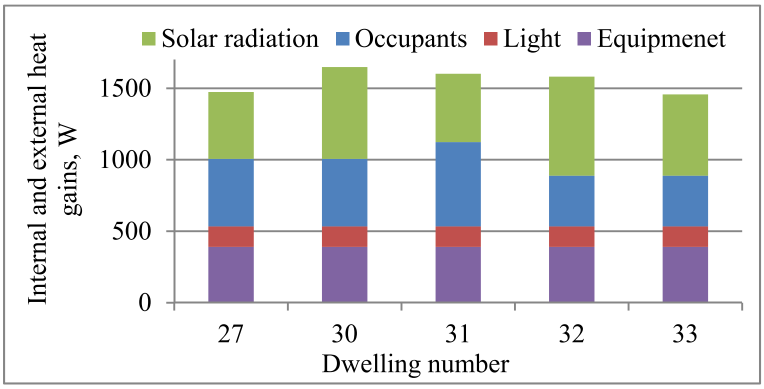

Fig. 9. Simulated internal (occupants, light and equipment) and external (solar radiation) heat gains for each dwelling

Thanks to technology, the total heat loss in a new building is reduced compared to old buildings; however, the occupants' behavior still has the major role in determining the total energy consumption. In all studied buildings, tap water usage was responsible for approximately 40 % of the total consumption. It shows that the awareness of people regarding correct use of energy still needs to be addressed. Hence, consideration of the user behavior is very important since it may cause substantial differences in energy use. This shows that savings can also be achieved through information.

The questionnaire distributed to occupants showed a common response of feeling a cold draught on the first floor, where fresh air was directly brought into the house without preheating, but all occupants stated that the comfort was better on the second and third floors, where ventilation radiators were installed. This response can be explained by the slow thermal control of UFH (large thermal mass flow of water) on the first floor, causing higher temperature fluctuations [19]. So, occupants may feel cold before the system has reached stable conditions. However, on the second and third floors with ventilation radiators, the temperature is more stable, causing them to feel more comfortable. Generally, all occupants were satisfied with the mean temperature in their dwelling.

The results of papers II and IV show that by implementing a VAV ventilation system the energy requirements for ventilation air and electricity for the ventilation fan decreased by 23 % and 38 %, respectively. In this VAV control, the low ventilation rate, i.e. 0.1 $l \cdot s^{-1} \cdot m^{-2}$ was applied during the whole unoccupancy from 8:00 till 18:00. The indoor air quality was also investigated in terms of concentration of pollutant gases such as CO_2 and volatile organic compounds (VOCs), see Eq. (7):

$$c = c_o \cdot e^{-nt} + \frac{\dot{m}}{q}(1 - e^{-nt}) \tag{7}$$

where c is pollutant concentration in room, ppm; c_0 is initial concentration at t=0, ppm; n is the number of air changes per hour, h^{-1}; $\dot{m}$ is emission rate, $kg \cdot h^{-1}$; t is time, h; and q is ventilation outdoor airflow rate, $m^3 \cdot h^{-1}$.

Results show that when decreasing the ventilation rate during the whole unoccupancy, the concentration of VOCs was slightly higher than acceptable (0.1 ppm [20]) when occupants arrived home, so in paper II it was suggested to increase the ventilation rate to the normal requirements, i.e. 0.35 $l \cdot s^{-1} \cdot m^{-2}$ two hours before a home is occupied. However, the concentration of CO_2 was within the acceptable range every day (below 1000 ppm), since decreasing the ventilation rate happened at the same time as the main pollutant source, i.e. humans, left the house. Fig. 10 shows how the pollutant gas concentration varied over 48 hours during two consecutive days when decreasing the ventilation rate from 8:00 till 18:00.

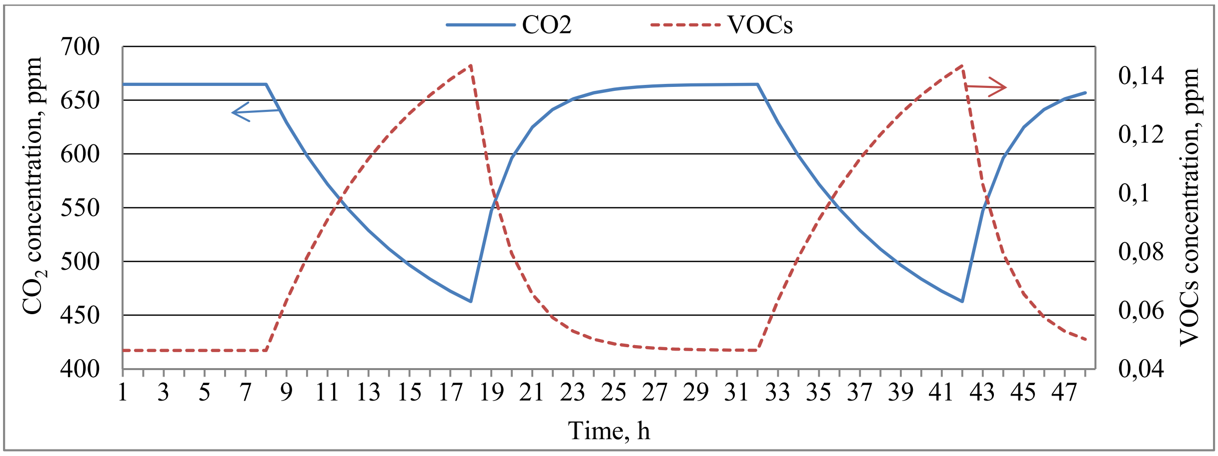

Fig. 10. Pollutant gas concentration (CO_2 and VOCs) variation over 48 hours showing the effect of decreasing the ventilation rate from 8:00 till 18:00

The ventilation efficiency can also be measured by concentration decay. For perfect mixing 63 % of pollutants have left the room after a period of τ_n (1.8 h) from initial concentration, and after $4\tau_n$ (7.2 h) this value is increased to 98 %. Eq. (8a, 8b) shows how the concentration level c in a ventilated room with time constant τ_n is related to real time t. τ_n is equal to the room volume divided by the ventilation flow rate.

$$-\frac{dc}{dt} = \frac{1}{\tau_n} \cdot c \qquad (8a)$$

$$c = c_0 \cdot e^{-t/\tau_n} \qquad (8b)$$

Theoretical concentration decay for different mixed airflow rates based on Eq. (8) is shown in Fig. 11. An exposure impulse was released at 11:00, i.e. seven hours before occupants arrive home. As can be seen, after 4 τ_n, i.e. 7 h, the concentration level for LVR10h is still 15 % of the initial value; however, for the CAV system only 2 % of the initial value has been left in the room.

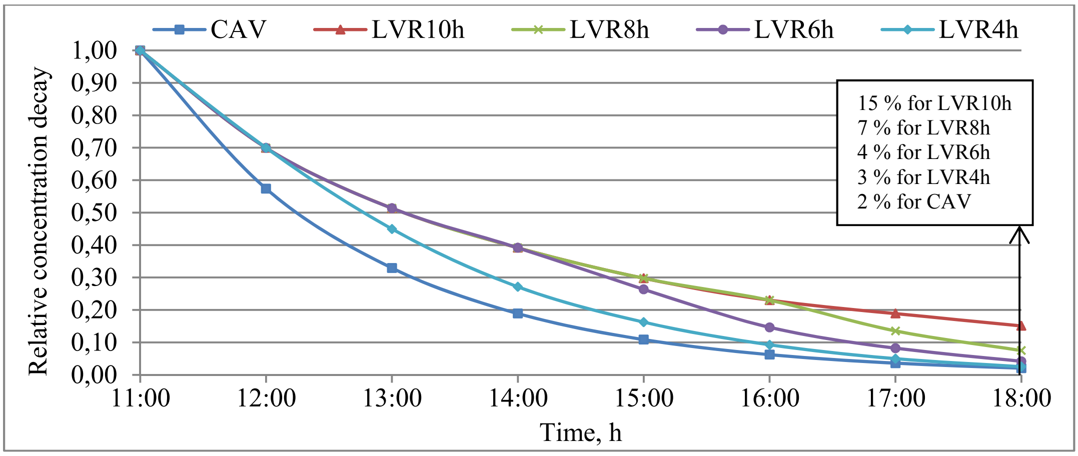

Fig. 11. Relative concentration decay for different VAV durations showing the concentration level seven hours before 18:00, i.e. when occupants arrive home

The investigation in paper V was based on increasing the efficiency and sustainability of the heating system by combining heating sources, i.e. heat pump, with renewable energy. The results show that a relatively small area of PV, approximately 7 m^2, can cover 15 % of total electricity demand. It is obvious that a larger PV can cover more, but in order to avoid overproduction to the heat pump, it was assumed that a balance between demand and production is required during the summer months. So, there was no mismatch between electricity consumption by heat pump and electricity production by PV during the period of highest production, i.e. the summer months. The results of monthly electricity production by PV and electricity demand of heat pump are shown in Fig. 12.

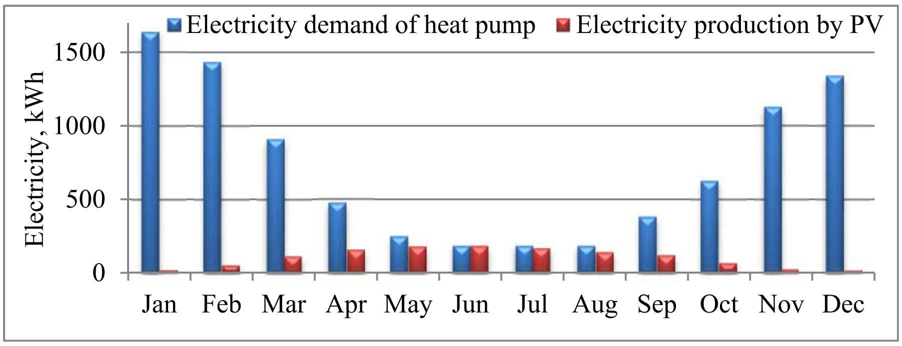

Fig. 12. Simulated monthly electricity demand of heat pump and electricity production by 7 m^2 PV

4. Conclusion

This work presents a summary of all efforts done by the author since September 2011. The work was first started to evaluate buildings with low-temperature heating system in terms of energy consumption and thermal comfort. On-site measurements as well as simulations by the IDA ICE 4 program were used for this purpose. Results of the first step showed that the investigated buildings were following Swedish building regulations, BBR. Also, the buildings provided a good level of thermal comfort.

The next step was to minimize this consumption by introducing a VAV ventilation system instead of CAV. The building was unoccupied for several hours during daytime and there was no need to have the same amount of ventilation air as provided during occupancy. In this phase, the aim was to study the consequences of ventilation air reduction during unoccupancy on indoor air quality and energy savings. The IDA ICE 4 simulation program as well as analytical calculations were used for this purpose. Results showed that reducing ventilation air during unoccupancy caused decreased energy requirements for ventilation by up to 23 %. However, the investigation regarding the indoor air quality revealed that the level of VOCs was not within the acceptable range when occupants arrived home. So, it was recommended to increase the ventilation rate to normal requirements, i.e. 0.35 $l \cdot s^{-1} \cdot m^{-2}$, two hours before the home was occupied. This strategy caused savings of up to 20 % in energy requirements for ventilation.

In this work, a combination of heat pump and photovoltaic was also introduced. The results indicated that by implementing a relatively small PV area of 7 m^2, 15 % of the annual energy demand to the heat pump can be covered.

All efforts presented here show the European goal of increasing energy efficiency by 20 % and using 20 % of renewable energy in buildings by the year of 2020 is not too far away.

5. Future work

A decreased supply-water temperature to the heating system compared to a reference system is a key element. With a heat pump this means energy gains due to higher coefficient of performance (COP) value. So, the main focus will be based on energy savings when combining low-temperature heating systems with a heat pump. Conventional heating systems with and without local preheating of incoming ventilation air also will be compared to each other in terms of total energy consumption in buildings. This will be done for the following room heating and ventilation systems:

- Low temperature radiator heating system without air preheating vs. low temperature radiator heating system with air preheating
- Under-floor heating without air preheating vs. under floor heating with air preheating
- Baseboard heaters without air preheating vs. baseboard heaters with air preheating

The aim of the next step is to make measurement in a climate chamber and to compare the following aspects when using different low-temperature heating and ventilation systems:

- Monitoring the energy consumption of buildings or HVAC systems
- Thermal comfort (operative temperature)
- Indoor air quality
- Ventilation system performance using tracer gas
- Mathematical analysis and modeling of complex flow systems, including heat, air and moisture transfer, contaminant migration and dispersal and thermal comfort

Previous modeling and simulation results will be used and extended. Required heating power, energy use and thermal comfort with different supply-water temperatures will be analyzed and compared to a reference system. Measurements of indoor air quality and thermal comfort will be performed in a climate chamber.

In previous simulation modeling the role of heat pumps in energy savings was not considered. In future work the modelling of heat pumps will be improved and the influence of them on energy saving when decreasing the supply temperature to the radiators and decreasing the ventilation air flow.

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Appendix:

1. In a typical week, how many hours do you spend in your house?
   ☐ 60 or less ☐ 61-80 ☐ 81-100 ☐ 101-120 ☐ More than 120

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2. Considering energy use, how efficiently is this building performing in your opinion?

Very energy efficient ☐ ☐ ☐ ☐ ☐ ☐ ☐ Not at all energy efficient

Comments/Kommentarer:

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3. a. How satisfied are you with the temperature in your house?

Very Satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very Dissatisfied

b. In hot weather, the temperature in my house is

Often too hot ☐ ☐ ☐ ☐ ☐ ☐ ☐ Often too cold

c. In cold weather, the temperature in my house is

Often too hot ☐ ☐ ☐ ☐ ☐ ☐ ☐ Often too cold

d. When is this most often a problem?

☐ Morning (before 11 am)
☐ Mid-day (11 am-2 pm)
☐ Afternoon (2 pm- 5 pm)
☐ Evening (after 5 pm)
☐ Weekends or holidays
☐ No particular time

---

4. Do you perceive (feel) any discomfort?

☐ Humidity too high (damp), Kommentarer: ……………………………………….
☐ Humidity too low (dry), Kommentarer: ……………………………………….

☐ Air movement too high, Kommentarer: ……………………………………
☐ Air movement too low, Kommentarer: ……………………………………
☐ Hot/cold surrounding surface, (Walls, floor, ceiling) Kommentarer:…………..
☐ Draft from Windows, Kommentarer: ……………………………………
☐ Draft from ventilation (supply), Kommentarer: ………………………………..
☐ Thermostat is inaccessible, Kommentarer: …………………………………
☐ Thermostat is adjusted by other people, Kommentarer: ……………………….
☐ Heating/cooling device does not respond quickly enough to thermostat
☐ Other, Kommentarer: ………………………………………………

5. How satisfied are you with the following aspects of your kitchen?

Temperature conditions
Very satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very dissatisfied

Air quality
Very Satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very Dissatisfied

6. How satisfied are you with the following aspects of your living room?

Temperature conditions
Very satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very dissatisfied

Air quality
Very Satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very Dissatisfied

7. How satisfied are you with the following aspects of your bed rooms?

Temperature conditions
Very satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very dissatisfied

Air quality
Very Satisfied ☐ ☐ ☐ ☐ ☐ ☐ ☐ Very Dissatisfied

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Energy performance of low temperature heating systems in five new-built Swedish dwellings: A case study using simulations and on-site measurements

Arefeh Hesaraki*, Sture Holmberg

Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Brinellvägen 23, 100 44 Stockholm, Sweden

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A B S T R A C T

In Europe, high energy consumption in built environments has raised the need for developing low energy heating systems both in new building and in retrofitting of existing buildings. This paper aims to contribute by presenting annual results of calculated and measured energy consumption in five new-built semi-detached dwellings in Stockholm, Sweden. All buildings were equipped with similar low temperature heating systems combining under-floor heating and ventilation radiators. Exhaust ventilation heat pumps supported the low temperature heating system. Buildings were modeled using the energy simulation tool IDA Indoor Climate and Energy (ICE) 4, and energy consumption of the heat pumps was measured. Results showed that calculated and measured results were generally in agreement for all five dwellings, and that the buildings not only met energy requirements of the Swedish building regulations but also provided good thermal comfort.

1. Introduction

Energy consumption in European buildings accounts for 40% of total primary energy usage [1], and energy required for heating is responsible for most of this, i.e. more than 70% [2]. In the building sector there is a considerable potential for saving energy, such as by using materials with low heat transfer coefficient (low *U* value), tight building envelopes or using energy-saving equipment in heat recovery from exhaust air, such as heat exchangers or exhaust air heat pumps. All these measures can lead to reducing the duration of the heating season and thus the total space heating load. In low energy buildings, the low temperature heating system usually works with a supply water temperature below 45 °C.

The number of heat pumps used as a heating source in Sweden has increased widely during recent decades. Myhren and Holmberg [3] showed that a combination of LTHH system and heat pump was more thermally efficient compared to a high temperature heating system. As a rule of thumb, the coefficient of performance (COP) of a heat pump improves by 1–2% for every degree reduction in supply water temperature [4]. The lower the supply temperature, the higher the COP and the more energy efficient and sustainable is the system thus created. In addition, LTHH systems are not only energy efficient, but also exergy efficient and environmental friendly [5].

Using an LTHH system increases not only energy efficiency but also thermal comfort [6–8]. Three common types of LTHH systems are under-floor heating (UFH), wall heating and ceiling heating. All are able to work with low supply water temperature due to a large heat transmitting surface area. Another less well-known type of LTHH system is the ventilation radiator, which due to high convection heat transfer can work with low supply water temperature. The ventilation radiator [9] (Fig. 1) is a combination of ventilator and radiator. The supply air vent is located on the wall behind the radiator and is connected to the radiator through a channel. In this combined system, cold fresh air is forced to pass through the radiator panels, due to buoyancy forces and a constant under-pressure in the building, created by exhaust fans. Hence, the air is preheated before entering the building.

The performance of the ventilation radiators has been laboratory tested [9]. Measurements have shown that the ventilation radiators have potential to raise the temperature of cold air by up to 30 °C or more in winter time. The critical factor in using a combination of ventilation radiators and mechanical ventilation is the airtightness of building [9] with respect to the efficiency of radiator and total energy consumption. In other words, a high infiltration rate will lead to an increase in the non-preheated air that does not

* Corresponding author. Tel.: +46 8 790 48 84; fax: +46 8 790 48 00.
E-mail addresses: arefeh.hesaraki@byv.kth.se, arefeh@kth.se (A. Hesaraki).

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Nomenclature

ACH	air change per hour
AHU	air handling unit
BBR	Swedish National Board of Housing, Building and Planning
BES	building energy simulation
CAV	constant air volume
CFD	Computational Fluid Dynamics
COP	coefficient of performance
DHW	domestic hot water
f_{cl}	ratio of man's surface area while clothed to man's surface area while unclothed
h_{cl}	convective heat transfer coefficient between air and clothes (W m^{-2} K^{-1})
HumOcc	humidity load from occupants (kg s^{-1})
HVAC	heating, ventilation and air conditioning
IDA ICE	IDA Indoor Climate and Energy
IEA	International Energy Agency
ISO	International Organization for Standardization
ITM	Swedish Institute of Applied Mathematics
LTHH	low temperature hydronic heating
M	metabolic rate (MET)
NCC	a leading construction and property development company in the Nordic region
NMF	neutral model format
PMV	predicted mean vote
PPD	predicted percentage of dissatisfied
P_{vap}	partial water vapour pressure (Pa)
Q_{cv}	convective heat load from occupants (W)
Q_{radocc}	radiative heat load from the occupants (W)
SBUF	The Development Fund of Swedish Construction Industry
SHGC	solar heat gain coefficient
SMHI	Sweden's Meteorological and Hydrological Institute
t_{air}	air temperatures (°C)
t_{cl}	surface temperature of clothing (°C)
t_{mrt}	mean radiant temperature (°C)
UFH	under-floor heating
U_m value	mean U-value (W m^{-2} K^{-1})
U value	heat transfer coefficient (W m^{-2} K^{-1})
v_{air}	relative air velocity to human body (m s^{-1})
VR	ventilation radiator
W	external work (W m^{-2})
WC	water closet

pass through the radiator, causing additional need for space heating. The airtightness of the building during test condition is usually measured at a reference pressure of 50 Pa difference between inside and outside. A suitably high pressure, such as 50 Pa, is chosen not only so as to minimize the effects of stack-induced and wind-driven air flow, but also so as to be able to compare different buildings at the same reference value. This factor is especially important when retrofitting existing buildings with ventilation radiators.

One of many positive qualities of ventilation radiators is enhancement of forced convection by blowing cold air between the radiator's panels, allowing reduction of supply water temperature to the radiator without sacrificing heat output. Convective heat transfer is not only increased by high air velocity of incoming air, but also by the great temperature difference between cold incoming air and the heating unit. A previous study [9] showed that higher air velocity and greater temperature difference produced increased heat output in the radiators as a result of heat transfer enhancement. In addition, both a theoretical study [10] and

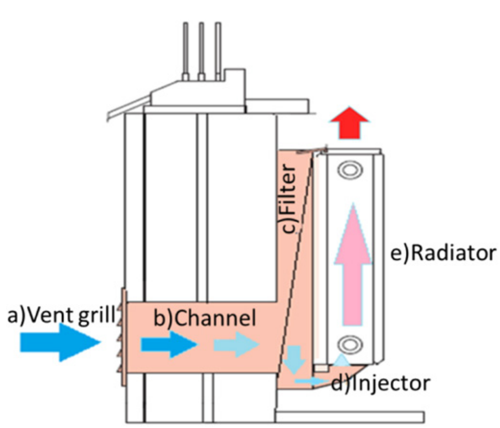

Fig. 1. Main parts of a ventilation radiator: (a) Vent grill on the external wall, (b) Channel through wall, (c) Filter, (d) Injector or inlet (with or without mixing of cold supply air with room air), (e) Traditional radiator [9].

laboratory measurements [9] have shown that working with the same supply water temperature, the integration of heating components (baseboards, radiators) with ventilation function leads to increased heat output from the heating system in comparison to conventional system. The combined heating and ventilation system gave twice the heat output per unit length compared to traditional radiator system due to increasing convection heat transfer. Furthermore, laboratory measurements [9] showed that a ventilation radiator working with 35 °C supply water temperature had the same heat output as a traditional radiator working with a supply water temperature of 55 °C.

Previous studies [5,9–11] focused on developing the ventilation radiator in the laboratory and on modeling by CFD (Computational Fluid Dynamics), with the question of how it functions in reality still to be addressed. According to the authors' knowledge there are no published studies that report the energy performance of a building with ventilation radiator. All previous studies are limited to modeling with CFD and analytical model, and no study report how the ventilation radiator functions in reality, or whether buildings equipped with ventilation radiators are following the Swedish building regulations. Previous laboratory measurements and simulations, however, led to the hope that this kind of radiator could be an energy-saving system. To evaluate its energy performance and thermal comfort in reality, five semi-detached buildings (Fig. 2) equipped with ventilation radiators were chosen. A detailed comparison of the measured and calculated electricity consumption for heating and ventilation (by heat pump and fan) in the five semi-detached houses was then made, from 14th December 2011 to 13th December 2012. The calculation of energy requirements was made using the building energy simulation (BES) tool IDA Indoor Climate and Energy (ICE) 4, hereafter called IDA ICE 4.

2. Description of the case study buildings

2.1. Building dimensions and construction

Five semi-detached, new-built dwellings in Stockholm were selected for both measurements and simulation. These five had responded positively to our request for reporting on the heat pump

Fig. 2. Some of the semi-detached houses in Stockholm that were selected for study.

electricity consumption. The dwellings were numbered 27, 30, 31, 32 and 33. The geometry of all houses was the same but compass orientation differed. They each had three stories with total floor area of 160 m^2 including kitchen, bathrooms, living room and bedrooms. The floor plan and installed ventilation system for a typical dwelling are presented in Fig. 3. *U* values for different parts of the building envelope are presented in Table 1. The average *U* value was 0.28 W m^{-2} K^{-1}, significantly lower than the limited mean value assigned by the Swedish building regulations (BBR) [12], i.e. 0.5 W m^{-2} K^{-1}. For the glazing, the solar heat gain coefficient (SHGC) was 0.5. This represented half of incident solar radiation admitted through a window and released inward, both by directly transmitted and absorbed radiation.

2.2. HVAC (heating, ventilation and air conditioning) system

2.2.1. Heating system

The heating systems in the buildings studied were all the same, i.e. hydronic low temperature systems that included under-floor heating for the first floor, and ventilation radiators for the second and third floors, (Fig. 4). In addition, bathroom had electrical under-floor heating. The actual temperatures of supply and return flows from radiators were 45 °C and 30 °C respectively. Return flow from the radiators was supplied to the under-floor heating. The heat source for both space heating load and domestic hot water (DHW) production was an exhaust air heat pump that extracted the heat from exhaust air to heat water (air–water source). As can be seen in Fig. 4, the heat pump is also equipped with electrical auxiliary heater of 4.0 kW that is used when instantaneous hot water is needed. The exhaust air heat pump used in the test buildings was ComfortZone CE50. This heat pump has a variable compressor speed, which is able to adjust flow temperature to heating demand so as to maintain indoor air temperature around a suitable level of 21 °C. However, since the heat pump was placed in the bathroom, where temperature is often higher than in other rooms, an incorrect indication of mean indoor temperature was possible. The average coefficient of performance (COP) of an exhaust air heat pump producing hot water at 50 °C is 2.7 [13].

2.2.2. Ventilation and infiltration

In the exhaust ventilation system, polluted air was extracted from exhaust devices placed in kitchen, bathroom and wardrobes at a constant air volume (CAV) of 60 l s^{-1} for 24 h·day^{-1}. Exhausted air was replaced by fresh outdoor air entering through inlets in bedrooms and living room due to a 10 Pa constant under-pressure in the building. The inlet for fresh supply air was placed above the windows of the first floor, and cold air was brought into the house without preheating. However, for the second and third floors the cold air was preheated by ventilation radiators. The total exhaust air flow rate was 60 l s^{-1}, i.e. 10 l s^{-1} extracted from kitchen, 26 l s^{-1} from bathroom and WC, 6 l s^{-1} from wardrobes and 18 l s^{-1} from third floor. The total air flow rate was 0.375 l s^{-1} m^{-2} of floor area, fulfilling the minimum requirement [12] for residential buildings, i.e. 0.350 l s^{-1} m^{-2} of floor area. In the buildings studied, the blower door test showed that air leakage was 0.63 l s^{-1} m^{-2} of external surface under 50 Pa pressure difference. This showed a good airtightness relative to 0.80 l s^{-1} m^{-2} requirements for Swedish houses.

2.3. Weather conditions

The buildings studied are located in Stockholm with average daily temperature of 6.6 °C. The weather data was given by Swedish Meteorological and Hydrological Institute (SMHI) for the Stockholm region where the tested buildings were located. The coldest temperature was −28.8 °C on 23rd February 2012 and the warmest temperature was 27.5 °C on 2nd July 2012.

3. Methods

3.1. IDA ICE 4

Simulation programs are often used nowadays to predict and analyze the performance of buildings and HVAC systems. IDA ICE 4 [14] is a dynamic simulation tool providing simultaneous dynamic simulation of heat transfer and air flow by creating a mathematical model [15] to calculate the heating and cooling load in a building, and predict the thermal comfort and indoor air quality based on building properties defined by the user. IDA ICE 4 was initially developed at KTH Royal Institute of Technology and the Swedish Institute of Applied Mathematics, ITM. The first version of the model library of IDA ICE 4 was written in neutral model format (NMF) within IEA Task 22. NMF [16] is a program independent language using differential algebraic equations for modeling the dynamic systems. In IDA ICE 4, the mathematical library is modeled with the equations from ISO 7730 [15]. Eqs. (1), (6) and (7) show convective (Q_{cv}) and radiative ($Q_{rad,occ}$) heat loads, and moisture (HumOcc) load from occupants. Symbols are explained in Nomenclature.

$$Q_{cv} = f_{cl}\cdot h_{cl}\cdot 1.8\cdot(t_{cl}-t_{air}) + 1.8\cdot 0.014\cdot M\cdot(34-t_{air}) \tag{1}$$

$$h_{cl} = 2.38\cdot(t_{cl}-t_{air})^{0.25} \quad \text{for } 2.38\cdot(t_{cl}-t_{air})^{0.25} > 12.1\sqrt{\nu_{air}} \tag{2}$$

$$h_{cl} = 12.1\cdot\sqrt{\nu_{air}} \quad \text{for } 2.38\cdot(t_{cl}-t_{air})^{0.25} < 12.1\sqrt{\nu_{air}} \tag{3}$$

$$f_{cl} = 1.00 + 1.29\cdot I_{cl} \quad \text{for } I_{cl} < 0.078 \tag{4}$$

$$f_{cl} = 1.05 + 0.645\cdot I_{cl} \quad \text{for } I_{cl} > 0.078 \tag{5}$$

$$Q_{rad,occ} = 1.8\cdot 3.96\cdot 10^{-8}\cdot f_{cl}\cdot\left(t_{cl}^4 - t_{mrt}^4\right) \tag{6}$$

$$\begin{aligned}\text{HumOcc} = 1.8\cdot\Big(&3.05\cdot 10^{-3}\cdot(5733-6.99(M\cdot 58-W)-P_{vap})\\ &+0.42\,((M\cdot 58-W)-58.15)+1.7\cdot 10^{-5}\cdot M\\ &\cdot 58\cdot(5867-P_{vap})\Big)\Big/2501000\end{aligned} \tag{7}$$

Validation of IDA ICE 4 was performed by several studies [17–21]. In IDA ICE 4, based on the set indoor temperature all

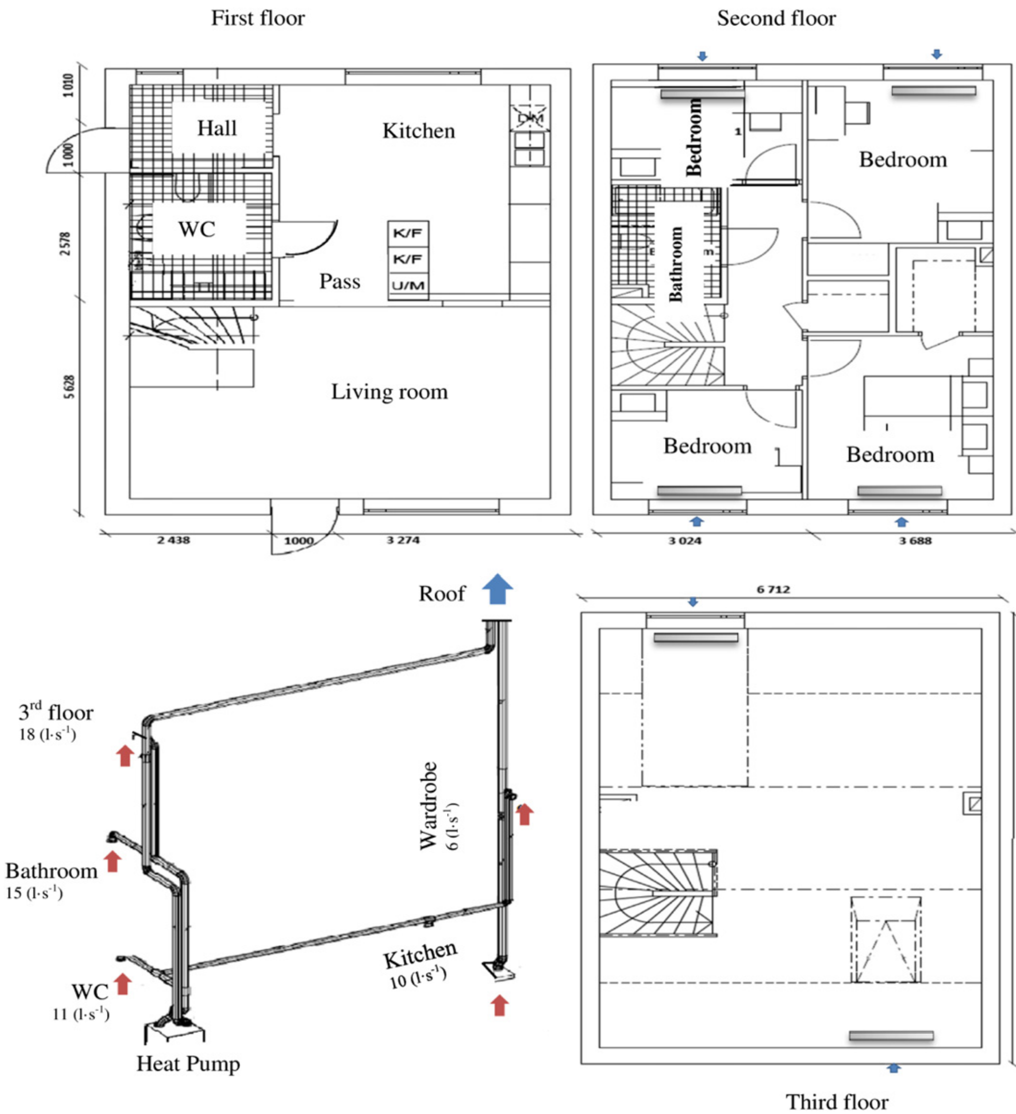

Fig. 3. Floor plan showing the location of ventilation radiators in second and third floors and the ventilation system (values are in mm and plan is not to scale).

supplied heat and heat losses are balanced by using dynamic time-steps simulation, finite difference method and transient calculations. Heat losses depend on indoor temperature, outdoor temperature, speed and direction of wind, thermal properties of building envelopes, type of ventilation system as well as tightness and orientation of buildings. Supply heat depends on active and passive heating, i.e. internal heat (released by people based on activity level, by lightening, and by equipment) and external heat (released by solar radiation through windows). Also, the storage and emission of heat in the structure of the building, which is important for the power demand calculation, is accurately calculated in IDA ICE 4. The major drawback of using IDA ICE 4 is the risk of unexpected program crashes and also errors in creating the mathematical model during the simulation.

Table 1
U value for building envelope.

Structure	Roof	Floor	Exterior wall	Windows	Doors	U_m
U value, W m^{-2} K^{-1}	0.13	0.15	0.15	1.10	1.50	0.28
Area, m^2	65.0	58.0	116.0	20.2	4.0	

3.1.1. Model description

Input data including floor plan, dimension of different parts of the building and HVAC system were provided by NCC, one of leading construction and property development companies in the Nordic region. The location and size of rooms, windows and doors in the model corresponded to the real situation (Fig. 5). For more accurate results of energy consumption in simulation, each dwelling was divided into 12 zones according to usage of the rooms. For each zone a different exhaust and supply air flow rate was defined. The ground was modeled according to ISO-13370 [22] for determining the heat transfer between building and ground. In ISO-13370 a 0.5 m layer of earth and a 0.1 m layer of insulation were assumed beneath the ground level floor of the buildings [23], and together with a constant ground temperature of 9 °C. Also, in simulation the maximum time step was set to 1.5 h and the tolerance level to 0.02. In simulation, cooling energy was not considered since there was no cooling device supplied. All dwellings were modeled with different orientation and different internal heat gain. The shading by roof overhang and neighboring buildings was considered as well. A mechanical exhaust air system with opening on external wall as supply device was defined for ventilation.

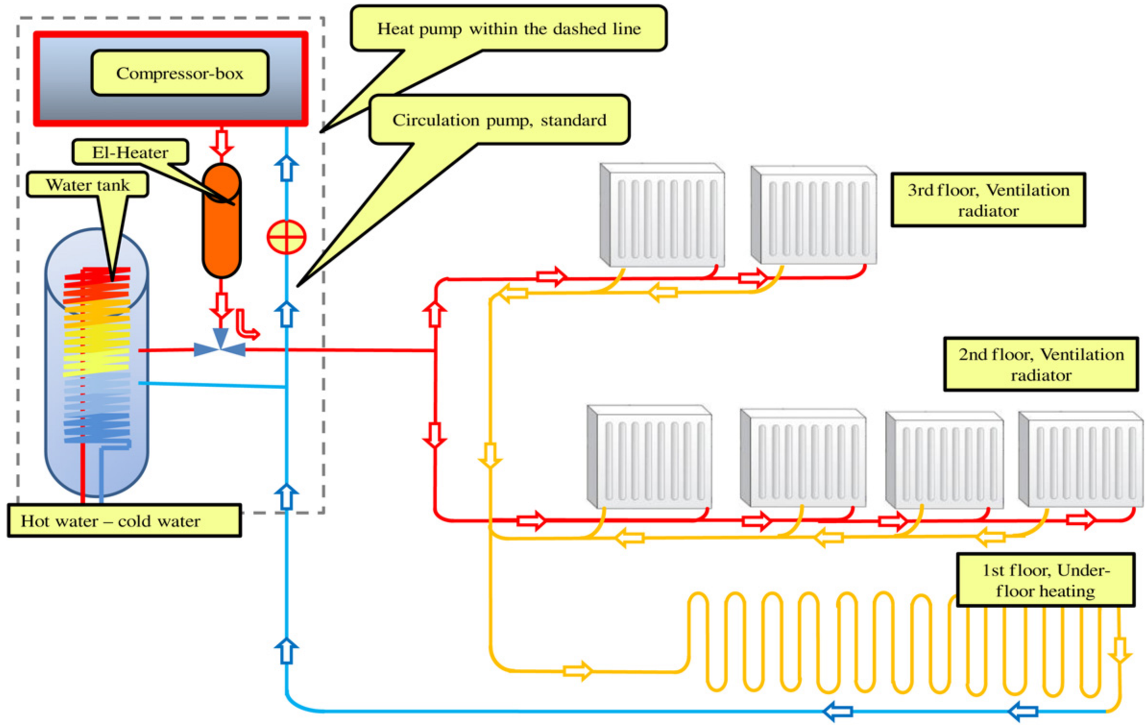

Fig. 4. Low temperature hydronic heating system connected to the heat pump.

Heating system was modeled as UFH for the first floor and radiator for the second and third floor. When modeling the radiators in IDA ICE 4, the maximum heat output as well as the supply and return temperature to the radiator were set equal to the corresponding values of the ventilation radiator. The maximum power of ventilation radiator was achieved by an Excel-based program called AIR-Simulator. The heat output given by this program is validated experimentally for the different applications used.

The heat pump was modeled as boiler with an average constant electric COP of 2.7 over the year, since it was not possible to set up a variable COP function for a heat pump in IDA ICE 4. This assumption seems reasonable since the temperature of heat source (room temperature) or evaporation temperature is constant over the year in an exhaust air heat pump.

The value 2.7 was obtained from Ref. [13] giving a mean COP for exhaust air heat pump installations in Sweden with hot water temperature of 50 °C. In IDA ICE 4 the boiler consumes energy, e.g. gas or electricity, and produces warm water with given temperature.

To predict indirect/passive heating from people, occupants were asked to report on the number of persons living in each dwelling as well as the number of hours that they spend inside during weekdays and weekend. Responses revealed that on average, occupants spent 14 h·day^{-1} on weekdays (Fig. 6) and 20 h·day^{-1} at weekends inside the house. The contribution of each person to internal heat gain depends on the activity level, e.g. around 135 W heat generation in sedentary activity in living room, 150 W heat when eating in kitchen, and in average 85 W in bedrooms. In addition, to predict the contribution of lights and equipment to space heating it was assumed that the lights were on when living room and kitchen were occupied, and some equipment was running full-time in kitchen and part-time in the living room. Of the total energy consumption 760 W, 70% contributed to indirect heating [12]. DHW consumption was estimated from heat pump electricity consumption during

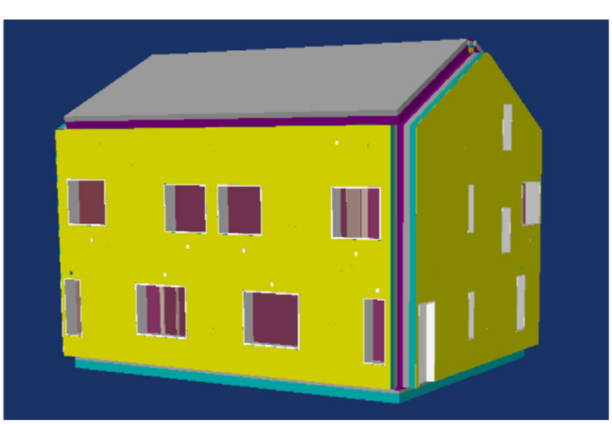

Fig. 5. Simulation of a pair of semi-detached houses using IDA ICE 4 software.

Fig. 6. Profile for the presence of the occupants on weekdays; 1 for fully occupied and 0 for absence.

Table 2
Measurement results and mean outdoor temperature in Stockholm (SMHI 2011–2012).

Mean temperature, °C	Jan, −2.6	Feb, −5.2	Mar, 2.9	Apr, 3.7	May, 10.6	Jun, 12.3	Jul, 16.4	Aug, 15.2	Sep, 10.9	Oct, 7.5	Nov, 2.4	Dec, 1.6	Total, kWh m^{-2}
Energy consumption, kWh													
27	1339	1233	962	638	535	370	257	328	425	582	809	1231	54
30	1303	1059	781	498	324	267	276	315	341	570	891	1146	49
31	1367	973	763	621	431	291	232	278	420	648	996	1208	51
32	1308	906	731	594	409	286	248	237	339	564	886	1130	48
33	1478	1127	716	453	468	352	260	281	407	657	1162	1320	54

summer months, since the only reason that the heat pump provides heating is because of DHW consumption in this period. A natural variation in DHW consumption between buildings was observed, probably depending on different living habits.

3.2. Site measurements

For site measurements occupants of the five houses were asked to read and report heat pump electricity consumption each month. This consumption was used for space heating, hot water and ventilation. The buildings were occupied on 14th December 2011. The starting point for the measurements was thus 14th December 2011, and the last reported value for this study was on 13th December 2012.

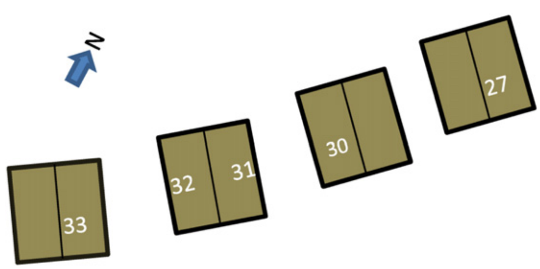

Fig. 7. Orientation of the five dwellings studied.

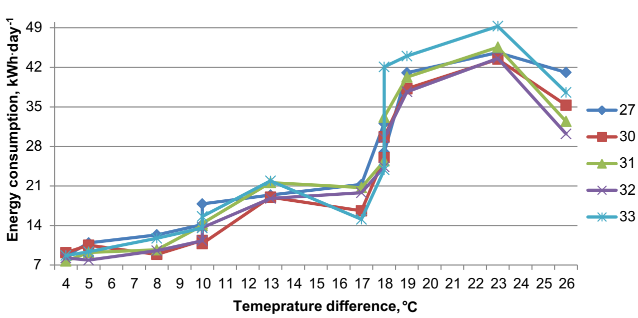

Fig. 8. Heating load as a function of temperature difference between inside and outside for the five dwellings.

3.3. Thermal comfort analysis

The buildings studied have a high level of airtightness and are well-insulated, so there is a high potential for achieving acceptable indoor climate [24]. IDA ICE software uses the ISO 7730 [25] assessment method to determine the comfort level achieved. The thermal comfort supplied by the model took account of the PMV related PPD value [26] and mean radiant and operative temperature at the given average wind velocity. Thermal comfort analyses and PPD calculations in IDA ICE are based on Fanger's models [27] and a simple model for linear vertical temperature gradient is used [28].

More detailed analyzes of thermal comfort with ventilation radiator is given in reference [11]. In this work, Myhren and Holmberg analyzed both ventilation and conventional radiators regarding their heat output and thermal comfort using CFD modeling. Simulation showed that ventilation radiator gives a more favorable, i.e. stable thermal climate. Also, their results of CFD modeling showed that the risk of cold draught was reduced when using ventilation radiator. In current study, to survey the thermal comfort in reality a questionnaire was distributed to occupants. In the questionnaire they were asked how satisfied they were with the temperature and air quality in different rooms and if they feel any discomfort with respect to draught.

4. Results

4.1. Measurements result (December 2011–December 2012)

Measurements included electricity consumed by heat pump with integrated fan used for space heating, domestic hot water and ventilation. The annual total electricity used by the heat pump from December 2011 to December 2012 for different houses is presented in Table 2. As can be seen, although all dwellings are constructed the same, electricity consumption varied for different dwellings due to different internal loads, living habits, orientation of buildings and shading effects of surrounding buildings. Fig. 8 shows the space heating load for the houses as a function of temperature difference between inside (21.0 °C), and mean outside temperature shown in Table 2, In February the temperature difference (26.2 °C) was higher than in January (23.6 °C), but the heating demand in February was less than in January, as shown in Fig. 8. A higher intensity of solar radiation in February is a likely reason for this, but

Table 3
Simulation results of energy consumption by IDA ICE 4 for different rooms and equipment from December 2011 to December 2012.

	Total, kWh m^{-2}	Living room (UFH)		Kitchen (UFH)		Bedrooms (VR)		Bathroom, WC (UFH)		Third floor (VR)		Hot water		AHU	
		kWh	%	kWh	%	kWh	%	kWh	%	kWh	%	kWh	%	kWh	%
27	55	1605	18	144	2	1683	19	287	3	1104	13	3520	40	480	5
30	53	1483	17	161	2	1686	20	305	4	1097	13	3360	39	480	6
31	52	1494	18	133	2	1588	19	252	3	1141	14	3200	39	480	6
32	51	1634	20	133	2	1659	20	251	3	1102	13	3040	37	480	6
33	55	1660	19	162	2	1717	19	252	3	1167	13	3520	39	480	5

Table 4
Details of annual mean passive heat and heat losses for different dwellings between December 2011 and December 2012, according to the IDA ICE 4 simulation.

	27	30	31	32	33
No of persons	4	4	5	3	3
Mean transmission loss, W	−804	−852	−849	−822	−814
Mean ventilation loss, W	−1174	−1323	−1283	−1298	−1193
Mean internal heat gain from occupants, W	473	473	591	355	355
Mean internal heat gain from equipment, W	390	390	390	390	390
Mean internal heat gain from lights, W	143	143	143	143	143
Mean external heat gain from solar radiation, W	468	643	477	693	569
Passive heat gain/heat loss, %	75	76	75	75	73

also living habits and building orientation may influence. The average total solar radiation towards a vertical surface facing south in Stockholm [SMHI] is 25 kWh m^{-2} in January and 50 kWh m^{-2} in February.

4.2. Simulation result (December 2011–December 2012)

Energy use analyses carried out by IDA ICE 4 included the effect of heat capacity, infiltration, internal and external heat gain, weather data, indoor temperature and detail description of the construction type and building geometry. IDA ICE 4 simulation was run for five different houses taking account of their orientation (Fig. 7) and internal heat gain during 14th December 2011 till 13th December 2012 (one year). The results of total consumption and required space heating based on the usage of the room are given in Table 3. As can be seen, heating demand in the kitchen was much less than for other parts of the house, due to more internal heat gain from installations. Also, energy consumption for DHW depends on the living habit and it differed for each dwelling.

Table 4 illustrates a detailed expression of annual mean passive heat gain and ventilation and transmission losses for each dwelling. The internal heat gain from people was the same for houses 27–30, and for 32–33. These groups had the same number of occupants. However, in reality the influence of internal heat gain on energy usage might be more dependent on occupant behavior than on the

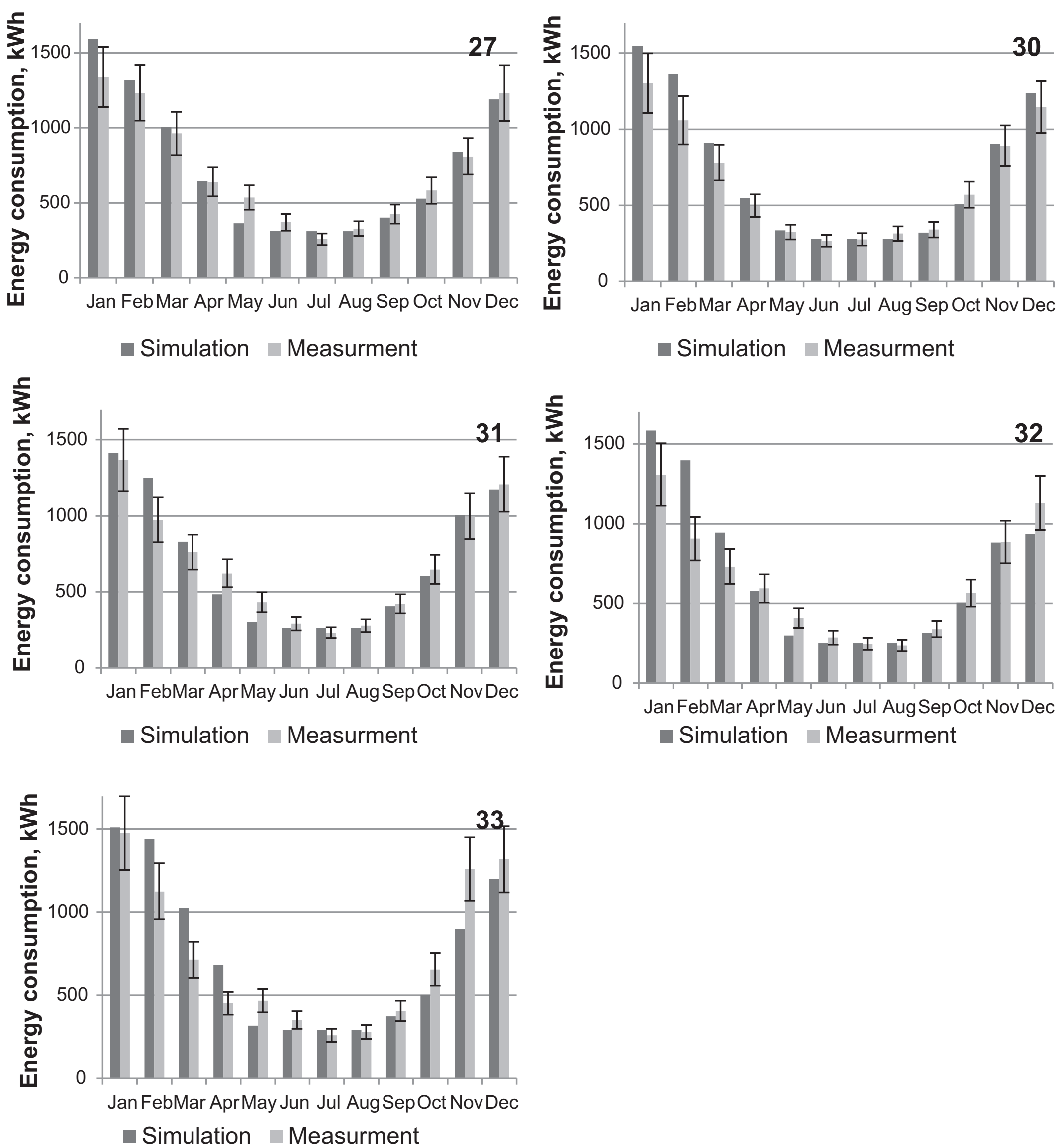

Fig. 9. Comparison of measured and calculated energy consumption in the five houses studied.

number of persons. As shown, the external heat gain by solar radiation through windows was higher in dwellings 30 and 32 due to more southward orientation of windows. Shading by surrounding buildings and having no windows toward south resulted in low solar contribution of heat in dwellings 27 and 31, lower than in the three other buildings. However, more persons were living in this dwelling, which meant the highest passive heat gain from people. Also, an equivalent number of lights and equipment in all dwellings gave a passive heat contribution of 532 W. The ratio of passive heat gain to heat losses in dwelling 30 was higher than in other dwellings, with dwelling 33 having the lowest value. This ratio shows how the total passive heating, i.e. number of occupants, solar gain due to orientation of building, etc. in different buildings influence the total heating demands.

4.3. Model validation: measurements vs. simulation

The IDA ICE model was experimentally validated by comparing the results of simulation with site measurements, see Fig. 9. The error bar in Fig. 9 is in a range of 15% deviation of measured value. There appeared to be a good agreement between simulation results and measurements, i.e. for most months the simulation results are in range of 15% deviation of measured value. However, the simulation sometimes overestimated/underestimated the heating demand. Some possible reasons for that could be slight difference in the ventilation radiator shape and different defined living habit in the simulation compared to reality. A possible explanation for the rather large deviation in February compared to other months can be explained by high solar radiation intensity; providing preheating of ventilation air and decreasing the space heating load, which may not be predicted in simulation. A comparison of total consumption in simulation and measurements (Fig. 10) shows that the total values are very close to each other, with a maximum deviation of 7% for house 30. Hence, with help of this validation it can be concluded that the simulation results are in good agreement with the real heating demand.

According to the Swedish building regulations [12], depending on location and heating source, all houses should meet energy guidelines for heating, ventilation, cooling and hot water. Table 5 shows the maximum allowable of energy requirement depending on climate zone. Sweden has three climate zones, north “1”, middle “2” and south “3”; Stockholm is located in climate zone 3.

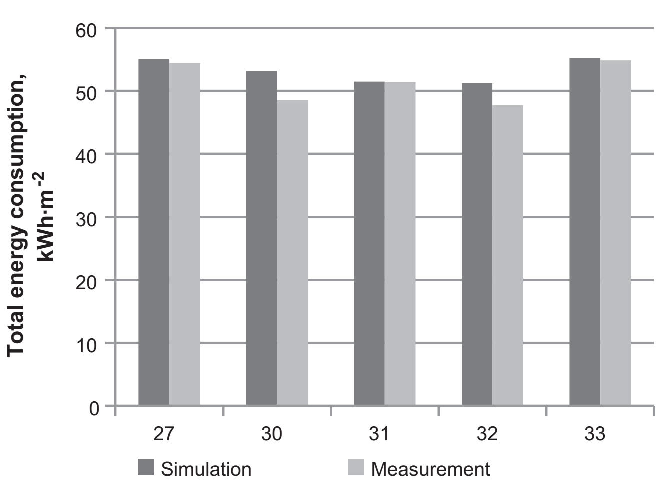

Fig. 10. Comparison of total consumption between measurement and simulation in the five houses studied.

Table 5
Swedish energy requirement [12].

Climate zone	1	2	3
Building heated by other means than electricity: total energy consumption, kWh m^{-2} per year	150	130	110
Building heated fully or partly by electricity: total energy consumption, kWh m^{-2} per year	95	75	55

Looking at the energy requirement of the dwellings in Table 5, all investigated buildings satisfy the maximum required energy consumption, i.e. 55 kWh m^{-2} per year.

4.4. Thermal comfort results

IDA ICE simulation showed that the PMV-based PPD is 12%. This is acceptable and thus lower than the (EN ISO 7730) standard regulation of 15%. The simulation also showed that the mean temperature level in all zones varied within 20 °C and 25 °C, which also fulfill standard regulations.

In the questionnaire some occupants reported cold draught on the first floor, where fresh air was directly brought into the house without preheating. This was not the case on the second and third floors, where ventilation radiators were installed. These reports can be explained by the slow reaction of UFH (large thermal mass flow of water) on the first floor, causing higher temperature fluctuations [29], so that occupants may feel cold before the system has reached stable conditions. Generally, all occupants were satisfied with the mean temperature in their dwelling.

5. Conclusion

The purpose of the study was to ascertain whether it is possible to have low temperature heating systems that meet energy requirements without compromising thermal comfort. Hence, dwellings equipped with LTHH systems were evaluated in terms of energy consumption and thermal comfort. The energy consumption for space heating, ventilation and hot water determined by IDA ICE simulation approximately corresponded to on-site measurements with small divergence. Looking at energy requirements and Swedish building regulations, the paper concludes that the dwellings equipped with LTHH system can meet limitation of energy consumption. Investigation of thermal comfort was also made, both by using IDA ICE software and by questionnaire. Simulations showed that the PMV based PPD was 12%, and mean temperature variation in all zones was in acceptable range. Also, the questionnaire showed that occupants felt more comfortable in floors equipped with ventilation radiators compared to the ground floor with under-floor heating. It should be noted, however, that this study was of five dwellings only; future study of other types and greater number of dwellings are needed before it is possible to generalize regarding all houses using low temperature heating systems.

Acknowledgement

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Original Article

Demand-controlled ventilation in new residential buildings: consequences on indoor air quality and energy savings

Arefeh Hesaraki* and Sture Holmberg
Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Stockholm, Sweden

Abstract:

The consequences on indoor air quality and potential of energy savings when using a variable air volume (VAV) ventilation system was studied in a newly-built Swedish building. Computer simulations with IDA Indoor Climate and Energy 4 (ICE) and analytical models were used to study the indoor air quality (IAQ) and energy savings when switching the ventilation flow from 0.375 $l \cdot s^{-1} \cdot m^{-2}$ to 0.100 $l \cdot s^{-1} \cdot m^{-2}$ during unoccupancy.

To investigate whether decreasing the ventilation rate to 0.1 $l \cdot s^{-1} \cdot m^{-2}$ during unoccupancy, based on Swedish building regulations, BBR, is acceptable and how long time the reduction can last for an acceptable IAQ, four strategies with different VAV durations were proposed. This study revealed that decreasing the flow rate to 0.1 $l \cdot s^{-1} \cdot m^{-2}$ for more than four hours in an unoccupied building creates unacceptable IAQ in terms of volatile organic compounds concentration. Hence, if the duration of unoccupancy in the building is more than four hours, it is recommended to increase the ventilation rate from 0.100 to 0.375 $l \cdot s^{-1} \cdot m^{-2}$ before the home is occupied.

The study showed that when the investigated building was vacant for 10 hours during weekdays, increasing the ventilation rate two hours before occupants arrive home (low ventilation rate for 8 hours) creates acceptable IAQ conditions. In this system the heating requirements for ventilation air and electricity consumption for the ventilation fan were decreased by 20 % and 30 %, respectively.

Keywords:
Controlled ventilation system, Energy performance, IDA ICE 4, Indoor air quality, Variable air volume

* Corresponding author: Tel.: +46 8 790 48 84; fax: +46 8 790 48 00.
E-mail address: arefeh.hesaraki@byv.kth.se
Address: Brinellvägen 23, 100 44 Stockholm, Sweden

Nomenclature

$\bar{\tau}$	mean air age, h
τ_n	nominal time constant for ventilation, h
τ_b	time constant for a building thermal inertia, h
ACH	air changes per hour
BBR	Swedish National Board of Housing, Building and Planning
c	pollutant concentration in the room, ppm
CAV	constant air volume
c_h	heat capacity, $J \cdot kg^{-1} \cdot °C^{-1}$
c_{inlet}	pollutant concentration in the inlet ventilation air, ppm
c_o	pollutant concentration in the room at start (t = 0), ppm
CO_2	carbon dioxide
f_ε	freshness efficiency, %
h	hour
IAQ	indoor air quality
IDA ICE	IDA Indoor Climate and Energy
LVR	low ventilation rate
$\dot{m}$	emission rate, $kg \cdot h^{-1}$
m	mass of the material, kg
n	number of air changes per hour, $1 \cdot h^{-1}$
P_{design}	design (maximum) heating power, W
q	outdoor airflow rate, $m^3 \cdot h^{-1}$
Q_{tot}	total specific heat loss including transmission, ventilation and leakage losses, $kW \cdot °C^{-1}$
RH	relative humidity
t	time, h
T	temperature in room, °C
T_o	temperature in the room at start (t = 0), °C
T_{indoor}	indoor temperature, °C
T_{DOT}	design outdoor temperature, °C
U	heat transfer coefficient, $W \cdot m^{-2} \cdot K^{-1}$
UFH	under-floor heating
U_m	mean U value, $W \cdot m^{-2} \cdot K^{-1}$
V	room volume, m^3
VAV	variable air volume
VOCs	volatile organic compounds

Introduction:

The main goal of a ventilation system is to create acceptable indoor air quality and thermal comfort taking into account health, comfort and productivity of inhabitants [1]. There are various types of ventilation systems, e.g. natural ventilation, mechanical ventilation or a combination of them, i.e. hybrid ventilation. Depending on many factors such as indoor air quality requirements, outdoor climate and cost, one type is preferred over the others. Ventilation airflow can be supplied to the room by constant air volume (CAV) or variable air volume (VAV). The VAV system is already dominant in industrial, commercial, school and office buildings [2], due to the different occupancy levels during a day and unpredictable fluctuations in CO_2 concentration. However, in residential buildings VAV is not a very common system.

It should be noted that a VAV system is only feasible if the pollution level and demand for ventilation air vary over time. Swedish building regulations, BBR [3], recommend a minimum ventilation rate of 0.35 $l \cdot s^{-1} \cdot m^{-2}$ of floor area in all occupied residential dwellings; corresponding to 0.5 ACH in a room with 2.5 m height. In BBR, it is also mentioned that when no one is at home, the ventilation rate can be decreased to 0.10 $l \cdot s^{-1} \cdot m^{-2}$. A base constant ventilation rate, i.e. 0.10 $l \cdot s^{-1} \cdot m^{-2}$, is required in order to take into account the constant pollutant sources, e.g. building materials and furniture.

In current residential buildings, people go to work or school from morning until afternoon. This gives opportunity for reducing the ventilation rate during non-occupancy hours. However, existing residential ventilation systems with constant ventilation rate are in conflict with ventilation on demand. This results in periods with wasting energy on heating unnecessary airflow. Hence, in order to achieve energy-efficient residential buildings the ventilation rate should be varied according to the demand, depending on occupancy level or pollutant emission.

VAV systems have become popular due to improved energy efficiency in comparison with CAV systems [4], and many studies [5,6,7,8,9] have focused on this aspect of VAV and demand-controlled ventilation systems. These studies based on site measurements, simulations and analytical models have shown that depending on the control system 5 - 60 % of initial energy demand could be saved in residential buildings.

There are many variables applicable for controlling the airflow rate in VAV systems, e.g. CO_2 concentration, relative humidity (RH), room temperature or occupancy detection. A CO_2 indicator is only appropriate with unpredictable variations in occupancy, and when the occupancy is high [7], e.g. in schools or commercial buildings. In addition, RH depends on the temperature and absorption or desorption of moisture by building materials, and it may cause erroneous indication of ventilation rate based on the inhabitants' demand [7]. In buildings with stable periods of occupancy, time control or occupancy detection could be appropriate control variables.

To control airflow rate in a mechanical ventilation system, ventilation fan speed can be adjusted in several ways based on occupancy detection. The simplest way would be manual control; this method requires a manual switch to adjust the ventilation fan speed, and responsibility is left to the occupants. Most likely, the occupants prefer this and feel more comfortable when they themselves have control of their own environment. Another method could be controlling the ventilation fan speed with a timer, i.e. clock control. With clock control the occupancy period is easily predictable, and the speed of ventilation fan can be adjusted accordingly. Another system could be a link to an infra-red sensor control system within the house to detect the presence of people. However, in all these control systems, the indoor air quality (IAQ) in terms of indoor pollutant gases such as CO_2 or volatile organic compounds (VOCs) should be considered.

In Sweden it is allowed to decrease the ventilation rate in an unoccupied building to 0.1 $l \cdot s^{-1} \cdot m^{-2}$, but there is still the question of how long the ventilation rate can remain at the decreased level taking into account the IAQ and thermal comfort.

To answer this question a new-built Swedish building with a CAV ventilation system was chosen. Then the ventilation rate was modeled to decrease from 0.375 $l \cdot s^{-1} \cdot m^{-2}$ to 0.100 $l \cdot s^{-1} \cdot m^{-2}$ during unoccupancy. Finally, the influence of this VAV concept on IAQ in terms of mean air age, CO_2 and VOC concentrations as well as the potential of energy savings were evaluated. The authors motivated a VAV system in the building by the fact that occupants spent approximately 14 hours of 24 hours per weekday and 20 hours per weekend-day in the building [10].

Heating requirements for CAV ventilation in the investigated building accounted for approximately 60 % of total space and ventilation heating demand. So, by implementing a VAV system there is a potential for saving energy by decreasing this demand. In this study, the controlled airflow rate was based on occupancy detection. The ventilation fan speed thus increased when home was occupied and decreased when the building was empty.

Description of the building

The investigated building was 160 m^2 with three stories including kitchen, bathroom, living room and bedrooms. A mechanical CAV exhaust ventilation system extracted in total 60 $l \cdot s^{-1}$ from exhaust devices in the kitchen (10 $l \cdot s^{-1}$), wardrobe (6 $l \cdot s^{-1}$), bathroom and toilet (26 $l \cdot s^{-1}$), and third floor (18 $l \cdot s^{-1}$). This corresponds to 0.375 $l \cdot s^{-1} \cdot m^{-2}$ of floor area and fulfilled BBR requirements [3]. A simplified drawing of the dwelling illustrating the schematic of airflow and heating systems on the different floors is shown in Fig. 1. Exhausted air was replaced by fresh outdoor air entering through inlets in bedrooms and living room due to constant negative pressure in the building. Supply devices were placed on the external walls above windows in the living room, kitchen, and hallway on the first floor, and behind radiators in bedrooms on the upper floors. This kind of radiators is called “ventilation radiator” since the vent supply is located behind the radiator and is connected to it for preheating of incoming air [11]. Table 1 shows the area and airflow rate in each zone.

The investigated building was a newly-built Swedish house with low transmission loss through construction due to a high level of air tightness and very low mean U value, i.e. U_m was 0.28 $W \cdot m^{-2} \cdot K^{-1}$. This value was lower than the limited U_m assigned by BBR, i.e. 0.40 $W \cdot m^{-2} \cdot K^{-1}$ [3].

The critical factor with respect to the efficiency of the VAV system and energy savings is the airtightness of the building. A high infiltration rate will increase the uncontrolled air volume causing need for additional heating. A blower door test showed that the airtightness of the investigated building was 0.63 $l \cdot s^{-1} \cdot m^{-2}$ (external surface) under 50 Pa pressure difference between indoor and outdoor.

Table 1. Floor area and supply/exhaust airflow rates in different zones of the building; the average supply/exhaust flow rate is seen in the last column

Zone	Living room	Kitchen	Hall-Way	Bed-room 1	Bed-room 2	Bed-room 3	Bed-room 4	Toilet	Bath-room	Ward-robe	Pass	Third floor	Total
Area, m^2	24.0	13.1	6.8	13.1	8.0	8.2	12.1	6.1	4.4	2.9	7.3	54.0	160.0
Supply, $l{\cdot}s^{-1}{\cdot}m^{-2}$	0.6	0.3	0.6	0.5	0.5	0.5	0.5	0.0	0.0	0.0	0.0	0.3	0.375
Exhaust, $l{\cdot}s^{-1}{\cdot}m^{-2}$	0.0	0.8	0.0	0.0	0.0	0.0	0.0	1.8	3.4	2.1	0.0	0.3	0.375

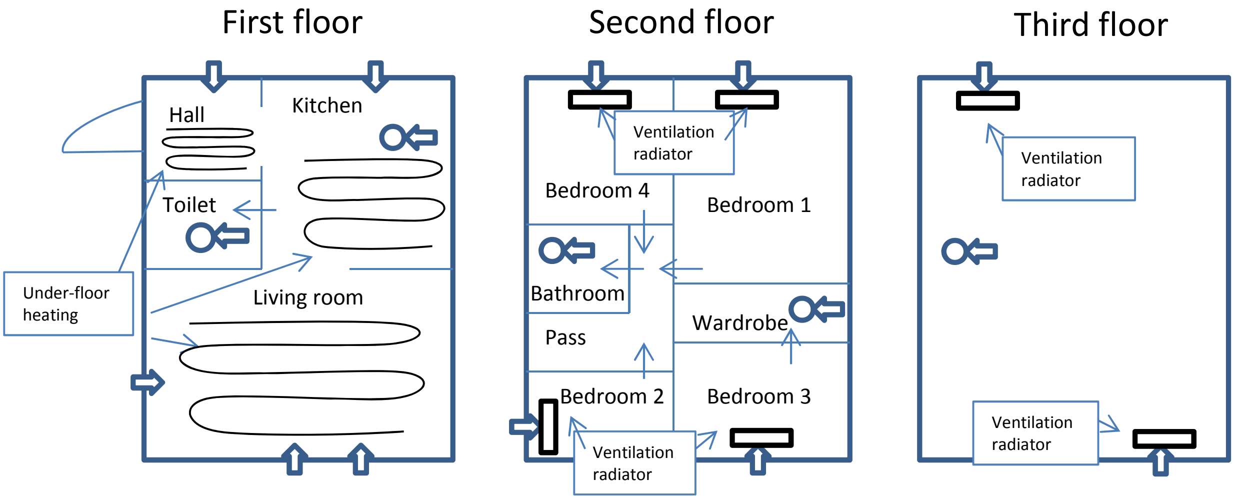

Fig. 1. Building plan showing the schematic of airflow and heating systems

Method

The main evaluation tool of this study was the IDA Indoor Climate and Energy (ICE) 4 simulation program [12]. IDA ICE 4 is a dynamic program used to predict the energy consumption, thermal comfort and indoor air quality based on building properties defined by the user [13]. In IDA ICE 4 a mathematical model is created to calculate the heating and cooling load in a building by simultaneous dynamic simulation of heat transfer and airflow. The results of IDA ICE 4 have been validated in several studies [14,15,16,17]. In IDA ICE 4, it is possible to define different airflow rates in supply and exhaust parts for different zones in the form of a VAV or CAV system. In addition to the clock-controlled fan speed and pressure, the VAV model can be controlled by humidity, CO_2 and temperature for each zone.

The investigated building was modeled in the same geometry as the actual building with a CAV system. In addition, with the purpose of investigating the indoor air quality and the potential for energy savings, the building was modeled with a VAV system as well. The building modeled was divided into 12 zones considering different rooms with different ventilation exhaust and supply flow rates. The ventilation system was modeled as a mechanical exhaust system. To define the supply air, the opening was considered on the external walls in bedrooms, living room, kitchen, hallway and third floor, and the exhaust devices were modeled in the kitchen, bathroom, toilet, wardrobe and third floor.

In the modeled VAV system the ventilation rate was switched between two flow rates, i.e. normal requirements (per floor area) of 0.375 $l{\cdot}s^{-1}{\cdot}m^{-2}$ to minimum requirements of 0.100 $l{\cdot}s^{-1}{\cdot}m^{-2}$ during unoccupancy. This strategy, based on the presence of occupants, was controlled by variable ventilation fan speed. The ventilation fan speed and as a result the ventilation flow rate was decreased to 27 % of the normal speed when reduced ventilation rate was needed, see Fig. 2. The value of 27 % shows the ratio of 0.100 to 0.375 $l \cdot s^{-1} \cdot m^{-2}$.

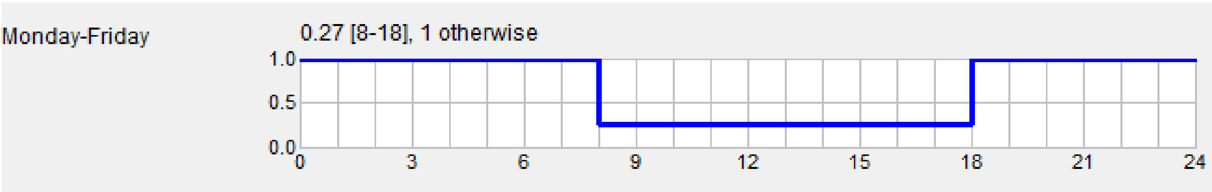

Fig. 2. Profile for the ventilation fan speed for weekdays; 1 for working in full power during occupancy (18:00-8:00) and 0.27 for working with 27 % of full speed during unoccupancy (8:00-18:00)

To investigate how long the reduced ventilation rate is acceptable in an unoccupied building, four different VAV duration periods were proposed. In the first system the ventilation rate was decreased during the period of unoccupancy from 8:00 until 18:00 (low ventilation rate for 10 hours, LVR10h). In the second scenario the ventilation rate was back to normal requirements two hours before occupants entered home (low ventilation rate for 8 hours from 8:00 until 16:00, LVR8h). The third strategy provided low ventilation rate for six hours from 8:00 until 14:00 (LVR6h), and the last VAV duration was for four hours from 8:00 until 12:00 (LVR4h), see Table 2. To study the relation between ventilation rates, indoor air quality and energy savings, each VAV control was modeled separately in IDA ICE 4.

Table 2. Start and stop time and mean air change for each proposed VAV control

	LVR4h	LVR6h	LVR8h	LVR10h
Start time	8:00	8:00	8:00	8:00
Stop time	12:00	14:00	16:00	18:00
Mean air change averaged over a day, h^{-1}	0.48	0.45	0.42	0.38

Also, to consider the IAQ when using the VAV system in terms of concentrations of indoor air pollutant gases, e.g. CO_2 and VOCs, an analytical model for all systems was applied. The aim was to investigate whether the indoor air quality with decreased ventilation rate is acceptable when occupants arrive home. In addition, the thermal inertia of the building and its influence on the design (maximum) heating power was calculated.

The building had two room-heating systems including ventilation radiator and under-floor heating. In order to investigate whether the type of heating system has any influence on energy savings with VAV control, two cases were considered. First, the building was modeled with under-floor heating in all three floors with both CAV and VAV systems. Then, all heating systems were changed to ventilation radiators on all floors, and simulations were run for both the CAV and VAV systems.

Results

The results of this study are presented in two parts, the first part deals with the indoor air quality when decreasing the ventilation air, and the second part shows results of energy savings.

Indoor air quality

In this study the indoor air quality was investigated in terms of mean air age, concentration decay and pollutant concentration level (CO_2 and VOCs) for different mixed ventilation rates. The main focus was on the time when the occupants arrive home, at 18:00.

Mean air age and theoretical freshness efficiency. To evaluate the VAV system with different strategies, mean air age as an indicator of the freshness of indoor air was considered at 18:00. In IDA ICE 4 mean air age is a measure of how long an average air molecule has spent in the building based on the ideal (full) mixing concept. If a zone is ventilated only by outside air and is at steady state, this number is equal to the nominal time constant τ_n, see Eq. (1). The mean air age is an important parameter when assessing indoor air quality in terms of freshness of air. The higher the age of the air, the more stale or stuffy air we thus have. Fig. 3 shows how the mean air age varies during a weekday for different modeled duration periods of VAV. As can be seen, the highest mean air age is 3.7 h at 18:00 when using LVR10h.

$$\tau_n = \frac{V}{q} \qquad (1)$$

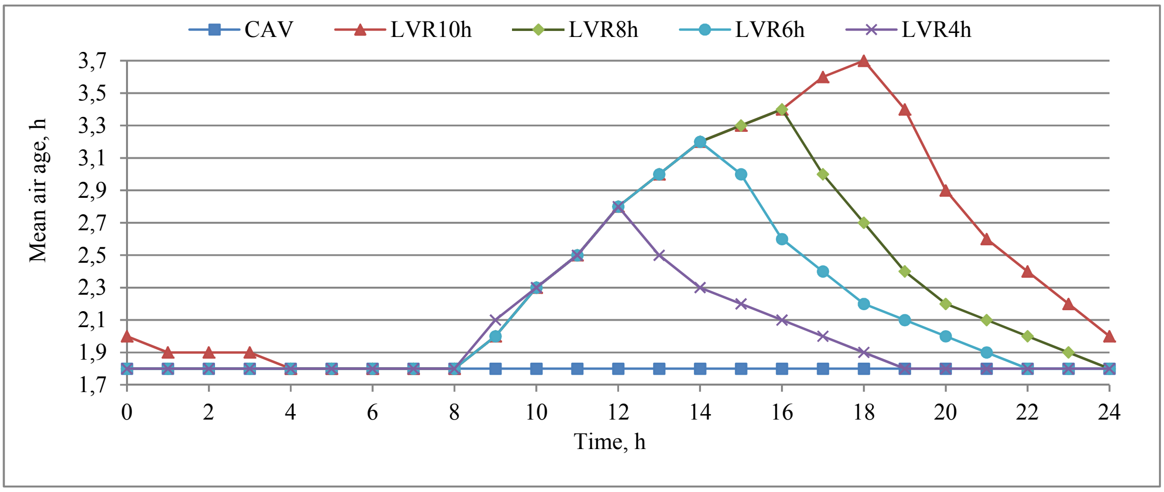

Fig. 3. Variation of mean air age during a weekday as function of different proposed VAV periods

The freshness efficiency (f_ε) was defined by the authors as a measure of the performance of the VAV system with respect to the mean air age in the room. The theoretical freshness efficiency represents the relation between nominal time constant (τ_n) and the existing mean age of air ($\bar{\tau}$) when occupants arrive home. It is called theoretical due to considering full mixing and ideal flow. Theoretical freshness efficiency can thus be represented by Eq. (2).

$$f_\varepsilon = \frac{\tau_n}{\bar{\tau}} \cdot 100\ \% \qquad (2)$$

For a perfect mixing CAV ventilation system, the theoretical freshness efficiency obtains its maximum possible value of 100 %. Table 3 shows the mean air age and theoretical freshness efficiency for different modeled systems.

Table 3. Theoretical freshness efficiency and mean air age for different VAV rates

	CAV	LVR4h	LVR6h	LVR8h	LVR10h
Ventilation rate at 18:00, h^{-1}	0.55	0.55	0.55	0.55	0.55
Nominal time constant (τ_n), h	1.80	1.80	1.80	1.80	1.80
Mean air age at 18:00 ($\bar{\tau}$), h	1.80	1.90	2.20	2.70	3.70
Theoretical freshness efficiency (f_ε), %	100.0	94.7	82.8	66.7	48.7

As can be seen, the high mean air age (more twice the nominal time constant) and the low theoretical freshness efficiency are a result of long duration with low ventilation rate, i.e. LVR10h. Hence, in order to have better indoor air quality in terms of mean air age and freshness when occupants arrive home, it is recommended to implement LVR8h, LVR6h or LVR4h.

Concentration decay. The ventilation efficiency can also be measured by concentration decay. For perfect mixing 63 % of the pollutants have left the room after a period of τ_n (1.8 h) from the initial concentration, and after $4\tau_n$ (7.2 h) this value is increased to 98 %. Eq. (3a, 3b) shows how the concentration level c in a ventilation system with time constant τ_n is related to the real time t.

$$-\frac{dc}{dt} = \frac{1}{\tau_n} \cdot c \qquad (3a)$$

$$c = c_0 \cdot e^{-t/\tau_n} \qquad (3b)$$

The theoretical concentration decay for different mixed airflow rates based on Eq. (3) is shown in Fig. 4. An exposure impulse was released at 11:00, i.e. seven hours before occupants arrive home. As can be seen, after $4\tau_n$, i.e. 7 h, the concentration level for LVR10h is still 15 % of the initial value; however, for the CAV system only 2 % of the initial value has been left inside the room.

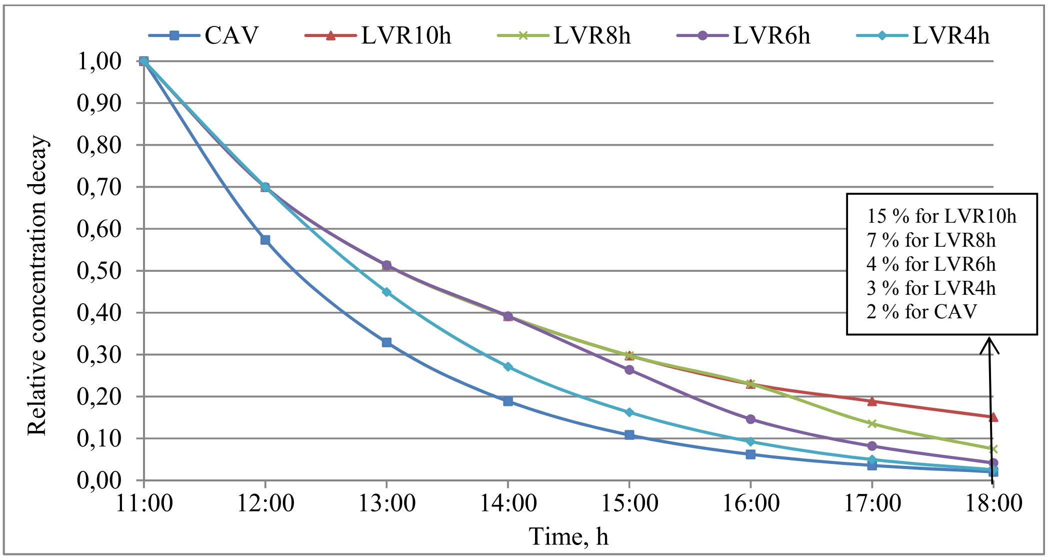

Fig. 4. Relative concentration decay for different VAV durations showing the concentration levels during seven hours before 18:00, i.e. when occupants arrive home

VOC concentration. The average VOC concentration in dwellings is about 0.1 $mg \cdot m^{-3}$ [9] corresponding to a generation of 20 $mg \cdot h^{-1}$. The maximum allowable VOC concentration in indoor air is 0.1 ppm [9].

Eq. (4) shows how the VOC concentration from building materials varies as function of number of air changes per hour and emission rates.

$$c = c_o \cdot e^{-nt} + \frac{\dot{m}}{q}(1 - e^{-nt}) \quad (4)$$

Fig. 5 shows the concentration level of VOCs from building materials when using LVR10h and LVR8h. As can be seen, VOC concentration increases when decreasing ventilation rate and after four hours of reduction in ventilation rate, at 12:00, the concentration starts to exceed the maximum allowed value, i.e. 0.1 ppm. Then, the concentration starts to decrease from 0.14 ppm at 18:00 due to increased ventilation rate. However, the recommended allowable VOC concentration in a room is 0.10 ppm, and it takes approximately one hour to reach to this acceptable level after increasing the ventilation rate. This time is shown with red pattern in Fig. 5. Hence, to avoid this inconvenient time, and to provide a better IAQ when occupants arrive home at 18:00, it is better to return back to the normal flow rate before occupants arrive home, i.e. use LVR8h.

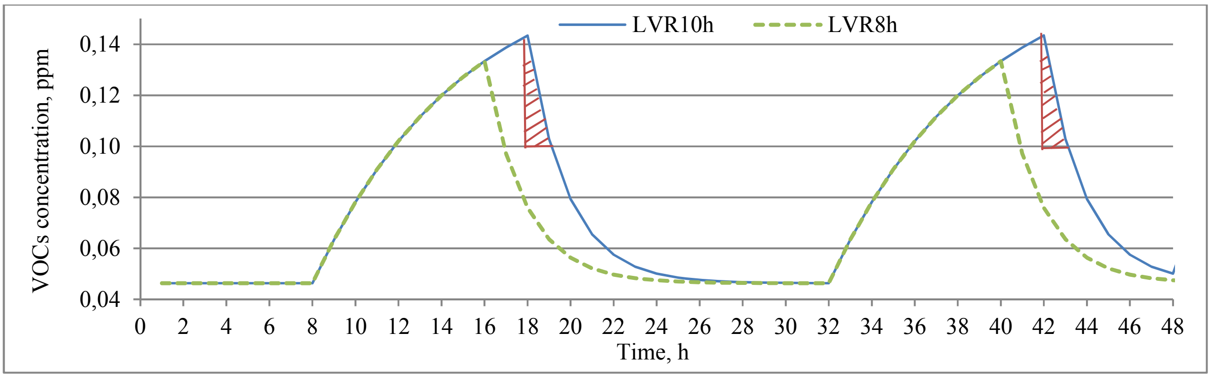

Fig. 5. VOC concentration in a room during two subsequent days for LVR8h and LVR10h showing the influence of flow rate reduction during unoccupancy on VOC concentration; the periods for LVR10h with greater concentration than allowable when occupants are home are marked in red.

CO_2 concentration. In addition to VOCs from building materials, CO_2 concentration was considered in order to evaluate the IAQ for a VAV system. With a CO_2 generation of 0.03 kg·h^{-1} per person in low to normal activity [18], and 400 ppm concentration in outdoor air, the variation of CO_2 was calculated using Eq. (5).

$$c = c_o \cdot e^{-nt} + (c_{inlet} + \frac{\dot{m}}{q})(1 - e^{-nt}) \quad (5)$$

The results for the LVR8h and LVR10h systems are shown in Fig. 6. As can be seen, the level of CO_2 concentration never exceeds the maximum allowable value, i.e. 1000 ppm. The reason is the lack of main pollutant source, i.e. humans in the room, when the ventilation rate was decreased in the middle of the day.

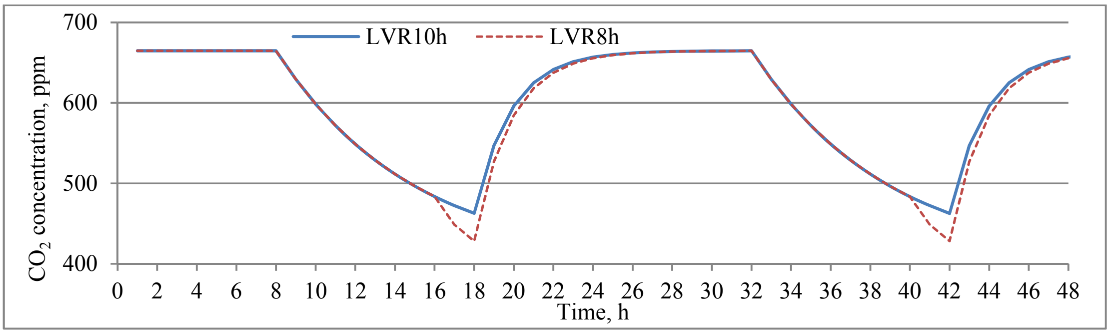

Fig. 6. Variation of CO_2 concentration in a room for two subsequent days based on the presence of occupants and ventilation rate for LVR10h and LVR8h

Energy savings with demand control

Heating power and energy savings with VAV were investigated in terms of the influence on heating requirements for ventilation and space heating.

Influence on heating requirements. In the *indoor air quality* part, it was revealed that LVR8h provided better conditions for indoor air quality than LVR10h. So, the following results are mainly presented for the LVR8h system.

IDA ICE 4 simulation showed that LVR8h (8 hours of reduced ventilation flow) decreased the heating required for ventilation and electricity for the ventilation fan up to 20 % and 30 %, respectively. Total monthly heating demand including ventilation and space heating for the CAV, LVR8h and LVR10h systems are shown in Fig. 7. Simulations showed that by using LVR8h, the annual heating demand decreased by 14 % from 4624 kWh to 4022 kWh. The results of total energy consumption and savings are presented in Table 4 separately for LVR8h, LVR10h and CAV. As can be seen, the total energy consumption including total heating demand, domestic hot water, and ventilation fan was reduced by 10 % from 52 $kWh \cdot m^{-2}$ to 47 $kWh \cdot m^{-2}$ for the LVR8h system compared to CAV.

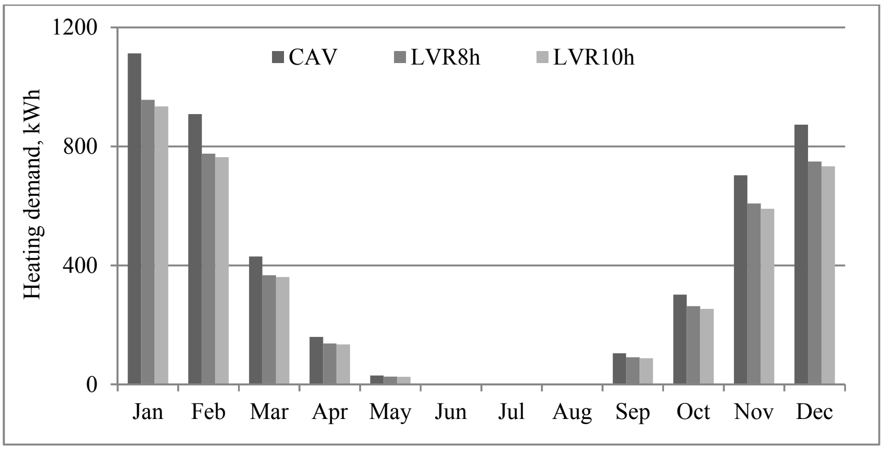

Fig.7. Monthly heating demand with CAV, LVR8h and LVR10h

Table 4. Energy consumption and savings with LVR10h, LVR8h and CAV systems

	CAV	LVR8h (saving in %)	LVR10h, (saving in %)
Heating requirements for ventilation, kWh	2651	2132 (20)	2041 (23)
Heating demand for ventilation + space heating, kWh	4624	3969 (14)	3885 (16)
Ventilation fan power, kWh	473	328 (30)	292 (38)
Domestic hot water, kWh	3200	3200 (-)	3200 (-)
Total energy consumption, $kWh \cdot m^{-2}$	52	47 (10)	46 (12)

Influence on design power. When decreasing the ventilation rate, the total specific heat loss (Q_{tot}) decreased, and as a result the time constant related to the building thermal inertia increased. The time constant represents a simplified quantification of the building's thermal inertia, which is the ratio of available heat capacity in the building ($m \cdot c_h$) to the total specific heat loss through the building, see Eq. (6). The total heat capacity depends on the individual material characteristics, mass (m_i) and individual heat capacities (c_{h_i}). In a newly-built Swedish building the time constant is approximately 5 days [18].

$$\tau_b = \frac{\Sigma(m_i \cdot c_{h_i})}{Q_{tot}} \quad (6)$$

The longer the time constant in a building, the more thermally heavy the building, and as a result the building is less sensitive to outdoor temperature variations [19]. This will reduce the requirements on heat distribution system and power supply. The time constant shows how rapidly a temperature change occurs based on an exponential decay function, see Eq. (7). As indicated by Eq. (7), if the external temperature is changed step-wise, the internal temperature will decrease by 63 % compared to the initial temperature after one time constant, τ_b.

$$T = T_o \cdot e^{-t/\tau_b} \quad (7)$$

By increasing the time constant, the thermal inertia for both LVR10h and LVR8h caused the temperature to change more slowly than for the CAV system.

With the LVR8h system the time constant increased by 9.3 % compared to the CAV system. This enhancement not only influenced the indoor temperature but also the design outdoor temperature (T_{DOT}) and hence the design (maximum) heating power (P_{design}). In Stockholm the design outdoor temperature with τ_b = 5 days is -14.8 °C [18]. This value changed to -14.5 °C with increased time constant of LVR8h. Calculation based on Eq. (8) showed that the maximum heating power can be decreased by 10 % for LVR8h.

$$P_{design} = Q_{tot} \cdot (T_{indoor} - T_{DOT}) \quad (8)$$

The influence of heating system type for energy savings with a VAV system. In order to investigate the influence on energy savings of a VAV system the building was modeled with both CAV and VAV and different types of heating systems. In the first scenario it was assumed that the heating system consisted of ventilation radiators on all floors, and in the second scenario the building was modeled with under-floor heating everywhere. Results were later compared with a combination of heating, i.e. under-floor heating on first floor and ventilation radiator on the second and third floors. Simulation results are presented in Table 5. As can be seen, when using CAV the modeled annual energy consumption is lowest with ventilation radiators, i.e. 49 kWh·m^{-2}. After implementing the LVR8h system the energy requirements when using ventilation radiators was reduced to 46 kWh·m^{-2}, a saving of 6 %. Simulations showed that the energy demand (57 kWh·m^{-2}) in the building modeled with only under-floor heating and CAV is higher than the BBR [3] limited value (55 kWh·m^{-2} in climate zone 3 including Stockholm). The reason could be a longer heating season, higher heat loss through the floor, and slow thermal control when sudden temperature changes occur. However, after implementing the LVR8h system the building with UFH met the Swedish requirements. In the building modeled with under-floor heating and LVR10h system the energy savings were more significant, i.e. 14 %, see Table 5.

Table 5. Energy demand in building with different heating systems, i.e. ventilation radiator and under-floor heating with CAV, LVR8h and LVR10h systems

Heating system	CAV	LVR8h, (saving in %)	LVR10h, (saving in %)	BBR requirements in climate zone 3
UFH + Ventilation radiator, kWh·m^{-2}	52	47 (10)	46 (12)	55
Ventilation radiator, kWh·m^{-2}	49	46 (6)	45 (8)	55
UFH, kWh·m^{-2}	57	52 (9)	50 (14)	55

Discussion

Indoor air quality should be considered when energy is saved by decreasing ventilation rate, since poor IAQ may cause health problems [20]. In this paper three parameters, i.e. mean air age, VOCs and CO_2 as indicators of IAQ were studied, see Table 6. As shown, for LVR10h the VOCs are above acceptable level (0.1 ppm) and mean air age in this system is two times higher than the nominal time constant.

Table 6. Energy savings and indoor air quality for different lengths of VAV compared to CAV.

	CAV	LVR4h	LVR6h	LVR8h	LVR10h
Mean ventilation rate weighted over a day, h^{-1}	0.55	0.48	0.45	0.42	0.38
Relative ventilation rate (compared to 0.55 h^{-1})	1.00	0.87	0.82	0.76	0.70
Space and ventilation heating demand including ventilation fan power, kWh·year^{-1}	5097	4668	4491	4297	4177
Savings in space and ventilation heating including ventilation fan power, %	-	8.4	11.8	15.7	18.0
Mean air age at 18:00, h	1.8	1.9	2.2	2.7	3.7
VOC concentration at 18:00, ppm	0.046	0.048	0.055	0.076	0.143
Relative VOC concentration (compared to 0.046 ppm)	1.00	1.04	1.20	1.65	3.11
Increase in time constant (τ_b), %	-	4.1	6.7	9.3	10.3
Decrease in design (maximum) heating power, %	-	4.4	7.2	9.9	11.0

Results on how energy savings and VOC concentration, as an indicator of indoor air quality, change with the mean airflow rate are presented in Fig. 8. IAQ decreases with reduced ventilation rate and energy savings increase, an acceptable balance has to be found. The figure shows that decreasing the mean ventilation rate to 80 % of the CAV system (LVR4h and LVR6h) does not influence VOC concentration very much, i.e. concentration increases by 13 % of total variation and energy savings are 13 % of reference consumption. However, decreasing the ventilation rate over 80 % of CAV (LVR8h and LVR10h) gives a sudden increase in VOC concentration level and the energy saving is only a few percent higher (5 %). The variation of VOCs lower than 0.1 ppm (55 % of total, see Fig. 8) shows an acceptable range of VOC concentration [9]. This point corresponds to energy savings of 17 % and decreased ventilation by 73 % of the constant rate. Hence, by decreasing mean ventilation by 73 % (decreasing the ventilation rate for approximately 8 hours) not only is acceptable indoor air quality achieved but also an energy-efficient building concept. Hence, it should be noted that decreasing the ventilation rate to 0.1 $l \cdot s^{-1} \cdot m^{-2}$ has limited duration when indoor air quality is to be considered. The study showed that for every $l \cdot s^{-1}$ reduction in ventilation rate, between 0.5 - 1.4 % energy for ventilation heating and ventilation fan consumption can be saved.

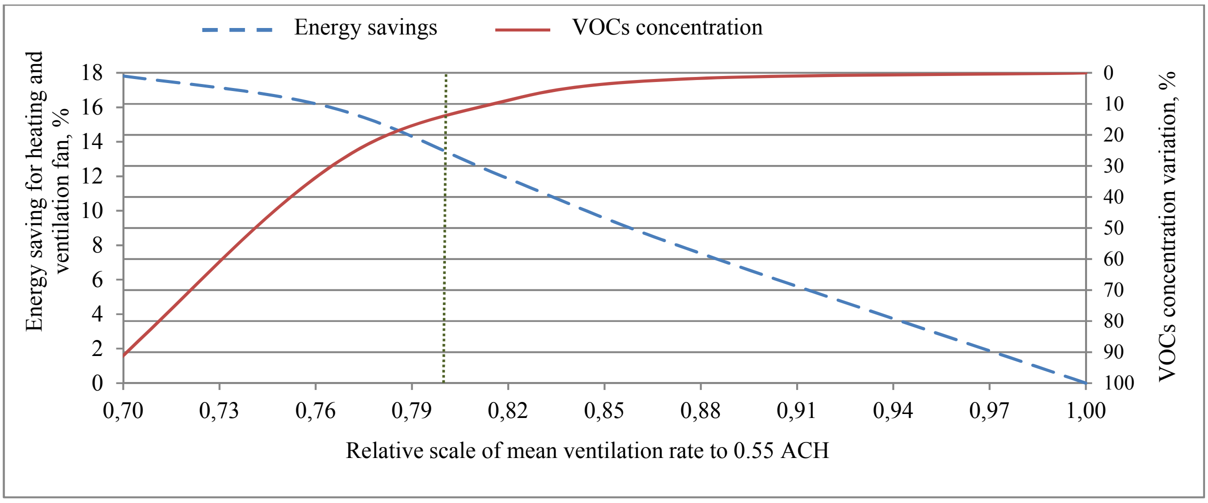

Fig. 8. Energy savings and VOC variation in percent of total concentration for different mean ventilation flow rates

Conclusion

The purpose of this study was to investigate consequences on IAQ and energy savings when ventilation supply air was controlled by demand. Swedish regulations allow a drop of ventilation rate from 0.35 to 0.10 $l \cdot s^{-1} \cdot m^{-2}$ during unoccupancy, but for how many hours this is allowable remains unclear. Four VAV durations were therefore proposed in a new Swedish dwelling with a CAV system. During weekdays the occupants leave home at 8:00 and return at 18:00. In the first strategy the ventilation rate was decreased during the whole unoccupancy (low ventilation for 10 h). In the second scenario the ventilation was returned to normal flow rate two hours before the home was occupied (low ventilation for 8 h). For the third and fourth cases the airflow was returned to normal flow rates four and six hours before occupants arrived home. The IDA ICE 4 simulation software and an analytical model were used to evaluate each system in terms of IAQ and energy savings.

The results of the study were divided into two parts: the first part dealt with indoor air quality when decreasing ventilation rate; and the second part showed energy savings with a VAV system. The indoor air quality investigation was based on four parameters:

1. Mean air age
2. Concentration decay
3. VOC concentration
4. CO_2 concentration

Results of the first part showed that when the reduced ventilation rate covered the entire time of unoccupancy, 10 hours, the VOC concentration was unacceptable, and the mean air age was high (3.7 h) when occupants arrived home. In order to take this inconvenience into account, it was suggested to increase the ventilation rate to normal requirements two hours before the home was occupied.

In the second part, energy savings were studied after decreased heating demand and increased thermal inertia of the building. Results showed that the total savings from ventilation fan and heating was 15 % when the airflow was reduced for 8 h during unoccupancy. In addition, as a result of increased thermal inertia of the building, the maximum (design) heating power was decreased by 10 % when lowering the ventilation rate for 8 h. Also, the influence of heating system type on energy savings with VAV was studied.

Energy-efficient ventilation considering IAQ is needed when the ventilation rate is adjusted to the occupants' demand. This study is a guideline to encourage the building sector to introduce VAV systems in residential buildings more often than before. The result of this study is in line with the European Union goal of reducing the consumption of energy in buildings by 20 % up to 2020.

Acknowledgement

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Topic 11. Computer tools and experimental techniques for assessment of building energy and built environments

Energy Performance Evaluation of New Residential Buildings with a Low-Temperature Heating System: Results from Site Measurements and Building Energy Simulations

Arefeh Hesaraki,* and Sture Holmberg

Division of Fluid and Climate Technology, School of Civil and Architectural Engineering, KTH Royal Institute of Technology, Handen-Stockholm, Sweden

**Corresponding email: arefeh.hesaraki@byv.kth.se*

Keywords: Building energy consumption, IDA ICE simulation, Measurements, Heating season

SUMMARY

The purpose of this study was to investigate the national energy requirements of a modern, newly built residential development including four semi-detached houses in Stockholm, Sweden. The apartments were equipped with heat pumps utilising exhaust heat, resulting in a hydronic heating system adapted to low supply temperature. Ventilation radiators as combined ventilation and heating systems were installed in the two upper floors. Efficient preheating of incoming ventilation air in the ventilation radiator was an expected advantage. Under-floor heating with traditional air supply above windows was used on the ground floor. Energy consumption was calculated by IDA ICE 4, a building energy simulation (BES) program. In addition site measurements were made for comparison and validation of simulation results. Total energy consumption was monitored in the indoor temperature controlled buildings during the heating season. Our results so far indicate that total energy requirements in the buildings can be met in a satisfactory manner.

INTRODUCTION

The building sector consumes a considerable part of the total energy consumption, i.e. approximately one third of the world's energy consumption (Federal Research and Development 2008). Population growth, increasing demands for building services, higher comfort requirements and increasing indoor activities are reasons that contribute to a raising energy demand in the future (Pe´rez-Lombard et al. 2008). However, through improvements in design, operation and renovation technology the building sector has good opportunities to reduce energy consumption. Sweden's national goal is to reduce the consumption of energy in buildings by 20% up to 2020, and by 50% up to 2050, in comparison with the levels for 1995 (Molin et al. 2011). Energy efficiency in buildings can be reached by minimizing transmission and ventilation heat losses, i.e. by using high insulation building envelope with good air tightness or by using energy saving equipment like heat pumps or heat recovery ventilation (Thullner 2010). By studying the diagram of energy consumption in a north European household, Figure 1, it is obvious that heating (73%) is consuming the main part of the total energy consumption (Anisimova 2011). Therefore, an energy efficient heating system has a high potential for energy savings. One of these potentials is using low temperature heating systems. Table 1 shows a classification schedule for hydronic heating system based on the supply water temperature.

Table 1. Water temperature ranges for heat emitters (Myhren and Holmberg 2007)

Heating system	Supply flow, ºC	Return flow, ºC
High temperature (HT)	90	70
Medium temperature (MT)	55	35-45
Low temperature (LT)	45	25-35
Very low temperature (VLT)	35	25

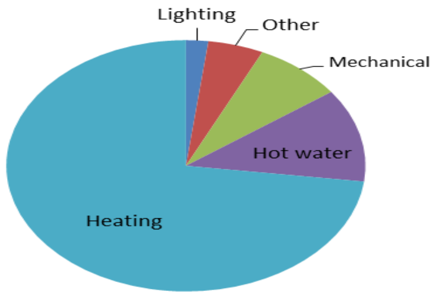

Figure 1 Structure of energy consumption in north EU households (Anisimova 2011)

A strict standardisation of temperature ranges do not yet exist but a heating system with a supply temperature around 40-45 ºC is normally called a low temperature system. For heat pumps a decreased supply temperature results in lower energy consumption for heat production and thus more efficient use of energy. LT heating systems have another environmentally friendly advantage since they can easily be combined with sustainable low temperature energy sources like solar energy, etc. Different types of low temperature heating systems are already on the market. Under-floor heating, ceiling heating and wall heating are typical examples. A ventilation radiator is another type of low temperature heating unit, where preheating of ventilation air is combined with radiator heating of the room (Myhren 2011). Cold outdoor air is filtrated and then preheated between the heat panels of the radiator before entering the room in a buoyant plume, Figure 2. The driving forces are thermal buoyancy and constant under pressure in the room generated by an exhaust ventilation fan.

Four semi-detached and newly (2011) constructed buildings are subject of the present study. Each building contains two identical dwellings. The dwellings contain three stories, four bedrooms, living room, kitchen and two bathrooms and four persons are living in this dwelling. Table 2 gives general characteristics of the dwellings. All dwellings are equipped with a heat pump (HP), which utilizes heat of the outgoing exhaust air (EA) to cover space heating and Domestic Hot Water (DHW) consumption. The heating system including under-floor heating (UFH) in the first floor and ventilation radiators (VR) in the second and third floor was adapted to low supply temperature. A main purpose of this study was to evaluate the energy performance of the buildings. The goal was achieved through a detailed comparison of measured and calculated energy performance in a representative model of the buildings. When the residents moved in December 14 the measurements could start. Our assessment period here is from December 14 (2011) to March 14 (2012).

METHOD

Energy performance of an average semi-detached house was evaluated and compared by theoretical calculation of energy consumption and site measurements. In the first method energy consumption was based on balance calculation (supplied=lost energy). Consumption estimations were here made by statistical modeling in Excel as well as by the Building Energy Simulation program IDA ICE 4, a commercial software tool. The second method was site measurement carried out by following electricity consumption for three consecutive winter months, i.e. from December 14 to March 14. Results from site measurements were compared with results from IDA ICE simulations to validate the model. Figure 3 shows the semi-detached measurement buildings as well as the physical simulation model.

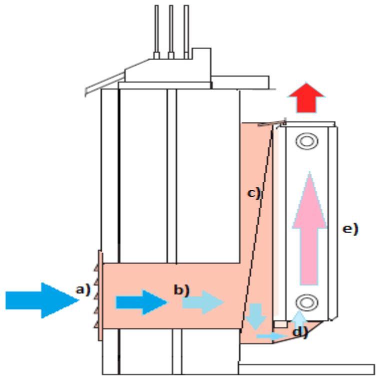

Figure 2. Main parts of ventilation radiator: a) Vent grill on the building wall b) Channel through wall c) Filter d) Injector or inlet (with or without mixing of cold supply air with room air) e) Traditional radiator (Myhren 2011)

Table 2. General characteristics of dwelling

$A_{total\ per\ dwelling}$, m^2	160
Ceiling height, m	2.5
Average U-value, W/m^2 K	0.28
Ventilation air change per hour, ach/h	0.54
Window Solar Heat Gain Coefficient	0.5
Airtightness (Leakage), (ach/h)	0.05
Heating system in the first floor	UFH
Heating system in 2nd, 3rd floor	VR
Heating source	HP
Ventilation system	EA

Figure 3. Left: photo of four identical semi-detached houses in Stockholm; Right: simulation model frame of the houses in the IDA ICE software

Energy balance calculation in monthly steady condition by Excel programming

With this method the total energy demand of a dwelling is calculated from in and out going energy flows between December 14 and March 14. The results of this simple calculation give a good estimation of energy consumption. Since the outdoor temperature is considered to be constant for a month, it is called monthly steady condition. Table 3 gives input values and details of the calculations. Areas of different construction and design were calculated according to the building's drawing. In Table 4 results of the different energy contributions in the dwelling is presented. Specific heating demand was based on transmission, ventilation and air leakage losses. Active heating demand was found by subtracting indirect/passive heating from the total demand. Frequency of clear and over cast days and mean outdoor temperature were adapted from Swedish Meteorological and Hydrological Institute (SMHI). Indoor temperature was set to 21 °C. According to the Swedish building regulation (BBR) DHW consumption is approximately 30 kWh/m^2 of floor area. With a floor area of 160 m^2 this means 4800 kWh/year of which 20% is lost in form of indirect heating. According to BBR the base household electricity consumption is 2500 kWh/year. On top of this consumption 800 kWh per person has to be added. Of the total consumption 70% contributes to indirect heating. For verification results from Excel energy balance calculations in Figure 5 were compared to results from IDA ICE simulations.

Table 3. Left: Dwelling specific transmission, ventilation and leakage losses; Right: passive heat contribution; Below: Limit temperature calculation

Transmission loss (specific)

$Q_{trans} = \sum(U_j \cdot A_j) + Q_{thermal\ bridge}$ [W/ºC] **(1)**

Construction	U-value, W/m^{2}K	Area, m^2 (Drawing of building)	U·A, W/K
Wall	0.15	116	17
Floor	0.15	58	9
Roof	0.13	65	8
Windows	1.1	27	30
Door	1.5	4	6
Thermal bridge			7
Total			77

Ventilation loss (specific)

$Q_{vent} = \rho \cdot c_p \cdot q_{vent}$ [W/ºC] **(2)**

Outdoor air inflow, m^3/s	0.06
Air density, kg/m^3	1.2
Air heat capacity, J/kg°C	1010
$Q_{ventilation}$, W/°C	72.6

Air leakage loss (specific)

$Q_{leakage} = \rho \cdot c_p \cdot q_{leak}$ [W/ºC] **(3)**

Air leakage, ach/h	0.3
Volume, m^3	441
Air leakage flow, m^3/s	0.037
$Q_{leakage}$, W/°C	44.4
Total loss, W/°C	**194**

Indirect/passive heat

People		**Hot water (Using BBR)**	
Number of persons	4	Consumption, kWh/year	4800
Heat generation per person, W	80	Utilized for heating, %	20
Presence, h/day	14	Total, kWh/year	960
Per day, Wh/day	4480		
Per day, kWh/day	**4.48**	**Per day,** kWh/day	**2.63**

Solar (SMHI)

Month	Over cast days, kWh/day	Semi-clear days, kWh/day	Clear, kWh/day
December	1.07	3.41	5.22
January	1.67	5.08	7.58
February	5	13.57	19.22
March	9.9	24.13	32.33

Electricity(Using BBR)

Base consumption, kWh/year	2500
On top per person, kWh/year	800
Number of persons	4
Total, kWh/year	5700
Percentage that turn to heat, %	70
Total contribution	3990
Per day, kWh/day	**10.93**

Month	**$E_{passive}$,** kWh	**No of days**	**Hours per day**	**Total hours**
December	336.60	17	24	408
January	658.75	31	24	744
February	764.32	29	24	696
March	585.60	15	24	360
Sum	**2345.27**			**2208**

$$T_{\lim} = T_{indoor} - \frac{P_{Passive}}{Q_{tot}} \quad \textbf{(4)} \quad , \quad P_{Passive} = \frac{E_{Passive}}{hours} \quad \textbf{(5)}$$

$$P_{Passive} = \frac{2345.27 * 1000}{2208} = 1062\,\mathrm{W}$$

$$T_{\ limit} = 21 - \frac{1062}{194} = 15\ °\mathrm{C}$$

Table 4. Heating demand for winter months by the energy balance method and degree hours calculation

Total heat losses, kWh/day	**Indirect/passive heating, kWh/day**							**Active heating, kWh/day**
	Electricity	*People*	*Hot water*	*Solar heat gain*	*No of days*	*Energy*	***Indirect/passive***	$E_{active} = E_{trans+vent+leak} - E_{ind/pass}$
Heating Demand in December, Mean temperature : 1.6 °C (SMHI 2011)								
90	10.93	4.48	2.63	Over cast days	12	1.07	**19.11**	**71**
90	10.93	4.48	2.63	Semi-clear days	5	3.41	**21.45**	**69**
90	10.93	4.48	2.63	Clear	0	5.2	**23.26**	**67**
								Total December: 1197 kWh
Heating Demand in January, Mean temperature : -2.6 °C (SMHI 2012)								
112	10.93	4.48	2.63	Over cast days	19	1.67	**19.7**	**92**
112	10.93	4.48	2.63	Semi-clear days	9.1	5.08	**23.1**	**89**
112	10.93	4.48	2.63	Clear	2.9	7.58	**25.6**	**86**
								Total January: 2812 kWh
Heating Demand in February, Mean temperature : -5.2 °C (SMHI 2012)								
122	10.93	4.48	2.63	Over cast days	16.2	5.01	**23.04**	**99**
122	10.93	4.48	2.63	Semi-clear days	8.6	13.57	**31.61**	**91**
122	10.93	4.48	2.63	Clear	3.2	19.22	**37.26**	**85**
								Total February: 2658 kWh
Heating Demand in March, Mean temperature : -0.4 °C (SMHI 2012)								
100	10.93	4.48	2.63	Over cast days	2	9.9	**27.9**	**72**
100	10.93	4.48	2.63	Semi-clear days	9	24.13	**42.1**	**58**
100	10.93	4.48	2.63	Clear	3	32.33	**50.3**	**50**
								Total March: 816 kWh

Table 5. Nomenclature

$U, W/m^2 °C$	Heat transfer coefficient
E, kWh/year	Energy Demand
ρ, kg/m^3	Air density
c_p, J/kg°C	Air heat capacity
q_{vent}, m^3/s	Outdoor air inflow
q_{leak}, m^3/s	Assumed air leakage
Q_{tot}, W/ ºC	Total Loss
D_h, h	Degree hours
A, m^2	Area

$$E = Q_{tot} \cdot D_h \qquad \textbf{\textit{(6)}}$$

$$E = Q_{tot} \text{ W/ ºC} \cdot Area \text{ ºC h / year}$$

$$E = 194 \cdot 80280 = 15570 \text{ kWh/year}$$

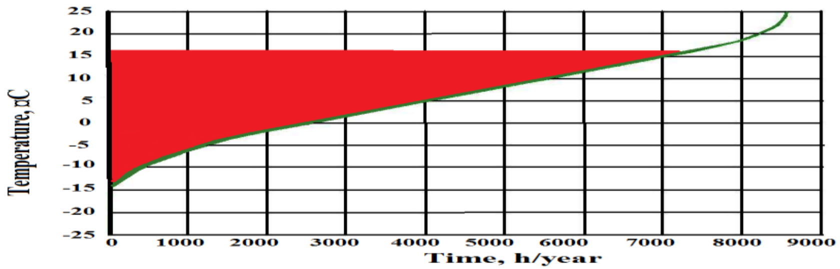

Figure 4. Degree hours calculation

Energy balance calculation by building energy simulation

Building Energy Simulation (BES) is widely used to predict energy consumption in buildings. IDA ICE (Indoor Climate and Energy) is a BES tool for thermal comfort, indoor air quality and energy consumption in buildings. It was originally developed at KTH Royal Institute of Technology, Division of Building Services Engineering and the Swedish Institute of Applied Mathematics, ITM (Jokisalo et al. 2008). In this project the program calculates monthly energy consumption of a building by solving heat balance equations on description of the building geometry, construction, HVAC conditions and internal heat gain. An early validation study of IDA ICE showed good agreement between simulated and measured data (Travesi et al. 2001). The program has since then continuously been developed and compared to many different applications, and the number of users has continuously risen. Here IDA ICE predictions are compared to site measurements. Climate data in IDA ICE includes outdoor air temperature, relative humidity, wind direction, wind speed and solar radiation (direct normal radiation and diffuse radiation on a horizontal surface) for every hour. By a finite difference method and transient calculations the program predicted energy balances for every 1.5 hour. Calculations for each time steps were completed when the residuals of all variables were less than the tolerance level, normally 0.02. In the energy calculations by Excel (steady state) only the average monthly outdoor temperature and estimated monthly solar heat gain are taken into account. Also, the storage and emission of heat in the structure of the building are accurately calculated in IDA ICE. This is important for the power demand calculation in the building. In the Excel programming, on the other hand, influence of thermal inertia (time constant) of the building was neglected. Hence the building energy simulation by IDA ICE is supposed to be more accurate than Excel programming in calculating the energy consumption.

Model description

In the IDA ICE simulation, the physical model of the building was developed. Room heating model for the first floor was UFH and for the second and third floor VR. Supply air was placed above the windows for the first floor and behind the radiators for the second and third floor. Indoor temperature was set at 21±0.5 ºC. The model consists of 25 zones which follow the original geometries of the building. The different zones are assigned to the usage of the space, i.e. one zone for the kitchen, one zone for the living room, etc. Constant Air Volume (CAV) system is modelled for the ventilation system with different exhaust air flow rate for the different zones. Infiltration distribution is assumed to be in proportion to the external surface area. Periodic time-steps were used during the simulations. Default value of maximum time-step and tolerance were set to 1.5 hour and 0.02 respectively i.e. the energy balance calculation was done for every 1.5 hour till the relative difference between the two last calculations for all variables became less than 0.02. Climate data is based on local outdoor conditions during the period, observed by the SMHI. It is assumed that occupants spend 14 hours per day at home, except for holidays (20 hours). Occupants are wearing light clothing and they are resting, i.e. the activity level is 1 MET (metabolic equivalent) and the clothing is 0.85 CLO (overall clothing insulation). The coefficient of occupancy level varies from 0 when occupants are not at home to 1 when home is fully occupied.

Site measurement description

The energy consumption for space heating and DHW in this residential area was also measured by monitoring the electricity consumption during the winter period of three months. Occupants were asked to read and report the electricity consumption each month from the heat pump electricity metering device display. In order to achieve the heating demand this value was multiplied by the COP value (Coefficient of Performance) of the heat pump. According to the investigation by (Nowacki 2007) the COP for exhaust air heat pump producing hot water of 50 °C is 2.7. Finally, this consumption was compared to simulated figures.

RESULTS AND DISCUSSION

Comparison between energy balance calculations in Excel and IDA ICE simulations

To verify results from energy balance calculation in Excel a comparison with results from IDA ICE simulations is shown in Figure 5. Predicted heating demands with the two methods are in good agreement since both methods used the same concept of energy balance. The total energy consumption for the selected winter months predicted by IDA ICE is 58.4 kWh/m^2. This is 19% higher than the prediction by energy balance calculation in Excel giving 46.8 kWh/m^2. In February the difference between energy consumption is higher than other months which can be attributed to the more solar radiation and more sunny days than expected used in Excel programming.

Comparison between site measurement and IDA ICE results

Total energy consumption during the winter period is in this step compared between results from IDA ICE simulations and site measurements, see Figure 6. This way we hope to have a reasonable check (validation) of accuracy in presented results. An average variation of 9 % is found between IDA ICE (58.4 kWh/m^2) and measurements (52.8 kWh/m^2). Input data such as internal heat gains or occupants living habits are influencing the results. Also the influence of shading by surrounding buildings and roof overhang was ignored in IDA ICE simulations. This may result in a lower active heating demand due to an overestimation of the passive solar energy contribution in simulation.

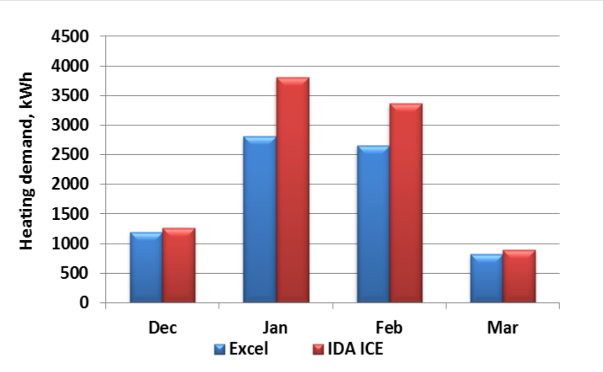

Figure 5. Comparison of results from energy balance calculation in Excel and IDA ICE simulations

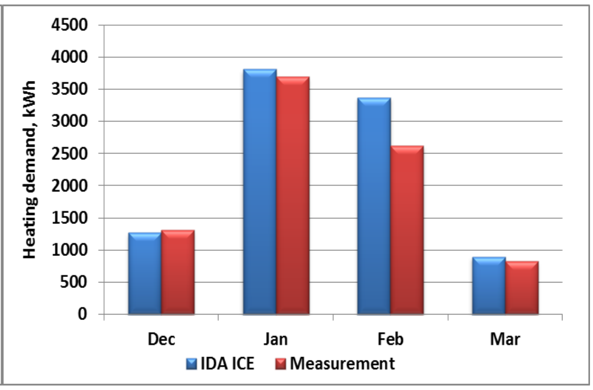

Figure 6. Comparison between site measurements and simulation results by IDA ICE

CONCLUSION

In this study the energy performance of the new residential area in Stockholm, which is equipped with low temperature heating systems, was evaluated by comparing the results of energy consumption by different methods. The first method includes energy balance calculations by Excel programming (steady state) and the IDA ICE simulation tool (transient). In this method the total active heating demand was found by using formulas of transmission, ventilation and leakage losses and indirect/passive heating contribution. The simulation results were based on input data of building geometry, construction, HVAC and internal heat gain. Results from the first method with energy balance formulas in Excel were verified by comparing them to IDA ICE results. Results from the comparison showed that a good agreement existed between the energy balance calculations in Excel and IDA ICE simulations and they are following the same concept of energy balance. To find the real energy consumption site measurements were arranged by monitoring the electricity consumption of the heat pump in the dwelling. Electricity consumption of the heat pump was then multiplied by the COP value to find the required heating demand. Living habits of occupants are influencing the results and more accurate information here will most likely give an improved picture of the heating demand.

To find the energy demand in the building for the whole year the concept of degree hours was used (Table 3 and Figure 4). This is a simplified method to calculate the building energy demand for active heating. The degree hours depends on the building location, the chosen indoor temperature and the indirect/passive heat supply. The integrated area (°C · h) from the duration curve of the outdoor temperature over a year to the horizontal limit temperature represents the need of active heating, i.e. degree hours (D_h). This area visualises the concept of degree hours. The heating contribution from the limit temperature to comfort temperature is given by indirect/passive heating. The duration curve in Figure 4 (Jansson J and Wetterstrand M) is for Stockholm (normal temperature 6.6 °C). According to the Equation 4 the limit temperature is 15 °C. So the annual energy demand (E) in the building is 15570 kWh/year (Equation 6). The total energy consumption of the building and also the average value for DHW leads to a total consumption of 47 kWh/m^2. This is below Swedish regulations for heat pump systems, where the upper limit is 55 kWh/m^2. How much the result has to do with the selected room heaters (ventilation radiators on two floors and floor heating in the basement) is too early to say but the result gives us an interesting indication.

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Demand Controlled Ventilation in a Combined Ventilation and Radiator System

Arefeh Hesaraki[#1] and Sture Holmberg[*2]

[#,*]*Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Brinellvägen 23, Stockholm, Sweden*

#arefeh.hesaraki@byv.kth.se

*sture.holmberg@byv.kth.se

Abstract:

With growing concerns for efficient and sustainable energy treatment in buildings there is a need for balanced and intelligent ventilation solutions. This paper presents a strategy for demand controlled ventilation with ventilation radiators, a combined heating and ventilation system. The ventilation rate was decreased from normal requirements (per floor area) of 0.375 l·s^{-1}·m^{-2} to 0.100 l·s^{-1}·m^{-2} when the residence building was un-occupied. The energy saving potential due to decreased ventilation and fan power was analyzed by IDA Indoor Climate and Energy 4 (ICE) simulation program. The result showed that 16 % of the original energy consumption for space and ventilation heating could be saved by utilizing ventilation on demand.

Keywords- *Controlled ventilation system; Energy performance; IDA ICE 4; Variable air volume*

1. Introduction

Demand-controlled ventilation (DCV) system adjusts the ventilation supply air based on the occupants ventilation need. In designing the air handling unit for commercial buildings, the variable air volume (VAV) system is widely used to optimize the energy consumption. However, the usage of VAV system in residential buildings is not a very common solution. In current residential buildings, people go to work or school from morning until afternoon. This gives opportunity to reduce the ventilation rate during non-occupancy hours. Existing Swedish building regulations, BBR [1] states that an occupied house must have a minimum ventilation rate (per floor area) of 0.35 l·s^{-1}·m^{-2}, and if the house is unoccupied, this value is allowed to drop down to 0.10 l·s^{-1}·m^{-2}. Thus, there is an opportunity to save energy by reducing the ventilation rate during hours of un-occupancy. The control of the VAV system can be operated in many ways, such as automatic control based on the CO_2 level, humidity or clock control, or manual control that leaves responsibility of reducing the airflow rate to the occupants.

The goal of this study was to quantify energy savings when applying VAV system control theoretically to the actual building. Simulation program, IDA Indoor Climate and Energy (ICE) 4 was used.

2. Method

To investigate the feasibility of DCV, and to determine the DCV type in the studied building, the decision tree shown in Fig. 1 was used [2]. A first step in the flowchart was to determine whether a VAV system is compatible with law. In our case, DCV system is compatible with current Swedish building law, and the pollution sources in residential buildings include occupants, furniture and building materials [2]. Next step in the decision tree is to determine whether natural infiltration is enough for ventilation. In Sweden the residential buildings need to have a high level of air tightness, so mechanical ventilation is required for new and un-leaky buildings. Moving forward in the flowchart; the emission rate is variable over the day since occupants leave house in the morning and return home in the afternoon. Also, the emission rate in the investigated building is predictable. Hence, we suggest “A clock-controlled mechanical ventilation system”.

For the selected DCV option a VAV model was used in IDA ICE 4 to investigate possible energy savings. IDA ICE 4 is a dynamic multi-zone simulation program that provides a detail study of thermal indoor climate and annual energy consumption in the building [3].

2.1. Description of the Studied Building and the Building Model

A semi-detached house was used in this study. The building was 160 m^2 with three stories including kitchen, bathrooms, living room and bedrooms. Mechanical exhaust ventilation extracted in total 60 $l\cdot s^{-1}$ from exhaust devices in kitchen, toilet, wardrobe and bathroom. This corresponded to 0.375 $l\cdot s^{-1}\cdot m^{-2}$ of floor area and fulfilled Swedish minimum requirements of 0.350 $l\cdot s^{-1}\cdot m^{-2}$. Supply devices were placed on the external walls above windows in living room, kitchen and hallway on the first floor, and behind radiators in bedrooms on the upper floors. The air was transferred through hallways and staircases between different zones. The same ventilation system was modeled in IDA ICE 4; i.e. the supply air was forced by an indoor under-pressure to enter the building. Blower door test showed that the airtightness of the building was 0.63 $l\cdot s^{-1}\cdot m^{-2}$ (external surface) under 50 Pa pressure difference. Table 1 shows the area and airflow rate in each zone.

In the investigated building occupants spent approximately 14 hours per week-day and 20 hours per weekend-day indoor. To model VAV system based on the presence of occupants, a simple control strategy was implemented. In this strategy the ventilation was switched between two flow rates: normal requirements (per floor area) of 0.375 $l \cdot s^{-1} \cdot m^{-2}$ and minimum base requirement of 0.100 $l \cdot s^{-1} \cdot m^{-2}$ during un-occupancy. This minimum value is 27 % of the normal requirement. Reduced ventilation rate was conducted by decreasing the fan speed.

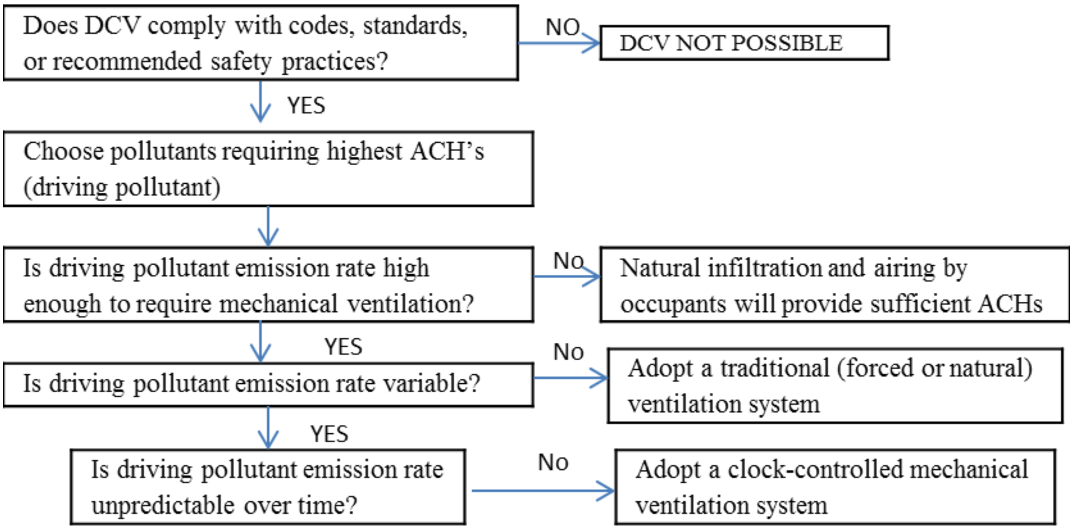

Fig. 1. Decision tree for designing demand controlled ventilation (DCV) [2]

Table 1. Floor area and supply/exhaust airflow rates in different zones of the building

zone	Area, m^2	Supply, $l \cdot s^{-1} \cdot m^{-2}$	Exhaust, $l \cdot s^{-1} \cdot m^{-2}$
Living room	24.0	0.50	0.00
Kitchen	13.1	0.30	0.80
Hallway	6.8	0.60	0.00
Bedroom 1	13.1	0.50	0.00
Bedroom 2	8.0	0.50	0.00
Bedroom 3	8.2	0.50	0.00
Bedroom 4	12.1	0.50	0.00
Toilet	6.1	0.00	1.80
Bathroom	4.4	0.00	3.40
Wardrobe	2.9	0.00	2.10
Pass	7.3	0.00	0.00
Third floor	54.0	0.30	0.30
Total	160.0	0.37	0.37

3. Results

The IDA ICE simulation model with CAV system was validated against measured energy consumption [3]. Table 2 shows results of heating demand from CAV and VAV simulations. By introducing VAV system, the total heating demand for ventilation and space heating decreased by 16 %, i.e. from 4624 kWh to 3884 kWh. Energy savings in heating and ventilation demand, fan power and total consumption (including DHW) is separately shown in Table 2. As shown, with VAV system the fan power and total energy demand was decreased by 30 % and 12 % respectively compared to CAV system. Domestic hot water (DHW) consumption was achieved by measuring electricity consumption during the non-heating period in the summer.

Table 2. Energy savings with VAV compared to CAV system

	CAV	VAV	Energy savings, %
Total heating and ventilation demand, kWh	4624	3884	16
Fan power, kWh	473	328	30
DHW, kWh	3200	3200	-
Total energy consumption, $kWh \cdot m^{-2}$	52	46	12

3.1. Influence of DCV (VAV) System on Ventilation Radiator

Heating demand in the investigated building was mainly performed by ventilation radiators, shown in Fig. 2. Heat output of ventilation radiator is affected by changing the airflow rate, since the ventilation air is supplied through the radiator. The heat output decreases by decreasing the airflow rate due to reduced forced convection. To ascertain whether ventilation radiators cover the heating demand while decreasing the airflow rate, heat output of radiators was calculated by an Excel-based program called Air-simulator. This program calculates heat output of ventilation radiators based on the indoor and outdoor temperature, hydronic supply and returns flows to the radiator as well as supply air flow rate to the radiator. Calculations showed that by using VAV system, the total heat loss was 4481 W. This was successfully covered by the heating systems since total heat output from ventilation radiators and under floor heating was 4524 W, see Fig. 3.

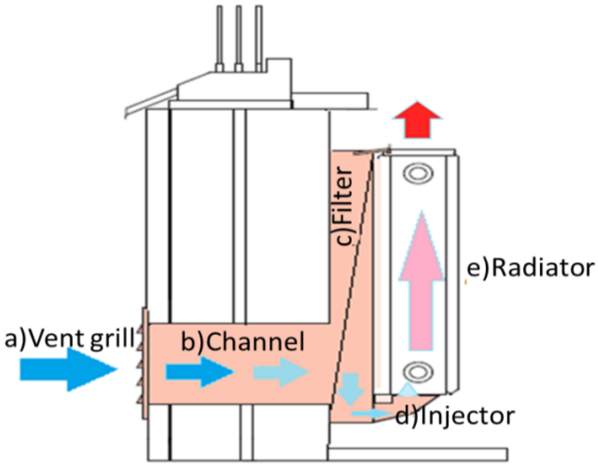

Fig. 2. Main parts of a ventilation radiator: a) Vent grill on the external wall, b) Channel through wall, c) Filter, d) Injector or inlet (with or without mixing of cold supply air with room air) e) Traditional radiator

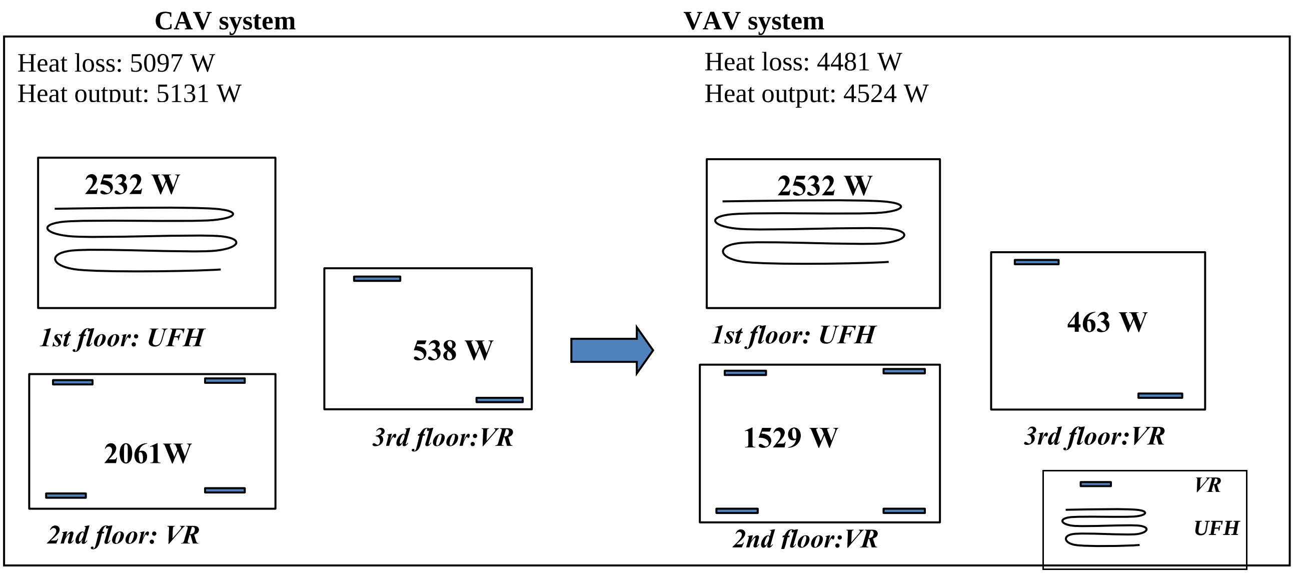

Fig. 3. Heat loss and heat output from ventilation radiators (VR) and under-floor heating (UFH) in constant air volume (CAV) and variable air volume (VAV) systems

4. Conclusion

DCV with variable air volume is an energy efficient system. Ventilation rate was adjusted to the occupant demands. This system reduces energy demand by decreasing ventilation and electrical energy for driving fans. The purpose of the study was to ascertain energy savings of a VAV system by IDA ICE 4 modeling. Results confirmed that VAV system was an energy efficient system with significant savings in comparison to CAV system.

Ventilation radiators were installed in the investigated buildings. Heat output from ventilation radiators was decreased by decreasing airflow rate; since ventilation was supplied through the radiator. This study showed that ventilation radiators can cover the heat losses in the investigated building with a VAV system. However, in order to investigate the long term real performance of the systems, an implementation in an occupied building is recommended.

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An investigation of energy efficient and sustainable heating systems for buildings:

Combining photovoltaic with heat pump

Arefeh Hesaraki and Sture Holmberg

Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Handen-Stockholm, Sweden

Arefeh.Hesaraki@byv.kth.se

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Abstract Renewable energy sources contribute considerable amounts of energy when natural phenomena are converted into useful forms of energy. Solar energy, i.e. renewable energy, is converted to electricity by photovoltaic systems (PV). This study was aimed at investigating the possibility of combining PV with heat pump (HP) (PV-HP system). HP uses direct electricity to produce heat. In order to increase the sustainability and efficiency of the system, the required electricity for the HP was supposed to be produced by solar energy via PV. For this purpose a newly-built semi-detached building equipped with exhaust air heat pump and low temperature-heating system was chosen in Stockholm, Sweden. The heat pump provides heat for domestic hot water (DHW) consumption and heating demand. Since selling the overproduction of PV to the grid is not yet an option in Sweden, the PV should be designed to avoid overproduction. During the summer, the HP uses electricity only to supply DHW. Hence, the PV should be designed to balance the production and consumption during the summer months. In this study two simulation programs were used: IDA Indoor Climate and Energy (ICE) as a building energy simulation tool to calculate the energy consumption of the building, and the simulation program WINSUN to estimate the output of the PV. Simulation showed that a 7 m^2 PV area with 15 % efficiency produces nearly the whole electricity demand of the HP for DHW during summer time. As a result, the contribution of free solar energy in producing heat through 7 m^2 fixed PV with 23º tilt is 15 % of the annual heat pump consumption. This energy supports 58 % of the total DHW demand.

1 Introduction

Due to the scarcity of fossil fuel sources and their environmental impact, renewable energy has become an increasingly important topic over the past decades. On a global scale renewable energy sources only contribute less than 15 % (Lund 2007) to the primary energy supply; however, during the last few decades this percentage has increased considerably in some countries.

Energy from solar radiation can be obtained in two ways, passively and actively. Passive solar design is based on the optimal design of a building's shape leading to the capture of as much solar radiation as possible for space heating. Active solar design is based on converting solar radiation into energy by using solar thermal collectors or photovoltaics. Photovoltaics (PV) convert sunlight into electric power by a solid-state device called solar cells. The common PV module converts 15 - 20 % (Tyagia et al. 2012) of the incoming solar radiation into electric energy, depending on the type of solar cells. Our total present energy demand can be supplied if 0.1 % of the earth's surface were covered with solar cells with 10 % efficiency (Tyagia et al. 2012). Solar thermal collectors convert solar radiation into thermal energy through a transport medium, liquid or gaseous. Solar cells are more efficient than solar thermal collectors, since they are performing during the winter months at low irradiation with constant efficiency while the solar collector has very low efficiency during hours of low intensity due to a high heat loss (Gajbert 2008).

The electricity generated by PV could be utilised by a heat pump to produce heat. Depending on the Coefficient of Performance (COP) of the heat pump, the energy required for space heating and domestic hot water (DHW) may be decreased by a factor of the COP. The outputs of 1 m^2 of plane solar collector delivering hot water at 50 °C and 1 m² of PV modules of 15 % efficiency in combination with a heat pump (PV-HP system) with COP=3 in Stockholm were compared using the WINSUN program, see Figure 1. These two systems, solar collector and HP-PV are comparable since both produce hot water. It can be seen that the combination of the PV module and the heat pump has a higher annual output than the solar thermal collector. During winter time, i.e. low temperature and low irradiance, the solar collector has zero efficiency due to high heat loss. The heat loss of the solar collector is dependent on the collector efficiency factor (F´), heat loss coefficient from absorber to ambient (U_L ,W/m^{2}K) and temperature difference between ambient temperature (T_a) and collector temperature (T_c), Equation 1 (Duffie and Beckman 1991). PV is usable year round performing even at low intensity since it works on light not heat. In the summer time both the solar collector and PV have good characteristics. As a result, a PV-HP system is more efficient since it provides a higher annual solar fraction in comparison with a solar thermal collector system. Solar fraction (SF) gives the fraction of energy provided by solar energy to an annual heating demand. SF varies between 0 when no energy is supplied by solar energy to 1 when all required energy is supplied by solar technology.

$$P_{loss\_collector} = F' * U_l * (T_c - T_a) \tag{1}$$

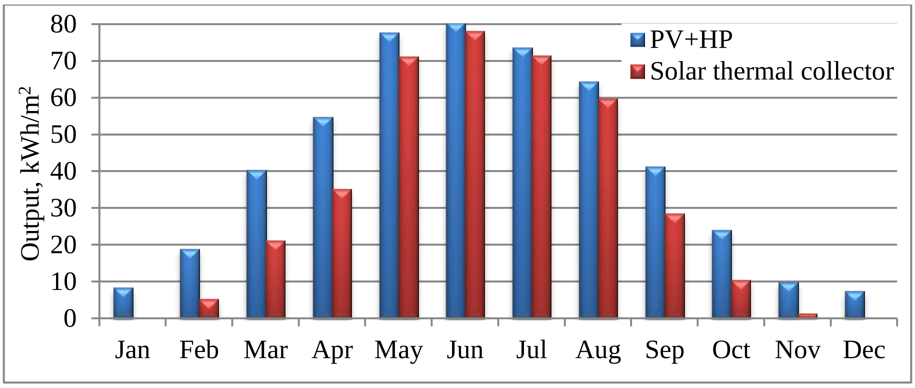

Fig. 1. Output comparison between photovoltaic + heat pump and solar thermal collector

The present paper points to the possibility of combining PV with heat pump in an actual building equipped with an exhaust air heat pump in Stockholm. For this purpose a Building Energy Simulation (BES) program, IDA Indoor Climate and Energy (ICE) was used to calculate the energy demand in the building. Then, using the System Simulation Program (SSP) WINSUN the output of PV with 15 % efficiency was calculated to produce the electricity for the HP. Since there is yet no regulation in Sweden concerning selling excess solar power to the grid, overproduction of electricity was avoided. Hence, the area of the PV modules required should be calculated to balance energy consumption and production when the production is at the highest point and the consumption is at the lowest value (during the summer season). However; in the future there might be a policy for selling extra electricity to the grid. It might then be profitable to produce more electricity than needed and export it to the grid.

2 Method

In this study two simulation programs were used, one for finding the energy requirements of the building and the other one for calculating the output of appropriate PV to partly meet this demand.

2.1 Building Energy Simulation (IDA ICE)

IDA Indoor Climate and Energy (ICE) is a Building Energy Simulation (BES) tool to calculate thermal comfort, indoor air quality and energy consumption in buildings. This program was partly developed at KTH (Jokisalo et al. 2008). An early validation of the IDA ICE program was conducted by comparing the results predicted by simulation with measurements; they showed good agreement (Travesi et al. 2001).

The purpose of using the IDA ICE program in this study was to calculate the monthly energy consumption for a whole year in order to find the electricity consumption of the heat pump. The electricity consumption for providing domestic hot water and space heating by the heat pump was calculated by dividing energy consumption by the Coefficient of Performance (COP) of the heat pump.

When performing IDA ICE analyses the input data include building location, geometry, construction type, HVAC system, internal heat gain (number of people, light and equipment) and DHW consumption. Referring to the Swedish building regulations (Boverkets byggregler, BBR), the average value for DHW usage is 30 kWh/m^2 of floor area. The area of the studied building is 160 m^2 leading to 4800 kWh/year of DHW. This theoretical value was used in this study since there is no available information regarding actual DHW consumption yet. Through creating a mathematical model and solving heat balance equations, the energy consumption in the building was estimated by IDA ICE. Details of the simulation and validation of results for the studied building may be found in a previous study (Hesaraki and Holmberg 2012).

2.2 System Simulation Program (WINSUN)

Simulation tools are preferred for analysing a system rather than implementing an actual device on site, since studying a model is essentially easy and inexpensive. WINSUN is a system simulation program developed at Lund University for designing solar collectors and PV (Hatwaambo et al. 2008). WINSUN is an abbreviation of Windows version of MINSUN usable in DOS (Boström et al. 2003). WINSUN is based and developed completely on PRESIM, TRNSYS and TRNSED version 14.2 (Boström et al. 2003). PRESIM is a graphical modeling program used for producing input data for TRNSYS, developed by the Solar Energy Research Center in Sweden (Beckman et al. 1994). In TRNSYS (TRaNsient Systems Simulation) thermal energy equations are solved based on a modular approach depending on the input data (Beckman et al. 1994). TRNSED is text editor program to create a user-friendly TRNSYS interface of a solar energy system, and convert the input file to a TRNSED-formatted document to be usable for other users, developed at the University of Wis-consin, Madison (Beckman et al. 1994, Hatwaambo et al. 2008). WINSUN aims to provide an output of a solar thermal collector and PV in kWh/m^2 depending on the location of system, azimuth, tilt, tracking mode and efficiency of PV. The program uses the weather data during 1983-1992 including diffuse and beam radiation collected by the Swedish Meteorological and Hydrological Institute (SMHI) (Boström et al. 2003). Comparing the solar radiation intensity during the long term at all weather stations, the divergence between measured solar radiations is less than 2 % (Hatwaambo et al. 2008). Hence, the results of the program though using old weather data appear reliable. The validation of the WINSUN program was conducted by comparing the simulation results with site measurements, which showed good agreement (Gajbert 2008). The results of WINSUN may be found in two ways: one as a table and plot file, and the other as an online plot where several variable changes may be watched and analysed at the same time as TRYNSY calculations are running to solve thermal energy equations. In this study, WINSUN simulation was conducted to evaluate the performance and output of fixed PV with 15 % efficiency at 23º tilt towards the south (according to the plan of the chosen building). Simulation was based on monthly net metering. Export of overproduction to the grid was avoided. Input data to the simulation include starting day of simulation, month and length of simulation, site of place, climate, tracking mode (1 for fixed, 2 for turning around vertical axes, 3 for turning around an axis in the plane of the glass, 4 for 2 axes tracking), ground reflectance (typical value 0.2-0.3), slope of surface from horizontal plane and azimuth of surface (azimuth angle from the south i.e. -90 for east, 0 for south and 90 for west).

3 Results and Discussion

3.1 IDA ICE Simulation Results

Using the IDA ICE program, the results for electricity usage in the heat pump to provide heat for space heating and DHW are shown in Figure 2. The DHW consumption was assumed to be equal for all months (Gajbert 2008) since it may not be dependent of weather condition. Considering the average COP of the exhaust-air heat pump producing hot water at 50 ºC, which is 2.7 according to investigation by (Nowacki 2007), heating demand was decreased by a factor of 2.7. The electricity consumption by the HP for space heating depends on the weather conditions, heat losses i.e. ventilation loss, transmission loss and leakage loss, and passive heating (contribution of free heating source such as people, equipment, solar energy and lights in heating the house).HP consumption for space heating varies for different months, it is at its highest level in winter and zero in summer; however, the electricity consumption by the HP for DHW was assumed to be constant over the year (1778 kWh/year or 148 kWh/month).

The efficiency of an electrical pump in converting input energy into useful energy is usually 80 % (Bhargava et al. 1991) due to losses in the compressor. Thus, for running the pump, the solar cells must generate electricity equal to P/0.8 = 1.25 P, where P is the heat pump consumption. So, the required electricity for the heat pump increases by a factor of 1.25, i.e. for DHW consumption the required electricity to be generated by PV is 148*1.25 = 185 kWh/month

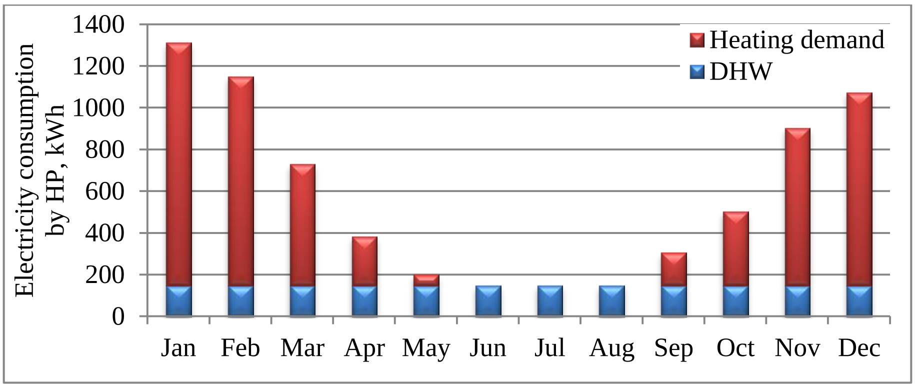

Fig. 2. Monthly electricity consumption of a heat pump with COP 02.7

3.2 WINSUN Results

Using the WINSUN program, the output for a fixed photovoltaic system is given in Figure 3. This is for one year with 15 % efficiency and southward orientation with a roof slope of 23º in Stockholm. As shown, the electricity produced by PV varies during the year depending on the solar irradiation. PV can work with acceptable output even during the cold and mostly dark winter time since they can work even on slight light. The output of the PV drops considerably during the winter months due to the low solar intensity in comparison with the summer months when the output has its maximum value. The annual output for the chosen PV is 228 kWh/m^2 year independently of the cell temperature.

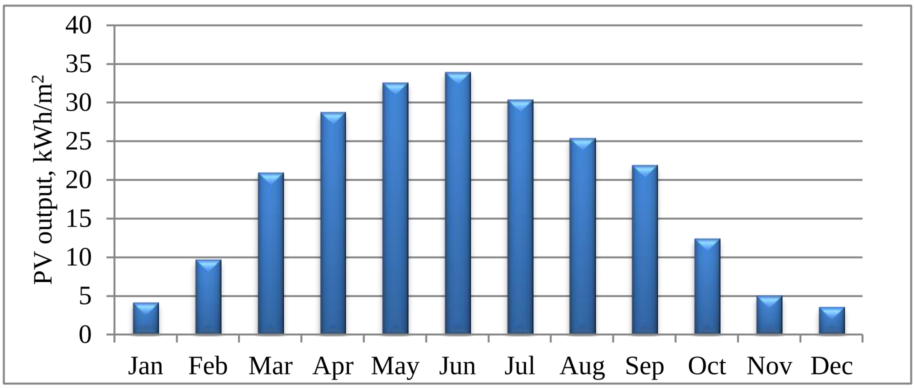

Fig. 3. Photovoltaic output at 15 % efficiency in Stockholm

The efficiency of the PV decreases by increasing the cell temperature, there are many factors that determine the operating temperature of PV such as ambient air temperature, type of module, intensity of sunlight and wind velocity.

The efficiency of a solar cell is usually determined under standard test conditions, i.e. cell temperature is 25 ºC and normal incidence is 1000 W/m^2. Solar cells are generally exposed to temperatures ranging from 15 ºC to 50 ºC (Singh and Ravindra 2012). Mainly, the operating temperature of the module is higher than 25 ºC and the angle of incidence is larger than 0° which is not considered in the WINSUN program. So, the output of PV should be multiplied by a correction factor φ. The correction factor varies for different months depending on the cell temperature, solar intensity and solar angle. Measurements conducted in Switzerland from 1992 to 1996 monitored the monthly correction factor for a PV, see Figure 4 (Häberlin 2012).

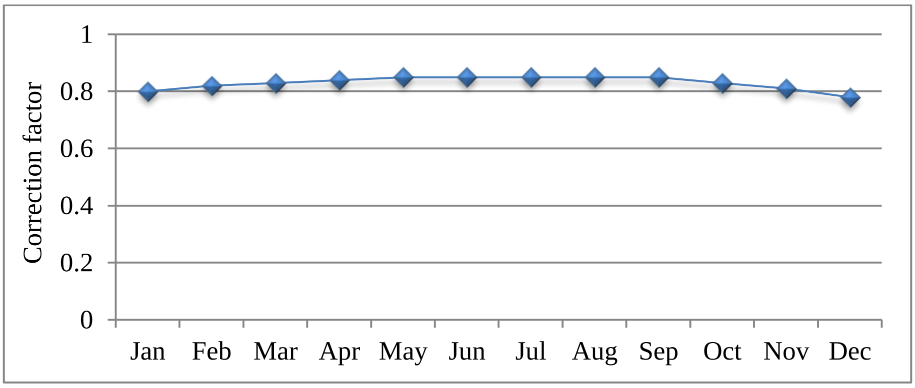

Fig. 4. Correction factor of PV output for different months (Häberlin 2012)

As mentioned before, to avoid overloading in the summer the PV should be designed to supply the electricity for DHW production. By monthly net metering of the simulation, to determine the module area (Equation 2), the month which gave the maximum output of PV was June. Thus, the production and load are in balance in June. Hence, to find the required area for the PV the energy demand (kWh) for this month was divided by the PV output (kWh/m^2) for the same month. So, by dividing 185 kWh by 27 kWh/m^2 the area demand of the PV was found to be 7 m^2.

$$\text{Area (m}^2\text{)} = \frac{energy\ demand\ in\ June\ \ (kWh)}{energy\ produced\ by\ PV\ in\ June\ (kWh/m2)} \qquad (2)$$

To find the annual solar fraction (SF, Equation 3) the electricity generated by 7 m^2 PV was divided by the electricity consumption by the HP during the whole year. Figure 5 gives the comparison between production and consumption of electricity for each month. Simulation results showed that the solar fraction is 15 %. However, if export to the grid were allowed the solar fraction might be improved by increasing the area of the PV and then ignoring the overloading problem.

$$\text{SF}=\frac{\text{Electricity generated by PV}\left(\frac{\text{kWh}}{\text{year}}\right)}{Electricity\ consumed\ by\ HP\ \left(\frac{kWh}{year}\right)} * 100 \qquad (3)$$

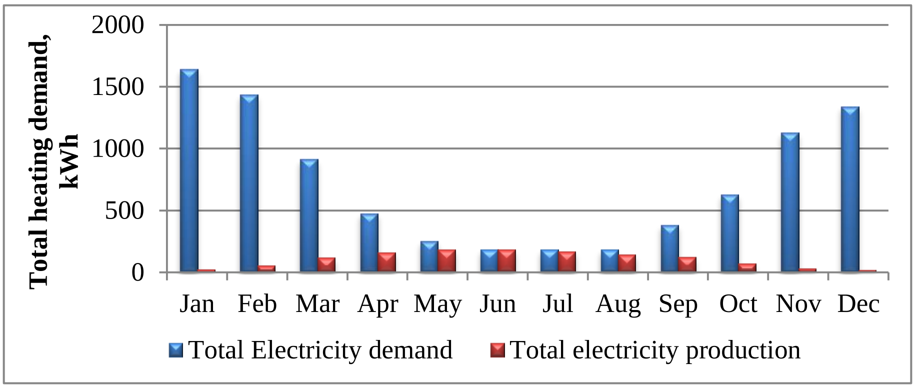

Fig. 5. Annual solar fraction calculation for total heating demand (space heating + domestic hot water)

4 Conclusion

The objective of this study was to investigate and evaluate the performance of the PV-HP system for the building in Stockholm. In addition, the solar fraction was calculated. The method included two simulation programs, one for calculating the energy demand of the building and the other for designing a PV system. Investigation of energy performance was conducted using the IDA ICE program. Simulation showed that the total energy consumption in the building is 13552 kWh/year including 8752 kWh for space heating and 4800 kWh for DHW. To avoid overproduction with monthly net metering, the PV-HP system should be designed to create a balance between production and demand during summer. WINSUN, a system simulation program, was used to design appropriate PV. Simulation showed that using 7 m^2 PV with 15 % efficiency would create a good balance during the summer season in generating and consuming electricity by PV and HP, respectively. In other words, the electricity consumed by HP is more or less totally supplied by PV during summer. The annual solar fraction (SF) in a designed HP-PV system was 15 %. If electricity generated by PV is used only for DHW the SF value is 58 % (Equation 4), i.e. more than half of the DHW need would be covered by implementing only 7 m^2 PV, Figure 6.

$$SF_{DHW}=\frac{\text{electricity generated by PV}\left(\frac{kWh}{year}\right)}{electricity\ consumed\ for\ DHW\left(\frac{kWh}{year}\right)}*100 \quad (4)$$

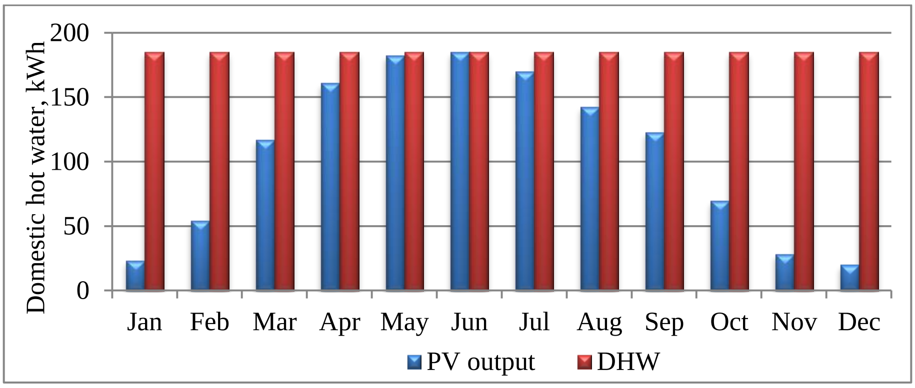

Fig. 6. Annual solar fraction calculation for domestic hot water consumption

It can be concluded that by implementing the HP-PV system with monthly net metering a relatively high solar fraction is achieved by the small PV area. The solar fraction may be increased when the heat loss is decreased which can be reached by minimising transmission, ventilation and leakage losses, i.e. using low U-value materials with high air tightness in the building envelope or using energy-saving equipment such as a heat pump or heat exchanger ventilation. Also, the solar fraction may be increased by increasing the PV area if the monthly net metering is disregarded. In this project, combining the heat pump with PV system (PV-HP) was introduced as an energy-efficient and sustainable solution. The need to supply 20 % of energy demand by renewable energy by 2020, as a European Union target (European Environment Agency Report 2006), will lead to the design of more sustainable and energy-efficient systems. Using renewable and sustainable systems particularly in the building sector will reduce environmental problems such as carbon dioxide concentration and the global warming effect.

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