Ontwerpstrategieën voor residentiële ventilatie

Design Strategies for Residential Ventilation Systems

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Nomenclature

B	Boundary condition	-
c	Specific heat capacity	J/(kg·K)
C	Concentration	-
Cp	Wind pressure coefficient	-
E	Emission rate	l/s
f	Conversion factor	-
$\dot{g}$	Flow rate	m^3/s
IF	Intake fraction	-
n	Envelope leakage	-
N(μ,σ2)	Normal distribution	-
O	Occupant	-
P	Probability	-
p	pressure	Pa
q	Flow rate specific energy use	$J/(m^3 \cdot h)$
Q	Energy	J
RH	Relative humidity	%
RR	Rebreathing Rate	-
t	Time	s
T	Temperature	K
v	Specific leakage rate	$m^3/(h \cdot m^2)$

Greek Symbols

β	Relative intake fraction	-
ε	Effectiveness	-
Δ	Difference	-
μ	Mean	-
η	Efficiency	
ρ	Density	Kg/m^3
σ	Standard deviation	-
Φ	Heat flux	W

Subscripts and Superscripts

50	At 50 Pa pressure difference
a	Air
adv	Adventitious
amb	Ambient
b	Breathing
cf	Constant flow

dc	Demand control
e	Exhaled
env	Envelope
eq	Equivalent
exh	Exhaust
exp	Exposed
g	Gas
HR	Recovered
in	Indoor
inf	Infiltration
LL	Lower limit
Mech	Mechanical
out	Outdoor
sup	Supply
thr	Threshold
tot	Total
trck	Trickle ventilator
UL	Upper limit
v	Ventilation
vent	Vent
wm	Well mixed

Acronyms

ACH	Air Changes per Hour
AHU	Air Handling Unit
AIVC	Air Infiltration and Ventilation Center
(A)LRI	(Acute) Lower Respiratory Infection
ASHRAE	American Society for Heating, Refrigeration and Airconditioning Engineers
CFD	Computational Fluid Dynamics
CF	Constant Flow
CO_2	Carbon dioxide
COPD	Chronic Obstructive Pulmonary Disease
DALY	Disability Adjusted Life Year
DC	Demand Control
DHW	Domestic Hot Water
DNS	Delayed Neurocognitive Sequelae
EAHP	Exhaust Air Heat Pump
EBD	Environmental Burden of Disease
ECBCS	Energy Conservation in Buildings and Community Systems
ECD	Electron Capture Detector
EMPD	Effective Moisture Penetration Depth
EPBD	Energy Performance of Buildings Directive
EU	European Union
GC	Gas Chromatography

HDD	Heating Degree Days
HRU	Heat Recovery Unit
HRV	Heat Recovery Ventilation
IDA	Indoor Air
IEA	International Energy Agency
LDS	Low Discrepancy Sequences
NIST	National Institute of Standards and Technology
NUTS	Nomenclature of Territorial Units for Statistics
PAF	Percentage of Affected Population
PNS	Persistent Neurocognitive Sequelae
ppm	Parts Per Million
SF_6	Sulfur Hexafluoride
SFP	Specific Fan Power
SPF	Seasonal Performance Factor
TRY	Test Reference Year
UK	United Kingdom
VOC	Volatile Organic Compound
WHO	World Health Organisation

Samenvatting

Residentiële ventilatiesystemen worden in de literatuur over ventilatie traditioneel slechts oppervlakkig belicht. Door zijn grote aandeel in het totale energiegebruik krijgt de residentiële sector sinds kort echter een centrale plaats in de plannen van de EU voor het beheersen van de klimaatverandering. Omwille van de snelle en succesvolle uitrol van doorgedreven isolatie- maatregelen maken warmteverliezen ten gevolge van ventilatie momenteel het leeuwendeel van de totale warmteverliezen uit in nieuwbouwwoningen, wat aanleiding geeft tot hernieuwd debat over de correcte dimensionering van de ventilatie in woningen. Ondertussen hebben vraaggestuurde ventilatie en warmteterugwinning uit afblaaslucht, twee technologieën die bedoeld zijn om deze ventilatieverliezen drastisch in te perken, een belangrijk en onderling fel bevochten deel van de markt voor residentiele ventilatiesystemen veroverd.

Dit proefschrift behandelt de afweging tussen het verzekeren van een adequate binnenluchtkwaliteit en het verminderen van de ventilatieverliezen die inherent met het ontwerpen van ventilatie is verbonden. Daarbij wordt dieper ingegaan op de effectiviteit van ontwerpstrategieën, van vraagsturing en van warmteterugwinning. Nadat in het eerste hoofdstuk een algemeen kader en een set van ontwerpdoelstellingen worden geschetst, wordt in het tweede hoofdstuk gezocht naar de resultaten die worden behaald in de huidige praktijk. Deze laatste wordt verondersteld besloten te liggen in de van kracht zijnde ventilatie normen in België en de ons omliggende landen. Uit de uitgevoerde analyse blijkt dat grote verschillen bestaan tussen de prestaties van systemen die zijn ontworpen volgens de verschillende normen. Zo komt onaanvaardbare binnenluchtkwaliteit slechts 5% van de tijd voor wanneer volgens de Belgische, Nederlandse of Franse norm wordt gewerkt, terwijl dit bij het toepassen van de Britse of ASHRAE norm 15% bedraagt. Onafhankelijk van de gekozen norm levert mechanische ventilatie steeds een robuuster resultaat, terwijl de prestaties van extractieventilatie sterk afhangen van de luchtdichtheid van de gebouwschil.

Vervolgens wordt in het derde hoofdstuk door middel van de Pareto-optimale prestaties, de waarde van de verschillende types ventilatiesystemen verder onderzocht. Hierbij komt naar voor dat er slechts kleine verschillen tussen de systemen bestaan indien de gemiddelde blootstelling aan CO_2 als criterium wordt gehanteerd. Indien wordt vergeleken op basis van blootstelling aan pieken in CO_2 liggen de resultaten verder uiteen, maar blijven de verschillen relatief beperkt. Extractieventilatie en mechanische ventilatie presteren echter substantieel beter bij het beheersen van overmatige vochtproductie en geurhinder dan natuurlijke ventilatie. Bij dit laatste systeem komen immers min of meer frequent luchtstromingen vanuit de natte ruimtes naar de droge ruimtes in de woning voor. Alle systemen behalen betere resultaten in luchtdichte woningen en tevens is het optimale resultaat voor alle systemen beter dan dat van een lekke woning waarbij geen extra ventilatiesysteem is geïnstalleerd.

Met deze resultaten als referentie behandelt het vierde hoofdstuk het potentieel van vraaggestuurde ventilatiesystemen in woningen. Startend vanuit een analyse van enkele systementypes die verkrijgbaar zijn op de Belgische markt, wordt vastgesteld dat er inderdaad een belangrijk potentieel in deze technologie besloten ligt. Uit metingen in voorbeeldprojecten met extractieventilatie wordt bovendien aangetoond dat dit ook in praktijk haalbaar is. Er kan tot 85% bespaard worden op het ventilatorverbruik terwijl extractiedebieten tussen de 55 en 70% lager uitvallen. Wanneer op basis van simulaties een schatting wordt gemaakt van de totale ventilatie- en infiltratiedebieten valt de besparing op de ventilatieverliezen echter terug tot 15 à 40 %. De analyse wordt daarop uitgebreid om rekening te houden met de veranderende randvoorwaarden die ontstaan door lokale dimensioneringsregels en klimaat. De voorgestelde definitie voor een reductiecoëfficiënt om het potentieel van een specifieke vraagsturing te beoordelen levert hierbij relatief consistente resultaten. Verdere analyse toont uiteindelijk ook aan dat de optimale prestaties van vraaggestuurde systemen tot 50% beter zijn dan die van standaard systemen.

In het vijfde hoofdstuk worden de prestaties van de rivaliserende technologie, de lucht-lucht warmtewisselaar, bij verschillende randcondities geanalyseerd. Daarbij is de omzetting van elektrische energie naar warmte die wordt gehanteerd van doorslaggevend belang. In dit proefschrift wordt gebruik gemaakt van primaire energie, CO_2 emissies, prijs en exergie als basis voor het berekenen van de rendabiliteit van het inzetten van deze vorm van warmteterugwinning. Daarbij wordt duidelijk dat hiermee in de gematigde klimaatzone in Europa geen duidelijk voordeel te halen valt, tenzij het verbruik van de ventilatoren kan worden beperkt. Dit geldt a fortiori in het middellandse zee gebied, terwijl in Scandinavië warmteterugwinning vrijwel steeds lonend is. Opvallend is dat de rendabiliteit van warmteterugwinning terugloopt bij stijgende thermische kwaliteit van de gebouwschil. Samengevat kan, mits het toepassen van goed vakmanschap en doordacht ontwerp, overal in Europa op een batige manier gebruik gemaakt worden van lucht-lucht-warmtewisselaars voor woningventilatie.

Vertrekkend vanuit de vaststelling dat de bewoners het grootste deel van de tijd in de slaapkamers verblijven en dat daarbij de meest gehanteerde criteria voor binnenluchtkwaliteit niet direct toepasbaar zijn, stelt het zesde hoofdstuk enkele van de ontwerpdoelstellingen die in het begin van het proefschrift op basis van de beschikbare literatuur werden geformuleerd in vraag. In een eerste deel wordt op basis van metingen in de slaapkamer en leefruimtes van 81 woningen aangetoond dat de kans om blootgesteld te worden aan onaanvaardbaar hoge CO_2-concentraties in slaapkamers tot tien maal hoger ligt in vergelijking met de leefruimte. De resultaten van een interventiestudie aan de hand van slaap-actigrafie, behandeld in het tweede deel van het hoofdstuk, suggereren echter dat dergelijke blootstelling geen aanleiding geeft tot aantoonbare acute effecten op de slaapkwaliteit. De neiging om te pleiten voor lagere ventilatiedebieten voor slaapkamers die hieruit zou voortvloeien wordt echter getemperd door labo-experimenten, die het voorwerp uitmaken van het derde deel van het hoofdstuk. De resultaten hiervan tonen immers aan dat de specifieke omstandigheden die worden gecreëerd binnen de micro-omgeving onmiddellijk rond een slapend persoon, aanleiding geven tot een verhoging van de blootstelling aan gasvormige polluenten met bronnen binnen deze omgeving ten opzichte van de schattingen die hieromtrent worden gemaakt in courante risico analyses voor consumptiegoederen.

Uit de conclusies van de eerste zes hoofdstukken, zoals die zijn samengebracht in het zevende en laatste hoofdstuk van dit proefschrift, blijkt duidelijk dat de ontwerpstrategieën die momenteel in de van kracht zijnde normen zijn opgenomen aanleiding geven tot sub-optimale prestaties. Anderzijds wordt duidelijk het potentieel van vraagsturing en warmteterugwinning aangetoond om deze prestaties verder te verbeteren. Uit de tegengestelde indicaties met betrekking tot de waardering van de luchtkwaliteit in slaapkamers volgt echter dat er ook nood is aan verder onderzoek op deze materie om tot een sluitende beoordelingsmethode te komen voor residentiële ventilatiesystemen.

Summary

In the literature on ventilation, residential ventilation systems have traditionally received less attention due to their relatively low capacity and complexity. Nevertheless, the large energy saving potential of the residential building sector has made it one of the key targets of climate change mitigation strategies in the European Union. With the rather successful penetration of insulation measures and weatherization, ventilation heat loss now dominates the total heating demand in newly built dwellings, opening a renewed debate on ventilation rates and sizing. Meanwhile, demand control and heat recovery technologies claim to provide substantially better performance than traditional systems. Both have acquired an important but highly competitive share in the market.

This dissertation addresses the performance trade-off between heat loss and indoor air quality inherent to ventilation and focusses on the effectiveness of design strategies, demand control and heat recovery for residential systems with respect to this trade-off. After sketching a context and defining a set of design objectives for residential ventilation in the first chapter, it explores the performance level achieved by the 'state of the art', represented by the design strategies included in a series of contemporary residential ventilation standards in the second chapter. The performance of systems sized in accordance with the different standards proves to be substantially different, with an occurrence of poor perceived air quality in 5% or less of the occupation time for the Belgian, Dutch and French standard, and about 15% for the British and ASHRAE standard. Mechanical supply and exhaust ventilation is shown to be more robust than natural or mechanical exhaust ventilation across all standards and a clear interaction between the performance of mechanical exhaust ventilation and envelope leakage is found.

Subsequently, the effectiveness of the different ventilation strategies is assessed in the third chapter by relating their performance to the Pareto optimal performance, revealing that, considering average exposure to carbon dioxide, optimized mechanical supply and exhaust ventilation concept performs only slightly better compared to mechanical exhaust ventilation, while the latter in turn achieves slightly better performance than natural ventilation. The spread in optimal performance increases when exposure to peak concentrations is considered instead of average exposure. Nevertheless, the differences remained moderate. A more substantial spread in optimal performance of natural ventilation cases with optimal mechanical exhaust and mechanical supply and exhaust ventilation cases is found with respect to exposure to humidity and odours, due to the more frequent occurrence of backdraft from the service spaces to living spaces. All system concepts show superior optimal performance in airtight than in leaky conditions. Additionally, dedicated ventilation systems achieve 30-40% lower exposure at equal ventilation heat loss than leakage as a means of ventilation.

Using these results as a reference, the energy saving potential of demand controlled ventilation is investigated in the fourth chapter. Again starting from the ‘state of the art’ based on a performance assessment of systems available on the Belgian market, a substantial energy saving potential for demand controlled residential ventilation systems is found. This potential is then confirmed in practise in a series of case studies. In the latter, fan power reductions found reached up to 85%, but in a case without frequency controlled fan, this reduction was only 20%. Exhaust flow rate reductions were between 55 and 70%. Taking estimations for adventitious ventilation and infiltration from the simulations into account, however, the estimated heat loss reductions for ventilation were only 15 to 40%. The analysis then expands to the impact of the local context created by ventilation standards or climate and demonstrates that the proposed definition for an energy reduction coefficient renders a relatively robust estimate for the energy saving potential of a specific demand control approach. Finally, further analysis demonstrates that the performance of optimized demand controlled ventilation is up to 50% better than that of continuous flow systems.

The performance of the main competitive technology, heat recovery ventilation, is studied under different boundary conditions in the fifth chapter. It strongly depends on the type of conversion coefficient between electrical energy and fuel combustion for heating that is used. With primary energy, carbon dioxide emission, price and exergy, 4 broadly accepted frameworks for the comparison of these types of energy are used to calculate the profitability of heat recovery ventilation. The results demonstrate that, unless low specific fan power is achieved, for the moderate climate region of middle Europe, natural ventilation, mechanical exhaust ventilation and heat recovery ventilation have no clear advantage over each other as far as operating energy and associated ecologic (CO_2) and economic (Household consumer price) effects are concerned. The choice between the different systems should be made based on building specific characteristics, investment and maintenance cost. In the Mediterranean basin, heat recovery ventilation can only be operated profitably in low pressure drop and low fan power systems, while it is advantageous under virtually all tested conditions in the Scandinavian region. In contrast to low fan power, high thermal building performance tends to create unfavorable conditions for heat recovery ventilation. Overall, heat recovery ventilation can be made profitable all over Europe with regard to primary energy, carbon dioxide emissions and household consumer operating energy cost by achieving realistic best practice low specific fan power.

The sixth chapter of the dissertation revisits the design objectives for residential ventilation by highlighting the dominance of the exposure in the bedrooms in the assessment of residential indoor air quality as well as the poor validity of available performance indicators and assessment methods in this environment. In the first section of the chapter, carbon dioxide measurements in the living room and bedroom of 81 cases confirm the observations made in simulations that the probability of exposure to carbon dioxide concentrations generally accepted to be an indication of undesirable indoor air quality is up to 10 times higher in the bedrooms than it is in the living room. Although the results from the first section are cause for some alarm, the occupants are asleep most of the time spent in the bedroom. The second part of the chapter presents the results from an intervention study using carbon dioxide measurements and sleep-actigraphy, from which one could conclude that, although indoor air quality conditions in the bedroom are less than optimal considering traditional assessment parameters based on perceived air quality and bio-effluents, there seem to be little to no obvious acute repercussions on sleep quality associated with these exposure levels. An experimental assessment of the impact of typical sleep micro-environments on the exposure to near field sources of gaseous pollutants presented in the third part of the chapter found that these are generally higher than the estimates based on well mixed conditions used in exposure risk assessments for consumer products. This, combined with the fact that the concentrations measured in the first section exceeded the level considered a threshold for health effects in just under 25 % of the occupancy time in the bedrooms, warrants great caution in lowering design flow rates in response to the results of sleep intervention study.

Combined, the conclusions from the different chapters summed up in the seventh and last chapter, clearly demonstrate that current practise in residential ventilation is suboptimal, while a substantial improvement in performance can be achieved with demand control and heat recovery technology. The questionable validity of the common design objectives for exposure in bedrooms, the most intensively used space of a dwelling, however stresses the need for further research in this area to achieve a comprehensive assessment methodology for residential ventilation systems.

1

Introduction

Ventilation is a hot topic. With the series of recent crises, in which the strain on energy supply and reserves has been a constant, popular attention has been drawn to it due to a series of problems with indoor air quality stemming from unthoughtful insulation and weatherization interventions during the first energy crisis in the 70-ies, reports in the media about occupants getting sick because of their ventilation system and the general reluctance to pay for a complicated ventilation system. Nevertheless, the public knowledge about the specific purpose of ventilation and the parameters involved in its performance is limited.

This is reflected in generally poor quality of residential ventilation systems, since, for that type of systems, the building occupant is the central stakeholder and actor, due to budget constraints that usually prohibit the involvement of HVAC engineers or tailor made designs.

This first chapter will focus on adverse effects stemming from airborne pollution, in other words: why do we need fresh air?, followed by a list of common sources of indoor air pollutants. In the second part of the chapter, different risk management strategies will be explored that can be used to avoid these effects and finally, as a conclusion, an outline of the remainder of the dissertation will be given.

1.1 Airborne Pollutants

The very first question in constructing a framework to assess the performance of ventilation systems obviously deals with the main purpose of ventilation. What do we hope to achieve by ventilating? And why? Since ventilation is nothing else than replacing the air in a room with air from outside the room, its effect can be no other than making the indoor air more like the outdoor air. This can only be desirable if indoor air and outdoor air are different and that there are either positive effects associated with exposure to outdoor air or adverse effects associated with exposure to indoor air. In other words, if indoor air is polluted.

A large number of pollutants is typically found in indoor air and new pollutants are introduced at an astounding rate [1]. The properties of these pollutants depend on their physical nature, in which the two main categories are gases [2] and particulates [3]. The distribution of both is governed by the general airflow field in interaction with local sources and sinks. In addition to that, gravity will cause particulates to settle. This effect is proportional to the size of the particulate [4], with the smallest particulates acting more or less like gases.

Gases can be both inorganic [5] and organic [6] in nature, while particulates usually have complex chemical compositions and often contain biological agents such as bacteria and viruses [7]. In both categories, the abundant presence of other pollutants and boundary conditions such as temperature and humidity levels can affect the stability of the pollutants [8] and trigger secondary pollutant formation in the air itself [9-13].

1.1.1 Effects

The, mostly adverse, effects of airborne pollutants have been the topic of numerous studies over the last decades. They can be divided into three main categories: effects on health, effects on productivity and effects on comfort. With respect to health effects, there is a large grey zone as to what effects should be included or not. In the overview of the reported effects below, all effects that relate to short term direct effects such as headache, dry or burning sensation etc., usually grouped under the term ‘Sick Building Syndrome’, are listed as comfort effects.

Health effects

WHO has prioritized a number of inadequate housing conditions based on their associated burden of disease, expressed in Disability Adjusted Life Years (DALY’s) [14], including exposure to airborne agents such as dampness, radon, second hand smoke, carbon monoxide and formaldehyde. A summary of the burden of disease they attribute to inadequately handled housing conditions related to indoor air is provided in Table 1.1. Their “WHO Guidelines for indoor air quality: selected pollutants” [15] associates benzene with acute leukaemia and attributes genotoxic properties to it. Furthermore it relates carbon monoxide exposure to increased ischaemic heart disease symptoms, naphthalene to respiratory tract lesions prone to inflammation and nitrogen dioxide exposure to increased susceptibility to respiratory infection. Lung cancer prevalence is increased along with exposure to radon and polycyclic aromatic hydrocarbons. Tri- and tetrachloroethylene are associated with kidney failure and carcinogenicity (the former). Formaldehyde evokes sensory irritation and is carcinogenic in high concentrations.

A series of epidemiological studies started in Scandinavia and expanding to other countries established a clear link between signs of dampness, water problems and mould in buildings and the prevalence of asthma and other respiratory and allergic symptoms. [16-23]. Some mould species have been shown to produce allergenic proteins, especially when growing on cellulose rich substrates [24]. A correlation between asthmatic symptoms and exposure to mites was also found [25]. The potential annual savings from reduced asthma and allergic symptoms by improved indoor environments in the US is estimated to be between 5 and 18 billion USD [26].

Pollutants stemming from indoor smoking lead to shortness of breath [27] in non-asthmatic non-smokers and increase asthmatic symptoms [28, 29]. Wheeze in asthmatic children is also correlated with polycyclic aromatic hydrocarbons exposure [30]. The latter are also a cause of oxidative stress on DNA sequences [31].

NO_x concentrations are associated with hospitalisations for respiratory [32, 33] and cardiovascular disease [34] as well as increased inflammatory biomarkers [35].

Ventilation rates are negatively correlated with the prevalence of inflammation, respiratory infections, asthma symptoms, short term sick leave and allergic manifestations [36, 37]. In these studies, ventilation rate is used as a proxy for pollutant concentration. The validity of this assumption will be discussed later.

Epidemiological research also suggests a link between both short- [38] and long-term [39, 40] increased ozone concentrations and mortality in sensitive populations. Others suggest that concentration variability is a better metric [41]. A major contributor and confounding factor in these studies are fine and ultrafine particulate matter concentrations [38], which in themselves have several adverse health effects such as stroke and ischaemic heart disease [34], increased blood pressure [42], lung cancer mortality [43] and increased general mortality [44-46], specifically in neonates [47]. Increased fine particulate concentrations were also shown to negatively impact lung functions such as peak expiratory flow rate, forced vital capacity and forced expiratory volume [48], as well as correlate with increased hospitalisation with short term respiratory complaints [33]. Ultra-fine and fine particulates proved to be genotoxic and to cause inflammatory responses [35, 49, 50]. Particulate exposure is considered the highest contributor to adverse health effects in the epidemiological part of an on-going effort to create health based ventilation standards [51].

Extremely high carbon dioxide concentrations (>25%) lead to acute respiratory problems, nausea and eventually death [52], while carbon monoxide is lethal in very low doses and remains a continued public health concern [53].

Finally, the presence of bacteria and viruses in the indoor environment, often produced by the occupants, are a vector for the spread of a variety of diseases [54, 55]. Legionella infection is a persistent cause of acute pneumonia, with a low survival rate [56].

Table 1.1 Summary of exposure / risk relationship (RR), percentage of affected population (PAF) and environmental burden of disease (EBD) associated with selected indoor air pollutants as published by WHO [14]

Exposure	Health outcome	RR	PAF	EBD
Mould	Asthma deaths and DALYs in Children (0-14 years)	2.4	12.3 %	45 countries of WHO European Region: Deaths 0.06/100 000 DALYs 40/100 000
Dampness	Asthma deaths and DALYs in Children (0-14 years)	2.2	15.3 %	45 countries of WHO European Region: deaths 0.07/100 000 DALYs 50/100 000
Lack of smoke detectors	Injury deaths and DALYs (all ages)	2.0	2 – 50 %	WHO Euro Region deaths 0.9/100 000 DALYs 22.4/100 000
Crowding	Tuberculosis (TB)	1.5	4.8 %	WHO Euro B and C subregions: TB cases 3.3/100 000 deaths 0.8/100 000 DALYs 17.6/100 000
Radon	Lung cancer	1.08 / 100 q/m3	2 – 12 %	3 countries: F: 2.1/100 000 deaths G:2.3/100 000 deaths C:3.2/100 000 deaths
Residential SHS	LRI, asthma, heart disease and lung cancer	1,2 – 2,0	0.6 – 23 %	WHO Euro Region deaths 7.3/100 000 DALYs 80.7/100 000
Indoor CO	Headache, nausea, cardiovascular ischaemia/insufficiency, seizures, coma, loss of consciousness, death	Case-fatality rate 3%; DNS/PNS incidence 3-40%	50 – 64 %	WHO Euro A subregion Persons with DNS /PNS 0.03 /100 000 deaths 0.03 /100 000
Formaldehyde	Lower respiratory symptoms in children	Odds Ratio (OR) 1.4	3.7 %	WHO Euro A subregion: 0.3 to 0.6% of wheezing in children
Indoor solid fuel use	COPD, ALRI, Lung Cancer	1,5 – 3,2	6 – 15 %	WHO Euro Region ALRI in children < 5y deaths 16.7/100 000 DALYs 577/100 000 COPD in adults ≥ 30y deaths 1.1/100 000 DALYs 19.3/100 000

Productivity effects

A number of independent studies have shown a positive relation between students' academic performance, across different lesson subjects, and the ventilation rate in the class room, both on intervention studies [57, 58] as in epidemiological studies [59, 60]. Particle filtration in the classroom did not have a similar effect [61].

As can be seen in Figure 1.1, similar positive effects of increased ventilation were observed for office workers [62], both for simple, repetitive tasks [63] and more complex strategic decision making [64]. Although the effect is rather small, a few % of productivity gain, the economic value of that increase in productivity is much higher than the cost of increased ventilation [65, 66]

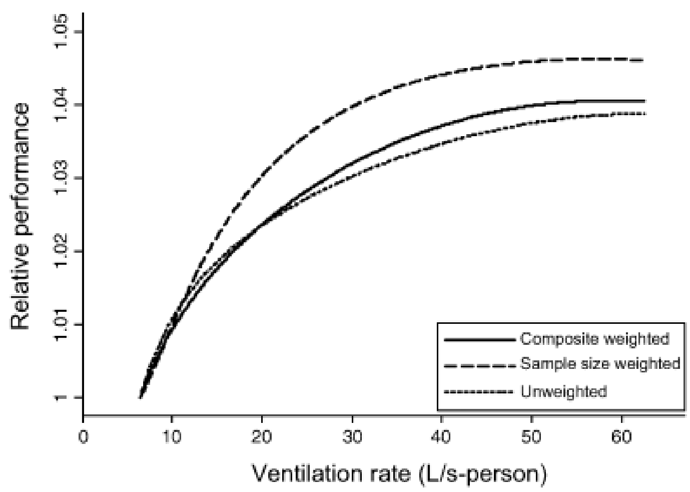

Figure 1.1 Relative performance of office workers in relation to the reference values 6.5 l/s-person vs. ventilation rate reported in [62]

Comfort effects

Ventilation flow rates, as a proxy for pollutant concentrations, have been shown to improve the perceived quality of the indoor air [67, 68] as well as a number of acute physiological complaints such a dry eyes, blinking rate, dry throat, head ache etc. usually grouped under the term 'sick building syndrome'[69]. The correlation fades at flow rates higher than 25 l/s/person.

Some of these symptoms were attributed to specific pollutant types, such as human metabolites [70] or volatile organic compound (VOC) emissions from common building materials [71, 72].

Occupants have reported discomfort at excessive humidity levels, especially in hot conditions [73, 74], and there are some early indications that high pollution loads have an impact on the quality of sleep and dreaming [75, 76].

Other parameters such as temperature, acoustics and air speed, definitely influence occupant comfort, but are not directly related to the reduction of pollutant levels, which is considered the primary goal of ventilation in this chapter, and are therefore not discussed further.

1.1.2 Sources

Within the indoor built environment, the main sources of the pollutants associated with adverse effects can be grouped into 3 categories: the building itself, and more specifically the building materials used in it, the activities of the building occupants inside the building in general or in specific locations of the building and pollutants coming from the outside environment penetrating into the indoor environment. In the following paragraphs, these categories are each discussed more in detail.

Materials

Part of the discomfort caused by volatile organic compounds in offices can be traced back to emissions from building materials [71, 77] [78-81] and typical plastic finishes of appliances [82] [2]. Even materials concealed behind finishing materials can emit pollutants into the space [83]. Wood based panels have become notorious for emitting formaldehyde [72, 84-86]. In addition to the release of their primary chemical compounds to the space, ozone deposition on building materials can trigger an important release of secondary pollutants [11, 87]. Desorption of previously sorbed chemicals similarly cause space surfaces to act as secondary sources [8, 88]. Stony materials and rock bed terrain are known sources of radon [89].

Especially important for the residential context, mattresses have been shown to be important sources of VOC's [90-92] or SVOC's [93] and act as incubators for house dust mites [94-96].

Typical for material emissions is that they show large initial source strengths, decreasing over time [97-100]. The emissions from building materials are not only dependent on the (chemical) composition of the materials, but are also affected by the boundary conditions in the space. They are sensitive to temperature increases and especially to increases in relative humidity [94, 101-105].

Hygroscopic materials, especially cellulose based ones, are prone to the development of fungi in long lasting high humidity conditions [106-111].

Activities

One of the most significant sources of a broad range of indoor pollutants is fire in all its forms, Be it tobacco smoking [28, 29, 112-117], the use of cooking [118-124] or heating [125-130] appliances or the burning of candles and incense [131, 132], all of them are significant sources of particles, volatile organic compounds, nitrogen oxides and carbon monoxide. In developing countries, these activities are the primary sources of indoor pollution and especially women are exposed to them [133, 134]. Although emissions from combustion engines are usually considered outdoor sources, the tendency to have attached garages introduces typical gasoline related pollutants such as MTBE indoors [135, 136].

In addition to the kitchen, the bathroom is a room where a lot of pollutant producing activities are concentrated, ranging from bathroom use (odours) and showering (moisture) to the intensive use of cosmetics and cleaning products (VOC's)[123, 137-141]. Similar to material emissions, emissions from household products may trigger secondary chemical reactions [142, 143].

Building activities [144] and painting [145] are often large temporary sources. In office environments, large amounts of particles are released by laser printers [146-149]. The operation of building services, such as a HVAC unit, can resuspend previously deposited particles [150-152]. Ill-conceived or ill-maintained building services can also become sources of biological contaminants such as odours, VOC's, mould spores [153, 154] and legionella bacteria [155-157].

The occupants themselves are sources of carbon dioxide, moisture, bio effluents, bacteria, viruses etc. [152, 158-164]. Due to the movement of the occupants, these pollutants are usually dispersed throughout the entire building [165-167] and often continuously resuspended [168]. Particularly high exposures can be found in episodes of (unintended) inter-occupant contact, such as cough or sneeze [169]. Self-evidently, overcrowding increases the intensity of these sources [113].

Outdoor environment

Although the basic idea of ventilation is that the outdoor air is less polluted than the indoor air, some pollutants are mainly produced in the outdoor environment, eg. traffic related emissions, emissions from industrial processes etc. Other, even typically indoor, pollutants may not have sources in a specific building, but are found in the ambient air [170]. These pollutants are therefore brought into the indoor space through air exchange with the outdoor environment [171, 172]. Typical determinants of outdoor pollution levels are of course the proximity [173] and timing of the outdoor sources [174, 175], such as locations near busy traffic on rush hours [176, 177] or dense cities [178], but modifying factors such as meteorology [179] and the proximity of other buildings can have a significant impact on indoor concentrations [180].

1.2 Exposure Risk Management Strategies

A number of approaches can be adopted to avoid the adverse effects caused by exposure indoors. Such risk management strategies include: reducing the strength of known sources, local exhaust around known sources, eliminating airborne pollutants by filtration or catalytic activity, dilution of pollutant concentrations by bulk ventilation or alerting occupants in case of alarming concentrations.

1.2.1 Source Control

The most effective way to reduce exposure is preventing the release of the pollutants in the indoor environment. This approach can be applied effectively for a wide range of sources. Through the careful selection of components, emissions from common building materials can be reduced by a factor of 10 [181-183]. Regular duct cleaning prevents resuspension of accumulated dust by HVAC operation [184, 185]. Resuspension can also be avoided by careful surface design [186]. The selection of well-designed cooking appliances and their adequate use can reduce particle and carbon monoxide exposure by more than half [187, 188]. Maintaining elevated water temperatures and avoiding stagnation throughout the distribution system stops Legionella development and kills viable legionella bacteria present in the system [189-191]. Self-evidently, banning polluting activities such as indoor smoking, candle burning etc. eliminates the associated primary sources [117, 192]. The entry of outdoor pollutants and radon can be mitigated by improved airtightness [193] and pressure management [186, 194-196].

1.2.2 Local Exhaust Ventilation

If a polluting activity is the purpose of the space, effective source control through banning the activity is not an option. Local concentrated exhaust, however, can in such a case prevent the spread of the produced pollutants to other parts of the space or building [197, 198]. In residential settings, the application of this principle to kitchens and bathrooms is widespread and very effective [199, 200].

1.2.3 Air Cleaning

In case the sources are of a more diffuse nature or can't be localised easily and in case of outdoor sources, cleaning the air of the pollutants present in it can be considered. This can achieved by actively filtering the air [42, 201-207], either locally or on building scale, although the technology available for this is usually only able to target a specific type of pollutants.

Organic pollutants can be targeted by passive degradation at catalytically active surfaces. Concerns with this technology include long term effectiveness, the accessibility of the bulk air to these surfaces in natural convection regimes and formation of hazardous secondary products. Most products also require incident UV radiation to activate the catalytic effect [208-217].

1.2.4 Dilution Ventilation

A further strategy to reduce indoor pollutant levels is diluting the indoor air with fresh air until acceptable concentrations are reached [100, 114, 218-220]. If the fresh air is targeted to the occupant, this is basically the inverse of local exhaust. This approach can even be adopted in outdoor environments [221]. A precondition for this strategy is of course the availability of fresh air with lower pollution levels than those present in the indoor air.

1.2.5 Alarms

Certain acute airborne health threats, such as carbon monoxide poisoning, require to alert and evacuate the occupants immediately. In such a case, the sensor has to target the right pollutants [222, 223] and be well positioned to detect the threat as soon as possible [224]. In addition to that, signal processing and integrated modelling can be used to locate the pollution source, allowing for instance to select an appropriate evacuation scenario [225, 226].

1.3 Ventilation: The Back-up Plan

The order of the mitigation strategies listed above reflects their effectiveness in reducing the risk of incurring adverse effects from exposure to airborne pollution indoors. The risk chain extends from the actual intake of a pollutant by the occupant to the concentration of the pollutant in the indoor air and eventually to the sources of these pollutants. The earlier in this chain the risk is eliminated, the better. Therefore, source control, or in case of unavoidable sources, local exhaust should always be the priority. Some sources, especially those related to the biological nature of the occupants, can't be reduced with source control and are moving through the building. In that case, only strategies that reduce the concentrations of the pollutants in the indoor air, namely ventilation and air cleaning can and should be adopted. Even if such concerns are not applicable, the emissions of a large array of sources, such as materials and household products, can be significantly reduced, but not eliminated. Adequate ventilation and/or air cleaning are therefore indispensable in any risk management scheme [227]. Although alarms are the final fail safe, adequate ventilation and/or air cleaning will also provide sufficient redundancy in case unexpected sources occur. The rest of the chapter will only focus on ventilation.

Due to the broad range of pollutants and sources that can manifest themselves in indoor environments, proposing universal or general sizing rules for ventilation is an ambitious goal. This is clearly reflected by the number of different performance criteria proposed in ventilation standards [228-230] and the large spread in observed ventilation rates applied in Europe [231]. In general, minimal ventilation rates of about 0.5 ACH are recommended for houses, about 1 ACH for the occupied area of an office space and about 0.1 ACH for weakly polluted unoccupied spaces.

Provided clean air is available, the risk of adverse airborne exposure effects is best mitigated by large volumes of fresh air. In most climates, however, maintaining thermal comfort indoors within acceptable limits associates the indoor environment with an energy cost, introducing a trade-off in ventilation sizing [232, 233]. For any level of risk, a corresponding expense for energy has to be accepted [234]. In the total risk-energy trade-off, a number of modifying factors are present, concentrated in three media: the ventilation system, the indoor space and the occupant. Each of these will be discussed further in the following paragraphs. Figure 1.2 illustrates the full trade-off.

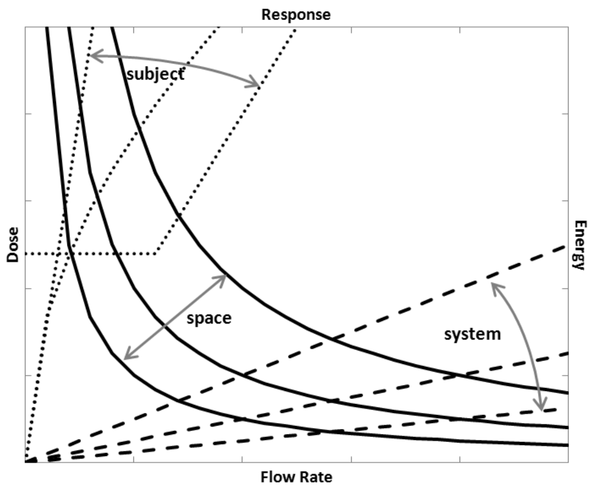

Figure 1.2 The energy-response relation is influenced by system, space and subject parameters and faces an inherent trade-off.

1.3.1 Subject

Examples of adverse effects associated with the intake of airborne pollutants were discussed in section 1.1.1. For any of these pollutants, there is a probabilistic dose-response function that determines whether a certain dose will be likely to trigger the effect or not [235-238]. The form of these functions is dependent on the specific pollutant. It can be both linear or non-linear, include a 'no effect' threshold or not [239, 240] and be modified by specific sensitivity of the subject such as age, allergy, pregnancy... Some examples of dose-response functions are shown in dotted lines in Figure 1.2.

1.3.2 Space

A chain of processes in the ventilated space create the conditions that allow getting from a flow rate, delivered by the ventilation system, to the dose inhaled by an occupant [241, 242]. First of all, the layout and size of the air supplies and exhausts [243, 244], the production of buoyancy forces in the space and other momentum sources create a specific and usually unsteady flow field in the space [245, 246]. Specific approaches include displacement ventilation, creating a more or less stable zone of fresh air in the occupied zone [247], and personal ventilation, which directs the fresh air directly at the occupant's breathing zone [248, 249]. The geometrical distribution and type of sources in the space, as was elaborated on above, will interact with the flow field and this results in a spatially distributed concentration and associated ventilation efficiency for each of the pollutants [250-252]. The presence, behaviour and whereabouts of occupants in the space [253-255] will both influence the flow field, due to their movements and the thermal plume generated by their metabolic heat output [256, 257], and determine their exposure to the pollutants. Flow conditions in the micro environment of the breathing zone, again influenced by movement and thermal plume, and personal factors such as breathing rate will then determine the intake and dose [258-260]. The closer the flow rate – dose function is to the origin in Figure 1.2, the smaller the trade-off is and the better the performance of the system. The complete set of modifying factors, including ventilation, influencing the relation between source strength, rather than the ventilation rate, and dose is integrated in the 'intake fraction' concept [261, 262].

1.3.3 System

A last set of modifying factors is imbedded in the ventilation system. Its characteristics will determine the energy required to deliver a pre-defined ventilation flow rate. This energy contains both the energy required to condition the air to acceptable temperatures and that to move the air [263, 264]. Other associated costs are investment costs and maintenance [265].

The layout of the supply and exhaust vents not only influences the flow field in the ventilated space, but can also have a significant impact on the duct configuration and thus on the total pressure drop in the system. The fan power required to provide the desired flow rate in fan driven systems is a cubic function of the latter and determines the achievable flow rates in systems relying on natural driving forces such as wind and buoyancy. The performance of systems relying on natural driving forces is less robust in terms of flow rate stability due to the variability of these forces [266-268]. Adding components such as pressure regulated trickle ventilators [269, 270], (solar) chimneys [271, 272] and wind cowls [273-275] helps to achieve more stable air flow rates.

Fan supported ventilation provides more robust indoor air quality, but require fan energy [276, 277]. This introduces the conversion between heat loss and electrical power consumption as an additional dimension in the trade-off. The choice of a conversion framework, eg. energy, primary energy [278, 279], carbon dioxide emission [280, 281] or exergy, has a significant effect on the performance of the fan supported systems [282].

The implementation of filter technology is virtually exclusively reserved for fan supported systems, but again requires additional fan energy [283, 284]. Fan supported systems also require adequate commissioning, cleaning and maintenance, as well as a minimum of occupant education to function properly. A large number of installed systems fail in one of these aspects [285, 286].

In this dissertation, the potential of air movement to create thermal comfort conditions is not discussed. The supply air is assumed to be at the minimal temperature required to eliminate the heating demand. During the heating season, when the outdoor air temperature is below this threshold, the supply air has to be heated to achieve this. Mutatis mutandis, the same can be said for the cooling season. Modifying factors determining the amount of energy required to do so obviously include climate conditions and the thermal performance of the building [282] as well as the extent to which the energy contained in the exhaust air can be reused to heat the supply air. Most heat recovery systems require additional energy input to function. Their performance is discussed in the following section dealing with specific types of ventilation.

1.4 Ventilation: Strategies and Technology

Ventilation, as a means of risk management, can be achieved by simple low-tech solutions relying on leakage and window opening by occupants, or by mechanical supply and exhaust systems that use fans to move the air where it is needed, or by any conceivable compromise between these extremes. In this section, the merits of a few of these options are discussed.

1.4.1 Leakage

Traditionally, occupants have relied on infiltration of outdoor air through the building envelope to maintain acceptable contaminant concentrations indoors [287]. Assuming a required air change rate of 0.5 ACH and an average available driving force due to wind and buoyancy of 1.5 Pa, a leakage rate of 10 ACH during a pressurization test at 50 Pa [288] is required to accommodate this. In the US, estimates show that about 50% of the dwelling achieve a median air change rate of 0.5 ACH due to infiltration [289], although mean values in traditional construction in Europe are smaller, at about 0.1-0.2 ACH [290-292].

Since the available driving forces are variable and leakage is usually concentrated around details in the building envelope [293, 294], the distribution of the fresh air in the dwelling is not necessary correlated with the needs. This reduces the overall efficiency of air leakage as a ventilation strategy, especially when compared to mechanical supply and exhaust.

Due to the intensification of energy performance requirements for new buildings, the building industry is moving into more airtight construction, favouring more efficient types of ventilation [295]. This is typically achieved by reducing the number of joints in the construction [296]. 10 ACH at 50 Pa is a typical value for older construction [297, 298] and mild climates [299, 300], but high performance buildings are up to 10 times more airtight. This is clearly visible in time evolution of air tightness in Belgium shown in Figure 1.3.

Draft problems, common in leaky construction, are an additional driver for more airtight construction [301]. Therefore, leakage rates are often reduced considerably in refurbishments of older buildings, along with the installation of more efficient ventilation systems [141, 302].

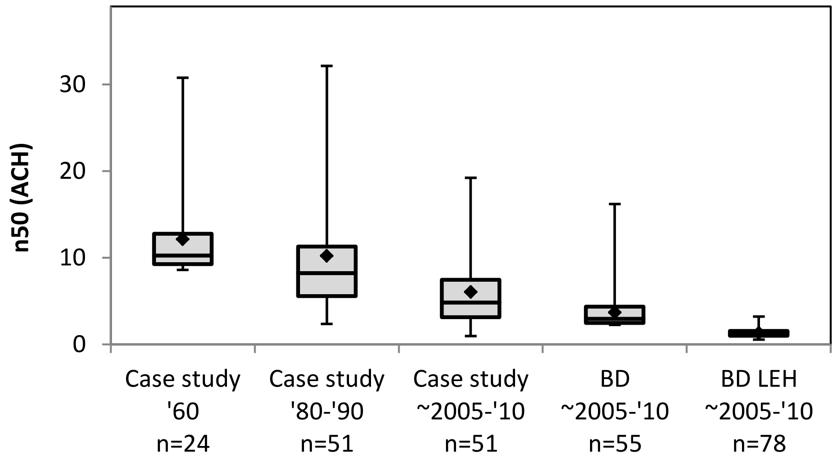

Figure 1.3 Evolution of leakage rates (n_{50}) for single family houses in Belgium. Boxplots for social housing from the sixties ('Case study '60'), standard construction from the early nineties and after 2000 ('Case study '80-'90' and 'Case study 2005-10'), frontrunners ('BD') and Low-Energy Houses ('BD LEH') [303]

1.4.2 Airing

Another traditional ventilation strategy is the opening of windows. Although this usually generates large flow rates of 4-20 ACH [304, 305], especially if cross ventilation is possible [306, 307], and is therefore very effective to evacuate acute high pollution loads, people tend to close the windows when they are present due to draft, especially when the outdoor temperature is low.

In a sense, windows are opened to prevent overheating rather than to reduce indoor air pollution [308-310]. Windows are also frequently closed, despite high indoor pollutant concentrations, because of acoustical discomfort [311]. The combination of these factors leads to the conclusion that the possibility of airing a building by opening windows is a necessary feature to restore the indoor air quality to acceptable levels within a short time span in case of acute high pollution loads, but is less appropriate as a ventilation strategy during occupancy.

1.4.3 System Sizing

From the discussion of leakage and airing as ventilation strategies, it is clear that both of these strategies fail to achieve good ventilation efficiency. In both cases, there is a mismatch between air supply and source strength or occupancy. These drawbacks are mitigated by the conception of dedicated continuous flow ventilation systems. As will be demonstrated in Chapter 3, in optimal conditions, they achieve much better indoor air quality for a given ventilation heat loss than ventilation through leakage. There is, however, no consensus about the design and sizing of such systems. This is reflected in the large differences found in the requirements put forward in ventilation standards [231, 312, 313], resulting in a large spread in performance [314, 315]. Suboptimal sizing self-evidently reduces system performance [316].

Fresh air is usually supplied in the heavily occupied spaces of the dwelling such as living room and bedrooms, and extracted from the dwelling in spaces where more polluting activities take place, e.g. kitchen and bathroom. Most standards distinguish between 4 types of dedicated continuous flow residential ventilation systems (see Figure 1.4 for a graphical representation). In a natural ventilation system, only carefully considered openings and shafts are added to the building envelope and the operation of the system relies on natural driving forces such as thermal buoyancy and wind pressure. In case these driving forces are not sufficient or too unstable, fans can be added to generate the required flow rates. Depending on the location where fans are added, mechanical supply ventilation, mechanical exhaust ventilation and mechanical supply and exhaust ventilation can be distinguished. These systems are coded 'B', 'C' and 'D', respectively, in the remainder of this dissertation, in accordance with the nomenclature used in the Belgian standard. Based on the same classification, natural ventilation systems are labelled A.

Roughly, mechanical supply and exhaust ventilation dominates the market in cold climates [182], mechanical exhaust ventilation is most prevalent in moderate climate regions [317] and natural ventilation is the dominant strategy in mild climates.

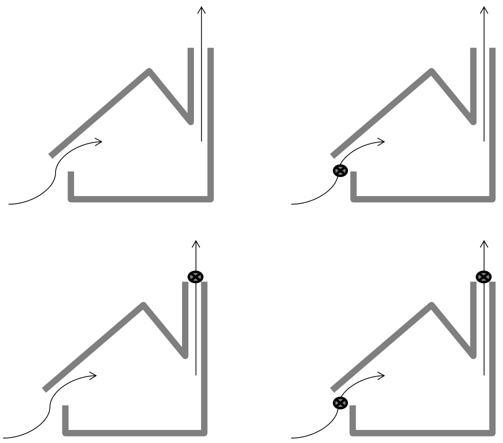

Figure 1.4 Four common approaches to dedicated continuous flow residential ventilation systems: natural ventilation (upper left), mechanical supply ventilation (upper right), mechanical exhaust ventilation (lower left) and mechanical exhaust and supply ventilation (lower right)

1.4.4 Ventilation Heat Recovery

Fan supported systems on the one hand, allow for the implementation of a range of heat recovery technologies, while on the other hand they require fan energy to operate. The trade-off between both, taking into account some conversion between heat loss and electrical power, frames the effectiveness of the heat recovery technology in question. The amount of heat loss that can be recovered is, next to the performance of the specific technology, determined by the climate and building energy performance [318-320]. Although some progress is made on the development of heat recovery systems for systems relying on natural driving forces, their performance and practical feasibility is still under debate [321].

The two most widespread heat recovery technologies available are air to air heat exchangers [322-324] and exhaust air heat pumps [325-328]. The thermal effectiveness of commercially available air to air heat exchangers reaches up to 80 % [329, 330]. Some of these systems include enthalpy exchange [331, 332]. Due to the small diameters within these heat exchangers, filtering is usually necessary to prevent excessive fouling and the associated loss in performance [333].

1.4.5 Demand Controlled and Hybrid Ventilation

By modulating the ventilation flow rate to the pollution load, thus providing ventilation if and when needed, the ventilation heat loss can also be considerably reduced [334]. In order to do this efficiently, sensors are usually used to assess the pollutant load [335], either directly [336] or through the bias of a proxy, such as occupancy [337, 338] or a tracer gas [339, 340].

Hybrid ventilation adopts the same strategy to reduce fan energy: fans are only operated if available natural driving forces are insufficient to provide the required ventilation flow rate [341, 342]. The latter can of course be demand modulated, combining hybrid fan technology with traditional demand control [343].

Most research is focussed on the technical issues of the control strategy [344-346], though the application of the technology has been studied in residences [347-349], offices [350, 351] and schools [352]. In each of these settings, a considerable ventilation heat loss reduction potential (20-50%) was reported.

1.5 Problem Statement & Research Objectives

As was discussed in section 1.3, ventilation as a risk management strategy for exposure to indoor air pollution is faced with an inherent energy cost. The first wave of energy conservation interventions in buildings, ushered into existence by the 1970’s oil crisis, considerably reduced the amount of fresh air infiltration through improved airtightness of newly built dwellings and intensive weatherisation campaigns. As an unintended consequence of this, the incidence of indoor mould problems peaked and reports on high prevalence of occupants complaining of a wide variety of symptoms or physical discomfort, jointly baptised ‘sick building syndrome’[65, 353-356], emerged. Political action in the aftermath of the reports on the consequences of inadequate ventilation saw to the introduction of ventilation requirements in building codes all over the western countries.

The continued scientific interest in these emerging problems was the basis for the indoor environmental science, a fast growing field over the last decades. In this chapter, we have first discussed the possible effects of exposure to airborne pollutants. Health effects from specific pollutants with common indoor sources include increased prevalence of cancer, cardiovascular disease and asthma as well as a long list of acute symptoms. The state of the art within the field has demonstrated positive correlations between indoor air pollution and human health, comfort and productivity. As people spend about 90% of the time indoors [357, 358], the reduction of these effects is essential. For occupant health, fresh air flow rates below 25 l/s per person for offices or building air change rates lower than 0.5 are associated with higher prevalence of symptoms of sick building syndrome and allergies respectively [36]. With respect to comfort, flow rates below 7 l/s per person are considered to result in unacceptably poor perceived air quality in European ventilation standards [228, 230].

Subsequently, ventilation was discussed as one of the possible risk management strategies to minimize these effects. If the exposure risk is well managed, health effects are effectively averted by other means, and only performance or comfort effects remain as design parameters for ventilation systems. Again, if adequate source control measures are taken, the latter are only related to exposure of the occupants to their bio-effluents. Since carbon dioxide is the most commonly used proxy for this exposure, exposure of the occupants to carbon dioxide will be used as the main design parameter for effect minimization in this dissertation.

In the EU, space heating accounts for about 26% of all final energy consumption [278, 359]. Since energy performance criteria are being tightened and ventilation including leakage represents about 50% of the total heat loss in well insulated buildings, the ‘right’ flow rate and the corresponding sizing rules are at sixes and sevens. Keeping the problems that surfaced after the reductions in flow rate made during the 1970’s energy crisis in mind, this debate should be based on a comprehensive analysis of the performance of residential ventilation systems.

Although a number of approaches to residential ventilation exist, as was shown in the previous section, there is little objective data available on the relative performance of these different approaches with respect to both energy use and resulting indoor air quality, while building codes require owners to choose and install one. This dissertation addresses this particular issue.

The performance of ventilation system approaches was presented as a trade-off between ventilation heat loss and effect minimization. To be able to compare them, an assessment method that is able to account for both needs to be selected. This topic will be treated in chapter 2 of this dissertation. In chapter 3, the selected method will then be used to optimize existing design strategies for residential ventilation. Finally, heat recovery by air to air heat exchangers as well as demand control were presented as promising technologies that allow to achieve equivalent indoor air quality at lower ventilation heat losses. Chapters 5 and 4 of this dissertation will elaborate on their respective performance. The 6th chapter will critically reflect on the results presented in the previous chapters to detect questions open still and possible means for further improvement in the performance of residential ventilation systems. A lack of understanding of bedroom indoor air quality will be identified as a major limitation for current performance metrics for residential ventilation and will therefore be the focus of the chapter.

2

Standards and Performance

Sizing rules put forward in different residential ventilation standards lack uniformity in both methodology and resulting design flow rates. Additionally, mere comparison of design flow rates is case sensitive and, due to effects of infiltration, adventitious ventilation and occupancy, ill-suited to assess performance of residential ventilation systems with regard to the achieved indoor air quality and the associated energy cost in terms of heat loss.

This chapter explores the ‘state of the art’ of residential ventilation by investigating what kind of performance is achieved by applying the design rules proposed in a number of ventilation standards to dwelling geometries through multi-zone simulation models. It first introduces the models and assessment parameters used for these simulations. These will also be applied in the next 2 chapters.

With the proposed methodology, mechanical exhaust ventilation systems are dealt with first, as their system layout is rather straightforward and therefore well described in most ventilation standards. This performance assessment of residential mechanical exhaust ventilation systems uses five common dwelling typologies and the sizing rules put forward in the Belgian, British, Dutch, French and ASHRAE residential ventilation standards. The results of this section have been published in Building and Environment [315].

The analysis is then expanded to that of both natural en mechanical supply and exhaust. In this part, the ASHRAE standard is replaced by the German standard because of both the poor performance of its design rules found in the first part of the chapter and the lack of a well described system layout for natural and mechanical supply and exhaust systems.

2.1 Context

As a reaction to the problems with indoor air quality that occurred after the 1970's oil crisis, as was discussed in the first chapter, ventilation standards were established in most western countries. Unfortunately, this did not happen on an internationally coordinated level, giving way to the introduction of a wide range of sizing rules. As there is no common methodology, like the one that was developed for non-residential buildings by CEN [230], that is used for the different standards, the flow rates proposed in them can't be compared easily. AIVC listed the requirements of 15 standards without attempting to analyse their performance [360]. A similar effort was done in the framework of the EPHECT project [231]. Two reviews, one by Yoshida [313] and another within the HEALTHVENT project [312] applied the sizing rules to a reference dwelling and found that the design air change rate in the majority of standards is around 0.5 ACH.

In the moderate climate region of West-Europe, especially in Belgium, the Netherlands, France and the UK, mechanical exhaust ventilation systems dominate the residential ventilation market [317, 361, 362], while heat recovery ventilation and natural ventilation are the most common residential ventilation systems in northern and southern Europe respectively. Such simple exhaust systems are composed of a mechanical exhaust fan, ducted to a series of vent holes in the different 'wet' spaces in the dwelling such as kitchens, toilets, bathrooms and service rooms, combined with externally and internally mounted air transfer devices [269]. The externally mounted air transfer devices, also called trickle ventilators, are intentionally made perforations in the building shell that deliver the make-up air for the air extracted from the dwelling by the fan, while their internally mounted counterparts, also called transfer grilles, allow the air to flow from one space to another. Since the introduction of ASHRAE 62.2, this kind of ventilation system is also rapidly achieving a dominant position in the US residential ventilation market, although the use of trickle ventilators is usually omitted and not treated as such in the standard. The sizing rules for the trickle ventilators in the standards of the 4 European countries also demonstrate little uniformity, requiring the design flow rate, which itself is different for all standards, to be achieved at a different design pressure difference across the ventilator, ranging between 1 and 20 Pa.

The total air change rate achieved by mechanical exhaust ventilation systems can be considerably different from the flow rate of the fan due to adventitious ventilation and infiltration [282]. The importance of the extra flow rate is mainly related to the sizing of the trickle ventilators relative to the flow rate of the fan [363]. Therefore, the ventilation heat loss of mechanical exhaust ventilation systems can't be assessed comprehensively by simple comparison of the design flow rates. In addition, the air flow in the system is controlled by the mechanical flow rate only in the 'wet' spaces, whereas the flow rate in the rest of the dwelling, which comprises the main living spaces, is governed by much less stable driving forces such as wind and buoyancy. Since the occupants spend the vast majority of time in the livings spaces [357, 358], the indoor air quality (IAQ) achieved in these spaces will be the dominant contributor to perceived air quality [67]. Again, the design flow rates will not be a good metric for assessing the performance of mechanical exhaust ventilation systems.

Presenting the results from a multi zone simulation based performance assessment of mechanical exhaust ventilation systems sized in accordance with the Belgian [364], British [365], Dutch [366], French [367, 368], and ASHRAE [369] residential ventilation standards, this chapter aims to contribute to this debate. The 4 European countries are chosen because of the dominance of mechanical exhaust ventilation in their ventilation market and their geographical clustering. Although mechanical exhaust ventilation historically also represents a large part of the residential ventilation market in the Nordic countries, their cold climate [282] and recent market evolutions favour heat recovery ventilation. Therefore they were not included. The ASHRAE standard was chosen for its large geographical applicability and it’s authority in HVAC design. Additional motives include the fact that its promotion of mechanical exhaust ventilation is novel in the US and its recent publication. The sizing rules of each standard are applied to 5 common dwelling typologies and Monte Carlo analysis is used to consider the sensitivity of the results to the boundary conditions used.

2.2 Methods

To assess the quality of the sizing rules in the standards discussed above, mechanical exhaust residential ventilation systems have been designed in accordance with the different standards for five different dwelling typologies. The geometries of the dwellings have been developed in the framework of a research project on the optimisation of building envelope and services for low-energy residential buildings [370-372]. Their size and layout is based on an extensive survey of 200 dwellings in Belgium built in the 1990’s [373] and have been used in several previous research projects eg. [374]. Their characteristics have been checked with the evolution of newly built dwellings and still correspond well with current building practise. Four dwellings are single family houses, one is a flat. All dwellings have the same useful floor area corresponding to the mean from Belgian national statistical figures. All houses comprise a living room, 3 bedrooms, kitchen, bathroom, toilet, service room, and hall way, with a total net floor area of about 150 m^2. The detached, semi-detached and terraced house hold a separate study.

Table 2.1 Geometrical characteristics of reference dwellings

	Bungalow	Detached	Semi-detached	Terraced	Flat
Compactness (m)	0.9	1.3	1.6	2.1	3.8
Heated volume (m)	557.3	528.7	521.0	493.6	450.0
Heat loss area (m)	611.3	395.4	330.1	231.9	118.4
Number of floors	1	2	2	3	1 (of 6)

The dwellings differ in building compactness, ranging from a detached bungalow to a flat in a 6-floor apartment building. The market share of newly built dwellings during the last decade in Belgium is typically 40% detached houses, 40% flats and 20% terraced or semi-detached. In all countries studied, all types are typically found, with an overbalance of flats and terraced houses in the cities, whereas detached dwellings are dominantly found in rural areas. Table 2.1 gives an overview of geometrical characteristics of the five reference dwellings. The compactness is defined as the ratio of the volume to the heat loss area. Graphical representations of the floor plans are given in Figure 2.2 to Figure 2.1.

The results presented in this chapter are based on airflow simulations. These were executed in the multi-zone airflow simulation package Contam [375], which takes effects of buoyancy, wind and fan pressure into account and is used in numerous ventilation studies eg. [eg. 136, 226]. The validation of multi-zone ventilation models against e.g. tracer gas measurements is well documented in literature [341, 376-378]. Multi-zone simulation models typically assume well mixed air in every room (simulated as a single node in the model). As a result, these models are not suited for detailed analysis of the distribution of contaminants in a single room. This aspect can be studied with computational fluid dynamics (CFD) [379-382]. However, this is not the scope of this section since, in contrast to a typical office setting, no specific occupied zone can be defined in a residential setting. In addition, CFD would be too computationally demanding for the scope of this chapter. To assess the heat loss through hygiene ventilation, only the bulk fresh airflow in the building is relevant. As Contam is a ventilation model only, it cannot calculate transient room or duct temperatures. Therefore, for simplicity, the temperature inside the building and all ducts has been set to 18 °C, the inside temperature fixed by the Belgian EPBD calculation procedure , which corresponds to the average temperature measured in Belgian dwellings [298]. The effect of this assumption has been discussed by Steeman [383, 384]. The test reference year for Ukkel, Belgium was used as the outdoor climate for all simulations, with hourly mean values for temperature, humidity, wind speed and direction.

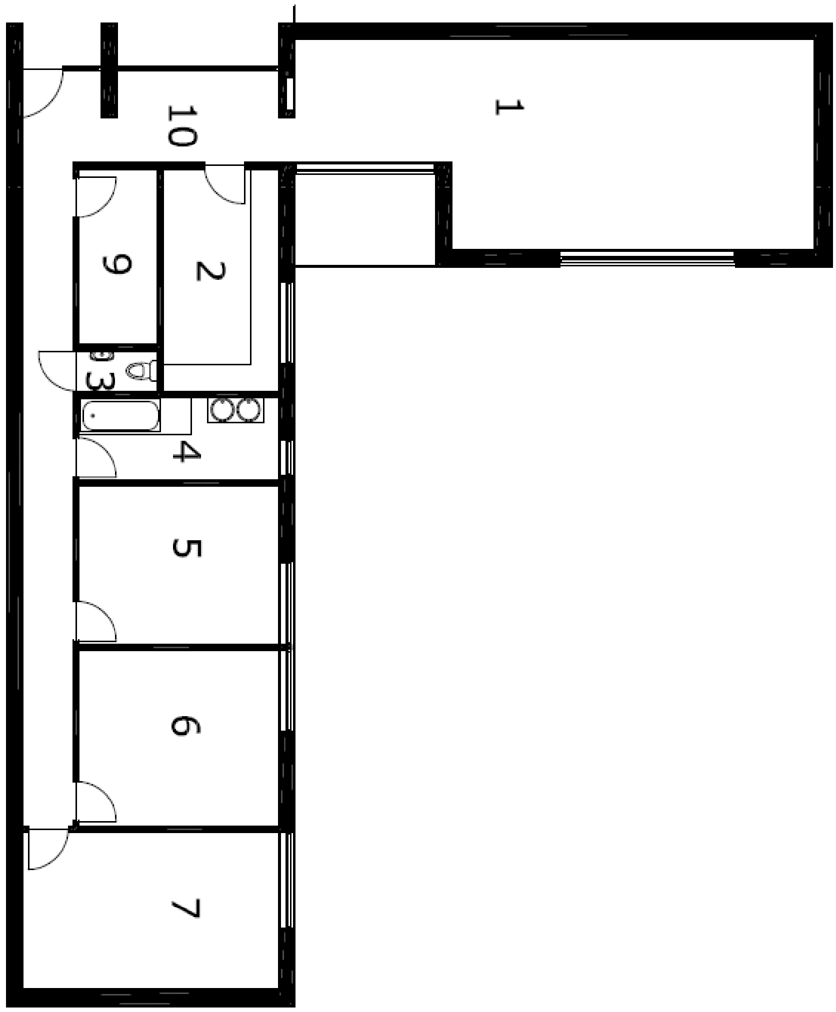

Figure 2.1 Floor plan of the bungalow

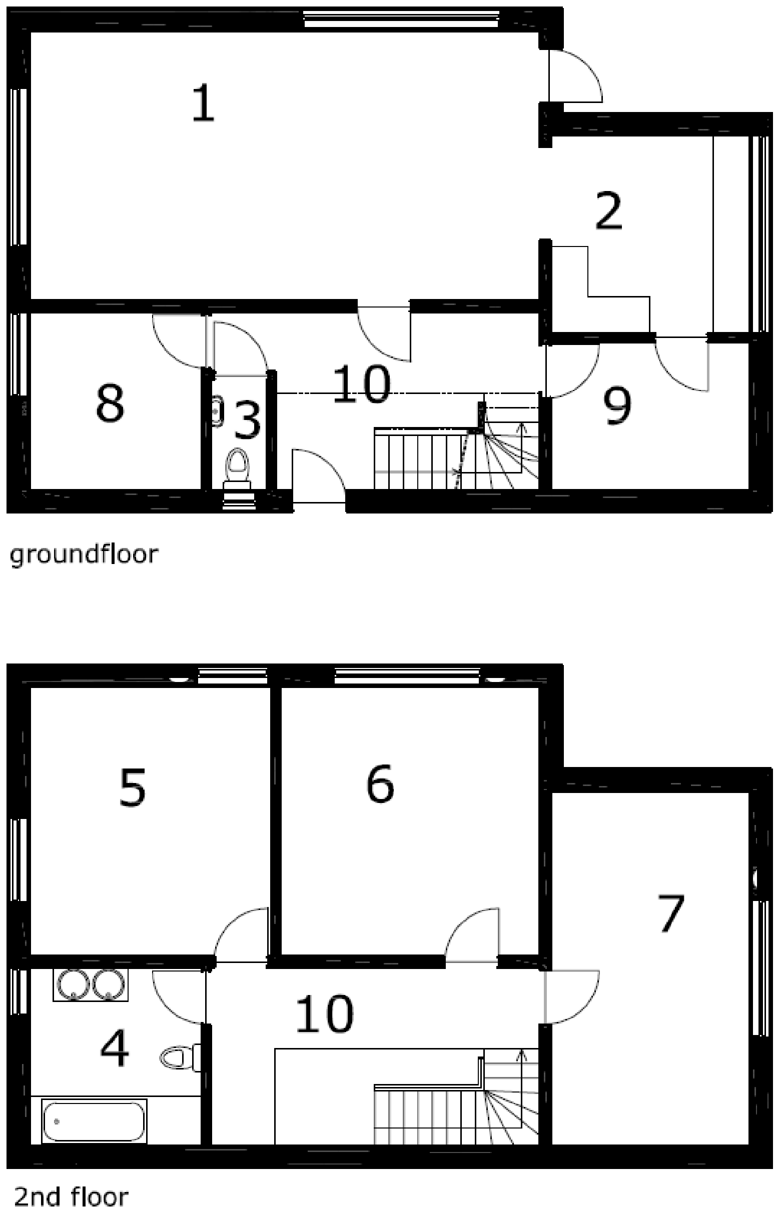

Figure 2.2 Floor plan of the detached dwelling

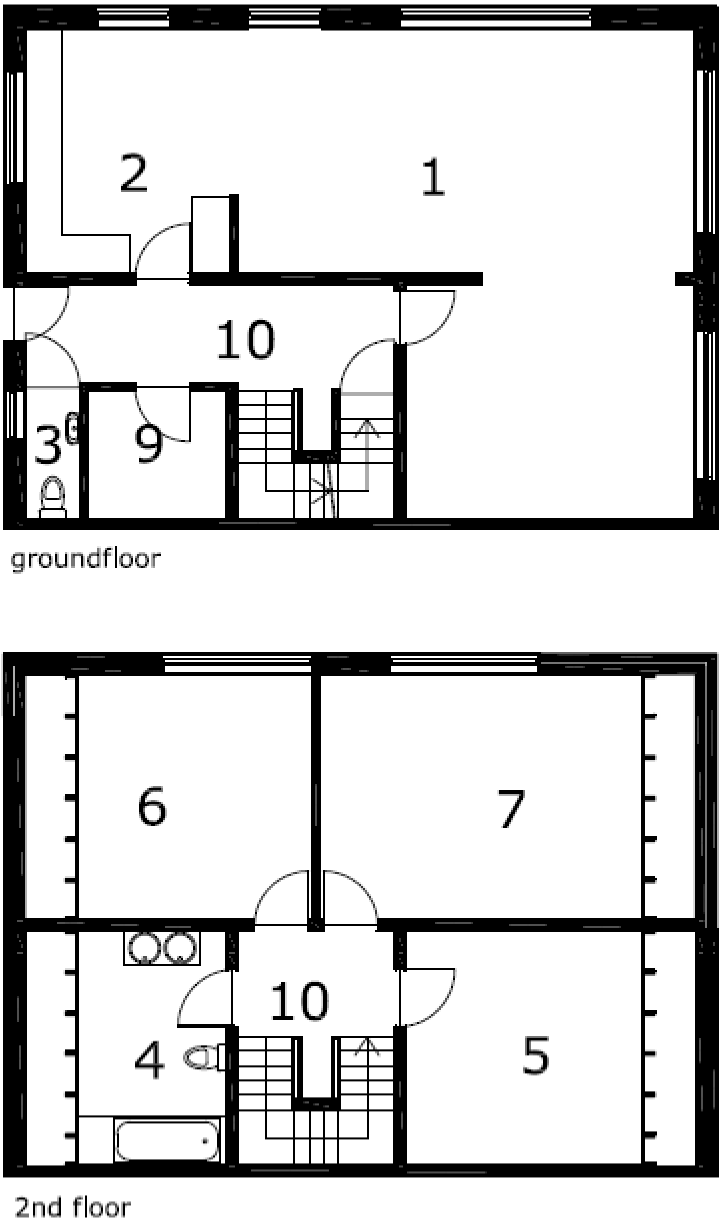

Figure 2.3 Floor plan of the semi-detached dwelling

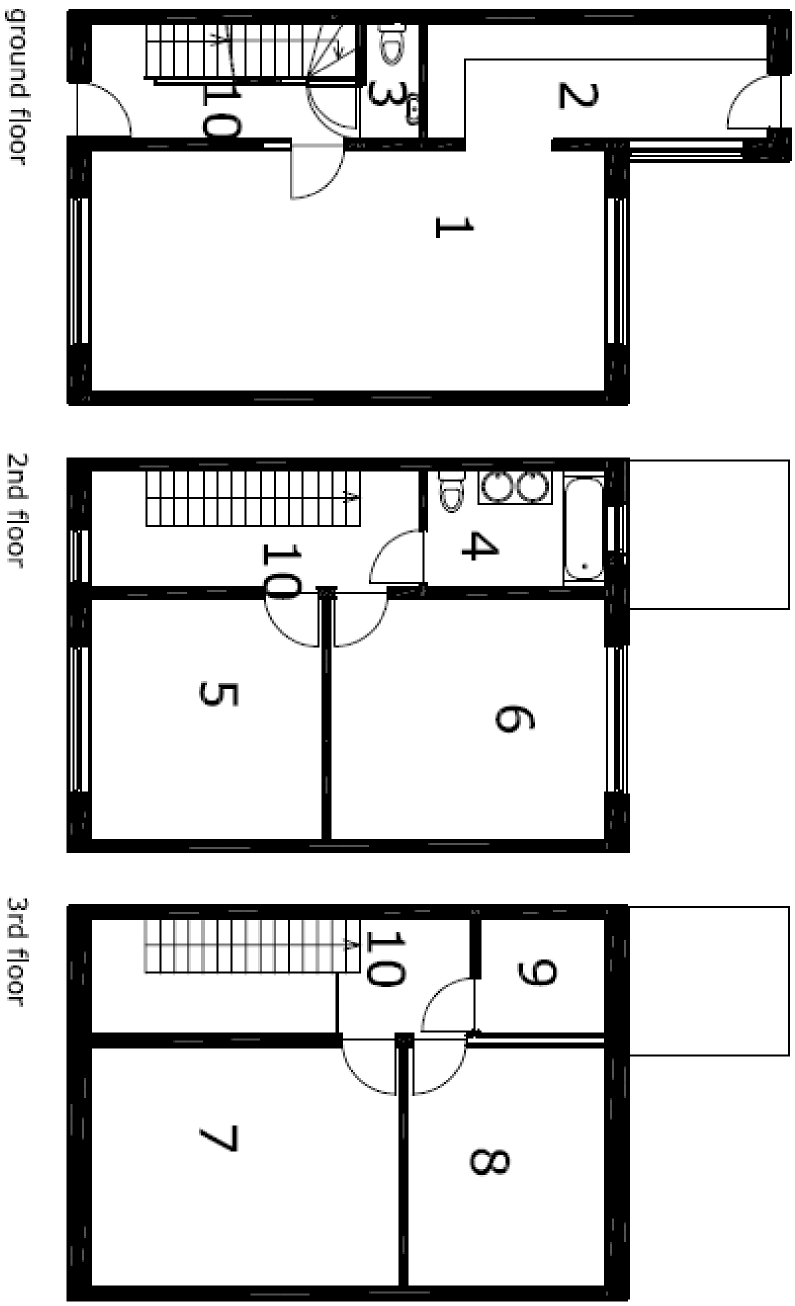

Figure 2.4 Floor plan of the terraced dwelling

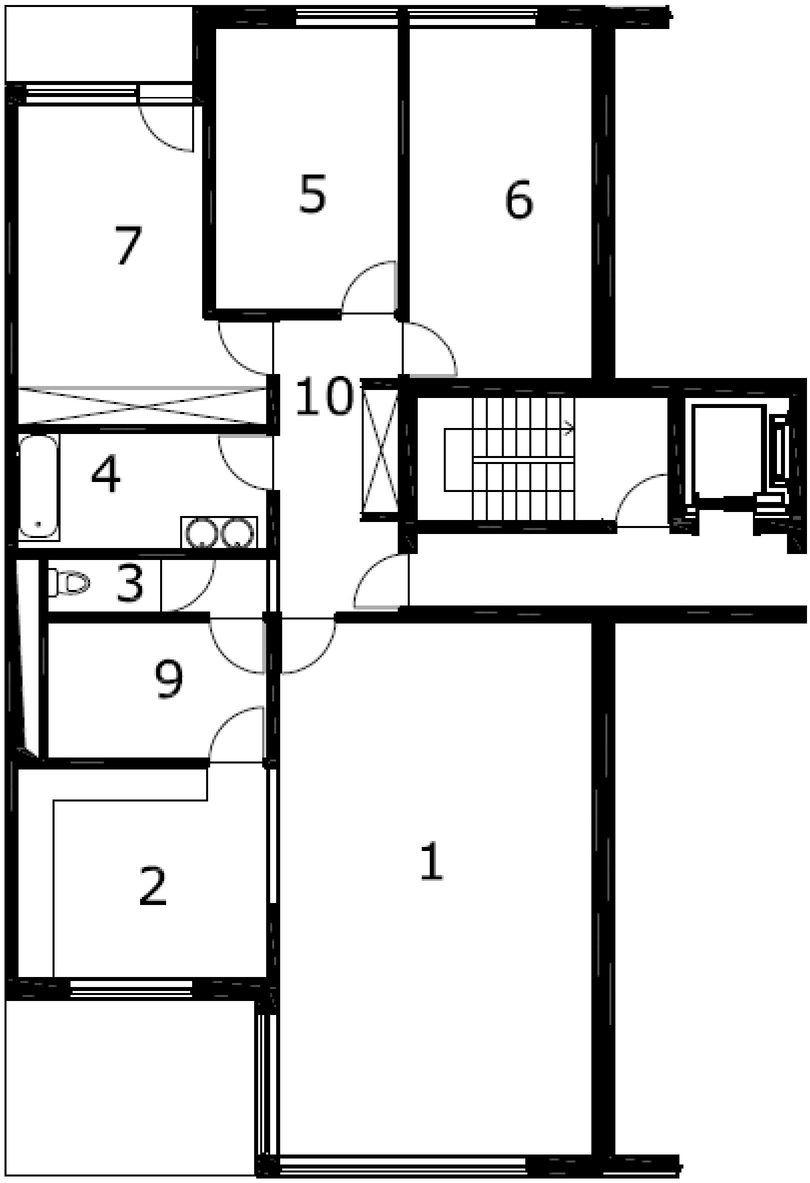

Figure 2.5 Floor plan of the flat

2.2.1 Building Model

The airflow in the dwellings has been modelled taking into account both the ventilation system and leakage. Overall leakage, characterized by the v_{50} value, is modelled by means of cracks in the roof and wall surface. The v_{50} value is the ratio of the air leakage rate at 50 Pa pressure difference and the building envelope heat loss area. According to observations by Bossaer [298], the specific leakage rate through roof and walls has a 2/3 ratio, which has been implemented in the model. Each wall is fitted with two cracks, one at 1/4 of its height and the second one at 3/4. The internal doors are simulated with additional cracks in the walls. For the indoor walls, a fixed specific leakage value is assumed. This methodology is in agreement with guidelines given in EN 15242 [385]. In the results presented, a specific air leakage (v_{50}) of 3 m/h is used, representing the best quartile of measured airtightness values in a measurement campaign in Flanders in the late 90's [298]. A recent measurement campaign [386], along with results from other countries [296], shows a tendency towards this level of airtightness in newly built dwellings.

The production of CO_2 within the model is only related to the occupants' metabolism and corresponds to their whereabouts. A constant outdoor background concentration of 350 ppm is assumed.

2.2.2 Ventilation System Design and Model

All mechanical exhaust vents were modelled as constant volume flow rate components in the respective zone node, while transfer grilles and trickle ventilators were modelled with single direction power law flow components with a flow exponent of 0.5 [270]. A cookerhood with an exhaust flow rate of 200 m^3/h is modelled in the kitchen and is activated during cooking activities. All systems were modelled with windows and internal doors closed, in order to simulate the performance of the systems as such, without user interaction.

2.2.3 Assessment Parameters

Through the correlation between excess CO_2 concentration and mean percentage of dissatisfied [228] and Fanger's Perceived Air Quality approach [67], excess CO_2 concentration is now widely accepted as a proxy for perceived indoor air quality [230], especially if the main pollution sources are related to the human metabolism. In contrast to the basic model, steady state conditions are rarely applicable to real ventilated environments. CO_2 concentrations are inherently transient, due to changes in environmental boundary conditions. Additionally, the relevant CO_2 sources tend to constantly move around in the multi-spaced dwelling, introducing discontinuous sources and further increasing the transient character of the indoor air quality. There is no consensus in literature about the way transient concentrations have to be interpreted. This lack of agreement is reflected in the disparate list of performance criteria provided in EN 15665 [229]. From the suggested parameters in this standard, 4 were selected for use in this chapter, namely the heating season average CO_2 concentration to which an occupant is exposed, the amount of time an occupant spends in an environment within the different IDA classes [230] and the dose of CO_2 over 1000 ppm excess CO_2. The latter is expressed normalised to both the total time of the heating season and to the time in excess of 1000 ppm. Further, a comfort zone between 30-70% relative humidity is considered.The first criterion is defined as

$$E\left(C_{\exp}(t)\right), \tag{2.1}$$

where

C_{exp}	=	CO_2 concentration to which an occupant is exposed (ppm).
t	=	simulation time (s).

This criterion provides a qualitative assessment of the average ability of the system to maintain an adequate indoor air quality level in the dwelling.

In addition to this, the second criterion shows a rough distribution of the indoor air quality. For each IDA class it is defined as

$$P\left\{IDA_{y,LL} \leq C_{\exp}(\mathrm{t}) < IDA_{y,UL}\right\}, \tag{2.2}$$

where

C_{exp}	=	CO_2 concentration to which an occupant is exposed (ppm)
$IDA_{y,LL}$	=	lower limit of IDA class y
$IDA_{y,LL}$	=	upper limit of IDA class y.
t	=	simulation time (s).

The best IDA class, IDA 1, corresponds to exposure to less than 400 ppm excess CO_2, while the lowest class, IDA 4, exposure to concentrations in excess of 1000 excess CO_2, is considered to correspond to poor perceived indoor air quality.

The third criterion is used as an indication for the exposure of the occupants to peak carbon dioxide concentrations:

$$E_O\left(\int_t \max\left(C_{\exp}(t) - C_{amb}(t) - C_{thr}, 0\right) \mathrm{dt} \mid \mathrm{O}\right), \tag{2.3}$$

where

C_{exp}	=	CO_2 concentration to which an occupant is exposed (ppm)
C_{amb}	=	ambient CO_2 concentration (ppm)
C_{thr}	=	threshold CO_2 concentration, in casu 1000 (ppm)
O	=	a single occupant
t	=	simulation time (s).

By normalizing the this to either the total simulation time or the time where C_{exp} – C_{amb} exceeds the threshold, two different metrics are obtained: the first one describes the average dose of high exposure per unit of time, while the second is an expression of the average height of the peak exposure when it occurs. An alternative criterion for peak exposure is the 95th percentile of C_{exp}.

The exposure to building material emissions themselves is not assessed. Exposure to emissions originating from building materials and their secondary effects can be reduced effectively with source control measures [182, 387]. Therefore, it is not considered as a performance indicator for the ventilation systems in this chapter. Likewise, exposure to emissions due to specific activities such as cooking[118, 119, 121, 128, 134], cleaning[10, 137, 138, 142], indoor smoking [28, 112, 117, 176] is not considered since it is best controlled by either source control or intensive local ventilation.

The ability of a sizing option to contain nuissant contaminants produced in the ‘wet’ spaces, such as odours, within these spaces and prevent their spreading through the dwelling is also assessed by the exposure to humidity, since moisture production is mainly linked to activities in these spaces. Since both dwelling geometries have a separate WC and toilet use is not linked with considerable moisture production, the intake fraction [261] of an odour tracer, assuming a constant average breathing rate, also called minute ventilation, of 12 l/min [259], is used to assess the containment of this specific pollutant class. The intake fraction is calculated as the ratio of mass intake to the respiratory system and mass emitted from the source:

$$IF = \frac{\int \mathrm{Q_b}(t) \cdot C(t)dt}{\int E(t)}, \tag{2.4}$$

where

IF	=	intake fraction (g/g)
Q_b	=	breathing rate (l/s)
C	=	concentration (l/l)
E	=	emission rate (l/s).
t	=	simulation time (s).

The total convective heat loss from the heated volume of the dwelling through the combination of mechanical ventilation, adventitious ventilation and infiltration,

$$Q_{v,tot} = E_B\left(\int_t \Phi_{v,tot}(t)\,\mathrm{dt} \mid \mathrm{B}\right), \tag{2.5}$$

where

$\Phi_{v,tot}$	=	total ventilation heat loss (W)
B	=	a single set of boundary conditions,
t	=	simulation time (s).

expressed in kWh per year and per m^2 of heated floor area,is used to assess the energy performance of the different sizing rules. The heat loss through mechanical ventilation corresponds to the heat content of the air supplied to / exhausted from the building by the fans:

$$\Phi_{v,mech} = c \cdot \Delta T_{in-out} \cdot \max(\sum \dot{g}_{\sup}, \sum \dot{g}_{exh}), \qquad (2.6)$$

where

$\Phi_{v,mech}$	=	mechanical ventilation heat loss (W)
c	=	specific heat of air (J/(kg·K))
$\Delta T_{in\text{-}out}$	=	indoor/ambient temperature difference (K)
$\dot{g}_{sup/exh}$	=	mass flow rate over the supply/exhaust fans (kg/s).

Adventitious ventilation heat loss is the difference between the heat loss through the total air flow rate across all trickle ventilators in mechanical exhaust systems or the exhaust vents in mechanical supply systems and the heat loss through mechanical ventilation:

$$\Phi_{v,adv} = c \cdot \Delta T_{in-out} \cdot \max(\sum \dot{g}_{trck}, \sum \dot{g}_{ven}) - \Phi_{v,mech}, \qquad (2.7)$$

where

$\Phi_{v,adv}$	=	adventitious ventilation heat loss (W)
$\Phi_{v,mech}$	=	mechanical ventilation heat loss (W)
c	=	specific heat of air (J/(kg·K))
$\Delta T_{in\text{-}out}$	=	indoor/ambient temperature difference (K)
$\dot{g}_{trck/ven}$	=	mass flow rate over the trickle/exhaust vents (kg/s),

while infiltration heat loss is the heat loss associated with air entering the dwelling through the leaky envelope:

$$\Phi_{v,\inf} = c \cdot \Delta T_{in-out} \cdot \sum \dot{g}_{env}, \qquad (2.8)$$

where

$\Phi_{v,inf}$	=	mechanical ventilation heat loss (W)
c	=	specific heat of air (J/(kg·K))
$\Delta T_{in\text{-}out}$	=	indoor/ambient temperature difference (K)
$\dot{g}_{env}$	=	mass flow rate over the envelope leaks (kg/s).

The breakdown of the total ventilation heat loss into mechanical ventilation, adventitious ventilation and infiltration heat loss is graphically explained in Figure 2.6 for a simulation with mechanical exhaust ventilation.

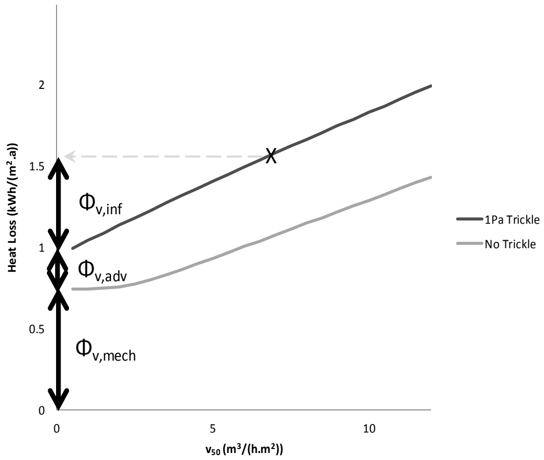

Figure 2.6 Graphic representation of the breakdown of the total ventilation heat loss into mechanical ventilation, adventitious ventilation and infiltration heat loss in a simulated case (x) with mechanical exhaust ventilation with trickle ventilators sized at a 1 Pa pressure difference.

Fan power was not taken into account because it is very system specific. Since heat loss and exposure reduction are opposing interests, they have to be traded off against each other [234, 388]. Several authors have proposed using weighted sums of these different criteria [389, 390]. The definition of these weighting coefficients, however, lacks scientific evidence. Therefore, the trade-off is addressed by means of the concept of pareto optimality. Pareto optimal cases are cases where none of the other standards achieve better results on both indoor air quality and heat loss.

2.2.4 Sensitivity

One of the main problems with simulation models is the uncertainty on input data, despite the fact that the sensitivity of the results to variation in the input data may be very high. A lot of variables have a distinct influence on the performance of the system and consequently the performance of the system will be different for each set of parameters. Therefore, the use of a calculation method that takes both the variation of the different parameters as well as the interaction between them into account is required to acquire statistically relevant data. Large sensitivity to input uncertainty often appears near equilibrium situations [391, 392] which occur for specific values of structural parameters or weather conditions [363].

To prevent this input dependency of the results, the Monte-Carlo (MC) approach, as proposed by Van Den Bossche et al. [347, 393], has been used in this study. In this approach, instead of fixing 1 value for each input data, a distribution is determined for the key parameters and multiple simulations are carried out with different values of these parameters. According to Furbringer

[391, 392] convergence can be reached within 100 simulations if the amount of input parameters is limited and the variance in not dominated by 1 parameter.

Sensitivity analysis based on a Monte Carlo algorithm has been implemented in building simulation by e.g. Breesch [394]. Dorer et al. [275, 342] presented work specifically for residential ventilation systems within the framework of the EC Reshyvent - EU cluster project.

The Monte-Carlo process can be sped up by using Low Discrepancy Sequences (LDS) instead of random numbers [395]. In contrast to randomly sampled points, they distribute the instances to empty areas in the sample space to prevent overlapping and clusters, which are very common with ordinary random numbers. Another advantage of LDS is that these sequences are entirely repeatable, giving the same sequence every time. These sequences are used to generate the parameter sets used in this chapter.

A sensitivity analysis has pointed out that wind related factors such as wind velocity and wind reduction parameters [396] and the number of inhabitants and their occupancy schedules have the biggest influence on the overall performance of the ventilation system [393]. As wind conditions can change considerably due to the specific site and seeing as the territory of several standards includes multiple climate zones, the sensitivity to wind is taken into account by considering a distribution on the wind exposure parameters rather than changing the climate data.

The following input variables are considered with a probabilistic approach (Normal distributions are mentioned as N(mean, standard deviation):

- Façade orientation - interval [0°; 359°]
- C_p coefficients - interval of the 6 AIVC tables [397]
- Terrain roughness α , partially correlated with the C_p coefficients – interval [0.149 – 0.377]
- Sunday is the ...th day of the year - interval [1;7]
- Moisture production from domestic activities - normal distribution (see below)
- Production of moisture and carbon dioxide by occupants - normal distribution (see below)
- Number of occupants - specific distribution
- Weekday / weekend occupancy schedules - specific distribution

The number of parameters can be considered to be small, so 100 datasets will be used to perform the simulations. Moisture production for domestic activities is based on data available in the EU technical report on design and dimensioning of residential ventilation systems [398]. The production in the bathroom is N(0.5, 0.05) l/s, in the service room cloth drying is N(1, 0.05)l/s and for cooking, a half hour cycle of N(0.6, 0.05) l/s, N(1, 0.1) l/s and N(1.5, 0.1) l/s for 10 minutes each is used. The production of moisture and carbon dioxide by occupants is modelled as a linear function of the metabolism, which varies for each activity (eg. N(0.8, 0.05) Met for sleeping, N(2, 0.1) Met for cooking). Based on EN 15251[218], the production rate is 11.875 l/h/Met for CO_2 and 34.375 g/h/Met for moisture.

The number of occupants and the occupancy schedules are considered with a specific distribution based on the social demography and time use studies in Belgium. Based on the available data, 100 different data sets were compiled with different occupancy schedules. Figure 2.7 shows the evolution of the probability for the occupants to be in a particular room over the course of a weekday. The number of occupants in the building varies from one to six (1: 3%, 2: 21%, 3: 31%, 4: 32%, 5: 10%, 6: 3%), with an average of 3.34 persons per building.

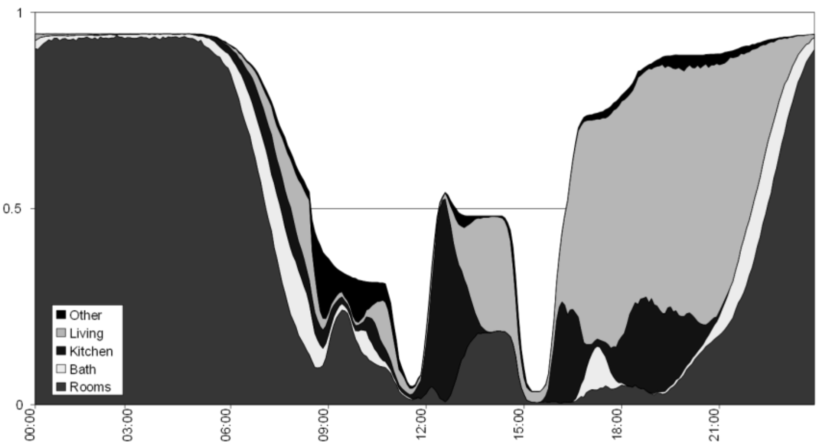

Figure 2.7 Probability for the occupants to be in a particular room over the course of a weekday.

2.3 Sizing of Exhaust Systems in the Standards

As was explained in the introduction, the sizing rules for mechanical exhaust residential ventilation systems put forward are different in the Belgian, Dutch, French, UK and ASHRAE standards. In this section, the specific rules found in each of the standards are summarized. If different sizing rules are provided for continuous and demand controlled systems, only those for continuous systems are considered.

2.3.1 Belgium

The Belgian standard requires a design flow rate of 1 l/s·m^2 for each occupied space. For the main living space, this design flow rate should be at least 21 l/sand can be limited to 42 l/s, while for bedrooms, studies… the minimum value is 7 l/s and the design flow rate can be limited to 20 l/s. For kitchens, bathrooms and service rooms, a minimum design flow rate of 14 l/s should be taken into account, while it can be limited to 21 l/s. The design flow rate for a toilet is 7 l/s. Table 2.2 provides a summary of the design flow rates.

The occupied spaces and the wet spaces should be connected to each other or via circulation spaces by transfer grilles sized at 7 l/s at 2 Pa pressure difference, which corresponds to 70 cm², except for the kitchen, in which the transfer grille should be sized twice as large. Each living space, bedroom, study… should be connected to the outdoor environment by a trickle ventilator sized at the design flow rate for that space at 2 Pa pressure difference.

Table 2.2. Design flow rates in the Belgian Standard

Space	Design Flow rate $(l/(s \cdot m^2))$	Minimum (l/s)	Maximum (l/s)
Living room	1	21	42
Bedroom/study/…	1	7	20
Kitchen/Bathroom/Service room	1	14	21
Toilet	-	7	-
Hall/stairwell	1	-	-

2.3.2 The Netherlands

With a design flow rate of 0.9 $l/(s \cdot m^2)$ for each occupied space and minimum design flow rates of 7 l/s in bedrooms, studies and toilets and 14 l/s in bathrooms and service rooms , the Dutch standard's sizing rules are quite similar to those in the Belgian standard. The minimum design flow rates for the kitchen, however, is set at 21 l/s instead of 14 l/s, while in the main living space, only 7 l/s is required as opposed to 21 l/s in the Belgian standard. Furthermore, Trickle ventilators should be sized at the design flow rate at 1 Pa pressure difference and transfer grilles should have a free face area of 12 cm² multiplied by the design flow rate for that space. As a consequence, the size of the trickle ventilators and transfer grilles is larger compared to the Belgian standard's sizing rules.

Table 2.3. Exhaust flow rates in the French standard (l/s)

N° of main spaces	Kitchen	Bathroom	Service	Toilet
1	21	5	5	5
2	25	5	5	5
3	30	9	5	5
4	34	9	5	5 or 9 *
≥5	38	9	5	5 or 9 *

* 5 l/s if multiple toilets are present, 9 l/s if only one

2.3.3 France

The design flow rate for each of the 'wet' spaces in the French standard depends on the number of 'main' spaces in the dwelling, eg. living spaces, bedrooms, studies.... These flow rates have been tabulated in Table 2.3. The design flow rates of the trickle ventilators in the remaining spaces are also defined as a function of the number of 'main' spaces. For dwellings with only 1 or 2 'main' spaces, the design flow rate is increased for higher total design flow rates in the 'wet' spaces (Table 2.4)

The trickle ventilators should be sized to the design flow rate at 20 Pa pressure difference, while the transfer grilles should be sized to the design flow rate at 5 Pa and 2.5 Pa for 'wet' and 'main' space grilles respectively. As a consequence, the size of components is typically smaller compared to the Dutch and Belgian standard's sizing rules.

Table 2.4. Design flow rates for trickle ventilators in the French standard (l/s)

N° of main spaces	Total exhaust flow rate	Living room	Other
1	25	25	-
1	29	25	-
1	36	33	-
2	33	17	8
2	36	17	13
3	42	17	8
4	50	13	8
5	58	13	8
6	58	13	6
7	63	13	6

2.3.4 UK

Mechanical exhaust ventilation is denominated 'extract ventilation' in the British standard. Design flow rates of 13, 8 and 6 l/s are required for kitchens, bathrooms and toilets respectively. Service rooms are treated as bathrooms. In addition to these design flow rates per space, the total extracted flow rate should not be less than 9 l/s, increased with 4 l/s for each bedroom.
The equivalent, referenced to a round sharp edged opening, free face area of transfer grilles is set at 76 cm², that of the trickle ventilators at 25 cm².

2.3.5 ASHRAE

The design flow rate for kitchens proposed in the ASHRAE standard is 5 ACH, while 10 l/s is required for bathrooms. The total design exhaust flow rate for a dwelling is at least 0.05 l/(s·m^2), increased with 7 l/s for the first bedroom and 3.5 l/s for each additional bedroom. No requirements for trickle ventilators or transfer grilles are included.

2.3.6 Summary

The design exhaust flow rate for the wet spaces in the dwellings according to the different standards are listed in Table 2.5. Since the design exhaust flow rates in most standards are size independent, the flow rates are mostly the same for all 5 dwellings. If this is not the case, the range of the design flow rates for that space in the 5 dwellings is mentioned. Since trickle ventilators are sized with respect to different reference pressure differences in the five standards, the flow coefficient at 1 Pa pressure difference, assuming a flow exponent of 0.5 and a simple power law flow profile, for the trickle ventilators in the various living spaces of the dwellings according to the different standards are listed in Table 2.6. Note that the ASHRAE standard does not require the installation of trickle ventilators.

Table 2.5. Design exhaust flow rates in wet spaces in the 5 standards (l/s)

Space	Belgium	France	Netherlands	UK	ASHRAE
Kitchen	14	38	21	13	33 - 48
Bathroom	14	8	14	8	10
Service room	14	4	14	8	10
Toilet	7	8	7	6	-
Total exhaust flow rate	49	58	56	35	53 - 68

Table 2.6. Trickle ventilator flow coefficient in living room and master bedroom in the 5 standards ($l/(s \cdot Pa)$)

Space	Belgium	France	Netherlands	UK	ASHRAE
Living room	20 - 29	3	26 - 40	2	-
Master bedroom	11 - 14	2	14 - 21	2	-

2.4 Performance of Exhaust Ventilation

As was mentioned in the methods section, the indoor air quality provided by the sizing rules found in the Belgian, British, Dutch, French and ASHRAE standards is assessed using 4 different criteria for CO_2 concentrations suggested by the EN 15665 standard [229]. All of these, however, integrate the transient concentrations into a single number, losing lots of information in the process. Therefore, the cumulative distribution functions of the exposure to CO_2 are also given for some configurations. In Figure 2.8, the 4 criteria are shown in the cumulative distribution chart.

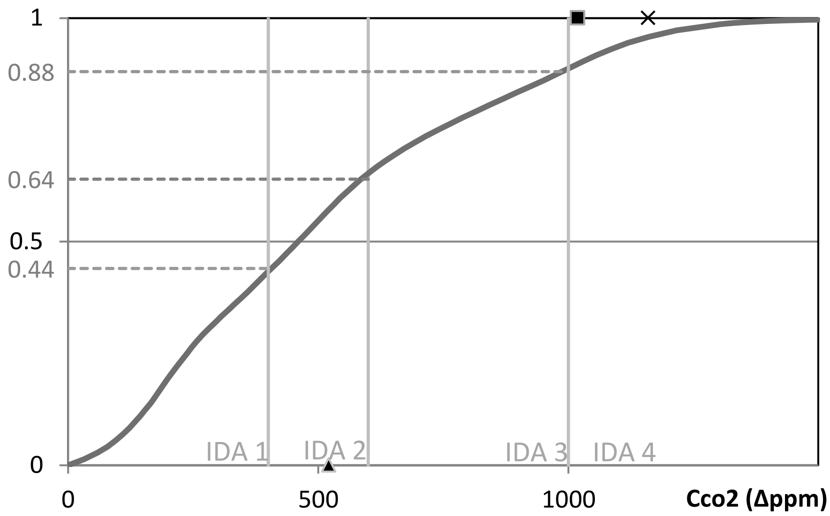

Figure 2.8 Cumulative distribution of the CO2 concentration to which the occupants are exposed for the flat case under the sizing rules of the British standard, with marks indicating the average concentration (Δ), the dose above 1000 ppm normalized to the total time (■) and to the time above 1000 ppm (X). Limits between IDA classes are indicated by vertical lines.

2.4.1 Carbon Dioxide Exposure

Table 2.7 lists the time fractions spent in the different IDA classes considering all 334 occupants from the 100 simulations in the monte-carlo analysis for all 5 standards in all 5 geometries.

Table 2.7 Time fractions spent in the different IDA classes considering all 334 occupants from the 100 simulations in the Monte-Carlo analysis for all 5 standards in all 5 geometries.

		Belgium	France	Netherlands	UK	ASHRAE
Flat	IDA 1	0.662	0.757	0.670	0.430	0.701
	IDA 2	0.184	0.185	0.162	0.221	0.114
	IDA 3	0.133	0.055	0.140	0.236	0.174
	IDA 4	0.021	0.002	0.028	0.114	0.010
Terraced	IDA 1	0.714	0.724	0.679	0.391	0.527
	IDA 2	0.164	0.130	0.151	0.191	0.197
	IDA 3	0.098	0.136	0.125	0.252	0.150
	IDA 4	0.024	0.011	0.046	0.166	0.127
Semi-Detached	IDA 1	0.695	0.702	0.719	0.452	0.634
	IDA 2	0.162	0.185	0.152	0.195	0.182
	IDA 3	0.116	0.106	0.118	0.227	0.161
	IDA 4	0.027	0.008	0.011	0.126	0.024
Detached	IDA 1	0.696	0.705	0.706	0.429	0.509
	IDA 2	0.163	0.142	0.148	0.183	0.185
	IDA 3	0.102	0.128	0.097	0.238	0.175
	IDA 4	0.039	0.025	0.048	0.149	0.131
Bungalow	IDA 1	0.719	0.666	0.693	0.436	0.482
	IDA 2	0.171	0.180	0.161	0.190	0.173
	IDA 3	0.103	0.144	0.137	0.215	0.205
	IDA 4	0.006	0.010	0.010	0.158	0.140

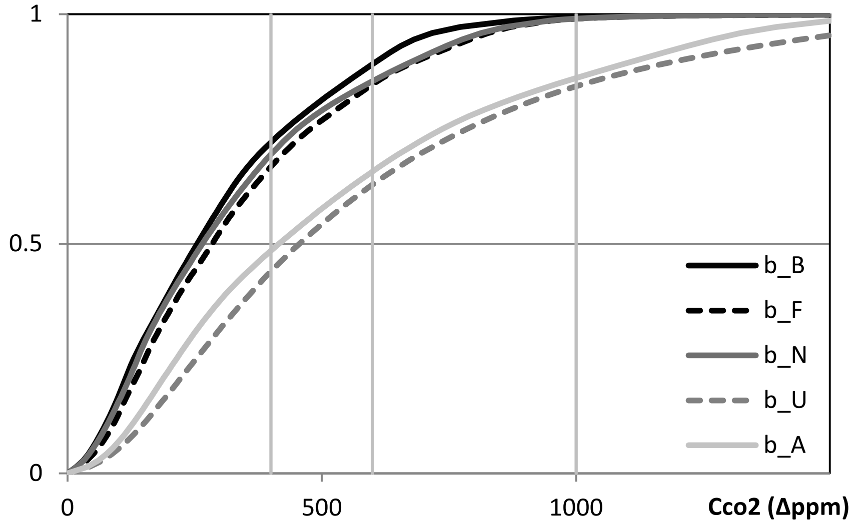

Figure 2.9 Cumulative distribution of the CO_2 concentration to which the occupants are exposed for the bungalow case (b) under the sizing rules of the Belgian (B), French (F), Dutch (N), British (U) and ASHRAE (A) standards.

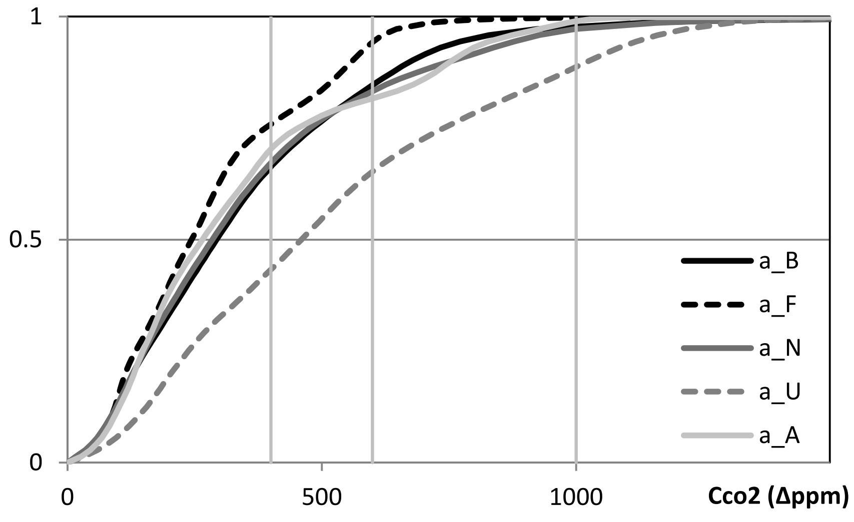

Figure 2.10 Cumulative distribution of the CO_2 concentration to which the occupants are exposed for the flat case (a) under the sizing rules of the Belgian (B), French (F), Dutch (N), British (U) and ASHRAE (A) standards.

Looking at the cumulative distribution of the excess carbon dioxide concentration for the bungalow (Figure 2.9) and the flat (Figure 2.10), a few typical results can be deduced. The Belgian, Dutch and French standards consistently achieve similar indoor air quality, at a level which is considerably higher than that achieved by the British standard. The performance of the systems sized according to the ASHRAE standard, relative to the other standards, is much more prone to variation due to the fact that the flow rate is mainly concentrated in the kitchen and expressed as a function of its volume. Although the flow rates are sim-ilar in magnitude to those prescribed in the French standard, the lack of transfer grilles in the ASHRAE standard prevents a good distribution of this flow rate through the rest of the dwelling. Position and size of the kitchen relative to the other spaces therefore has a large influence on the achieved performance.

This is also reflected in the average excess carbon dioxide concentration to which the occupants are exposed, as well as in the dose above 1000 ppm excess carbon dioxide, both normalized to the total time of the simulation and to the time above that concentration. These criteria are listed in Table 2.8 along with the heating season averaged specific convective heat loss of the dwellings considered. Although general trends are similar for all criteria, the ranking of the different standards sometimes flips completely from one criterion to the next. The Belgian standard, in the terraced house, for instance, is ranked first if average exposure to excess carbon dioxide concentration is the criterion, second if the dose over 1000 ppm of excess carbon dioxide normalized to the total time and the fraction of time in IDA 4 (carbon dioxide concentration higher that 1000 ppm) is considered and fourth measured by the dose over 1000 ppm normalized to the time in IDA 4.

2.4.2 Air Change Rate

Figure 2.11 shows the cumulative distribution of the air change rate in the semi-detached dwelling. The median as well as first and third quartile values for all dwellings and all standards are listed in Table 2.9. A clear distinction is seen between the Belgian and Dutch standards on the one hand and the British, French and ASHRAE standards on the other. The air change rate in the latter group is much less susceptible to variation due to changing boundary conditions due to the smaller sizing of trickle ventilators or the absence thereof compared to the Belgian and Dutch standard, which require relatively large trickle ventilators. The Belgian, Dutch, French and ASHRAE standards all achieve median air change rates close to 0.5 ACH, which, as was mentioned in the introduction, can be considered a consensus value for residential buildings, while the system sized according to the British standard consistently renders about 40% lower values.

Table 2.8 Average carbon dioxide concentration to which the occupants are exposed, carbon dioxide dose over 1000 ppm normalized to the total simulation time and to the time over 1000 ppm considering all 334 occupants, as well as average ventilation heat loss from the 100 simulations in the Monte-Carlo analysis for all 5 standards in all 5 geometries.

		Belgium	France	Netherlands	UK	ASHRAE
Flat	Average CO_2 (Δppm)	351	283	358	521	342
	Dose CO_2 > 1000 ppm / total time	8	1	8	18	1
	Dose CO_2 > 1000 ppm / time > 1000 ppm	373	334	281	159	59
	Average Heat Loss (kWh/(m^2·a))	26	30	29	18	35
Terraced	Average CO_2 (Δppm)	323	324	357	595	492
	Dose CO_2 > 1000 ppm / total time	7	2	11	49	34
	Dose CO_2 > 1000 ppm / time > 1000 ppm	275	215	236	295	270
	Average Heat Loss (kWh/(m^2·a))	30	27	32	17	26
Semi-Detached	Average CO_2 (Δppm)	333	317	312	529	370
	Dose CO_2 > 1000 ppm / total time	6	2	2	30	3
	Dose CO_2 > 1000 ppm / time > 1000 ppm	208	200	169	243	118
	Average Heat Loss (kWh/(m^2·a))	28	26	32	16	28
Detached	Average CO_2 (Δppm)	338	338	340	572	520
	Dose CO_2 > 1000 ppm / total time	9	3	12	51	52
	Dose CO_2 > 1000 ppm / time > 1000 ppm	235	131	251	344	396
	Average Heat Loss (kWh/(m^2·a))	33	29	37	19	27
Bungalow	Average CO_2 (Δppm)	304	342	326	577	523
	Dose CO_2 > 1000 ppm / total time	1	2	2	58	35
	Dose CO_2 > 1000 ppm / time > 1000 ppm	184	191	183	366	253
	Average Heat Loss (kWh/(m^2·a))	32	30	35	21	33

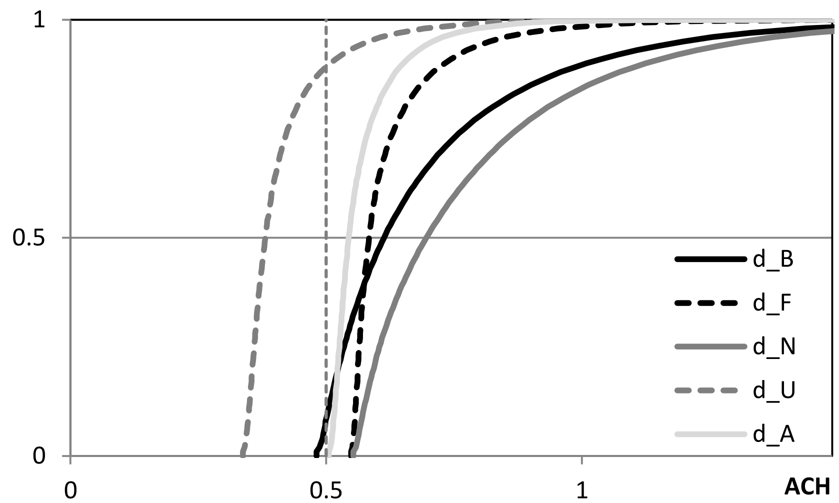

Figure 2.11 Cumulative distribution of the air change rate in the Semi-Detached dwelling for all 5 standards.

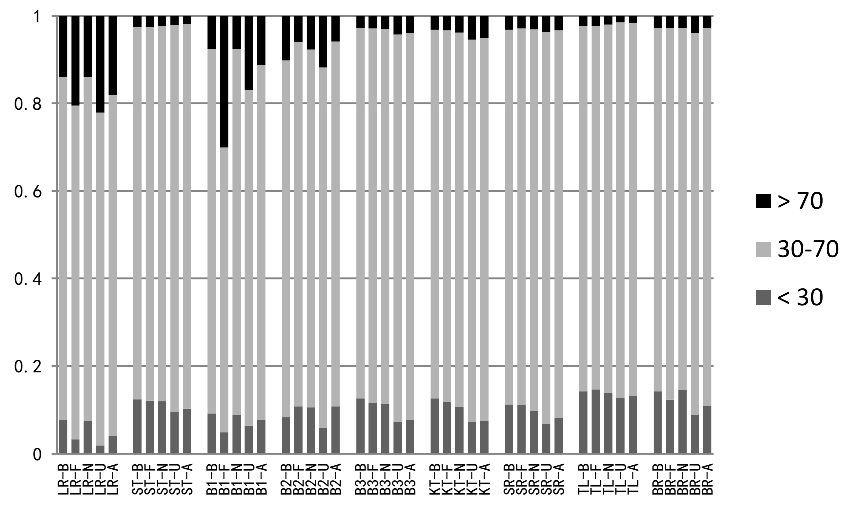

Figure 2.12 Relative humidity in all spaces of the Terraced dwelling for all 5 standards (Living Room – LR, Study – ST, Bedroom 1-3 – B1-3, Kitchen – KT, Service Room – SR, Toilet – TL, Bathroom – BR)

2.4.3 Relative Humidity

Relative humidity was within the acceptable range within the vast majority of time (80%) in almost all spaces in all dwellings and for all standards. As is shown in Figure 2.12, although moisture producing activities are concentrated in the 'wet' spaces, the highest frequency of excessive relative humidity is found in the living room and bedrooms. This is readily explained by the fact that exhaust systems mechanically assure a constant exhaust flow rate from these wet spaces.

2.4.4 Ventilation Heat Loss

Figure 2.14. and Figure 2.15 show the cumulative distribution of the ventilation heat loss for both the detached and the semi-detached house for all 5 standards, taking into account both intended and adventitious ventilation as well as infiltration. The same conclusions as with the air change rate apply.

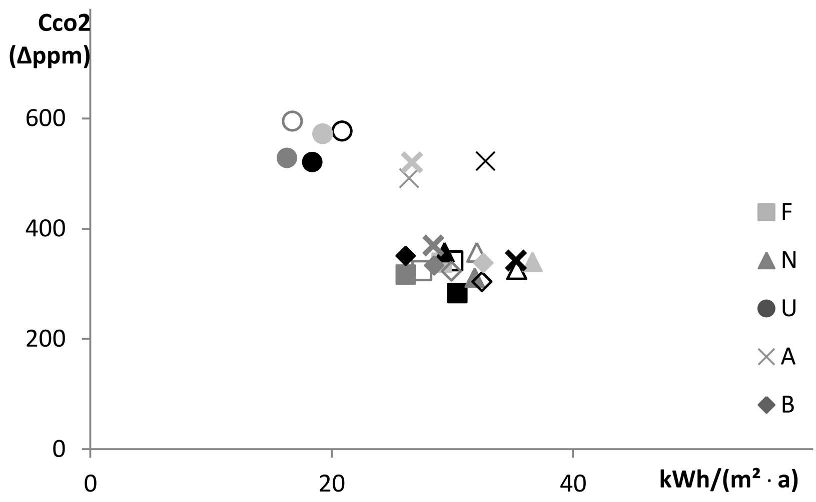

Figure 2.13 Average annual ventilation heat loss traded-off against average carbon dioxide concentration to which occupants are exposed during the heating season for all 5 standards in all 5 dwellings (flat – black solid fill, terraced – grey line, semi-detached – dark grey solid fill, detached – light grey solid fill, bungalow – black line, symbols correspond to the standards, as shown in the legend.)

If the trade-off between heat loss and indoor air quality is considered (Figure 2.13), using the average ventilation heat loss for the former and the average carbon dioxide to which the occupants are exposed as the criterion for the latter, the French and British standard provide pareto optimal solutions for each dwelling, although the fact that the indoor air quality achieved by the British standard is to be considered 'poor' 15% of the time is a cause of concern. Compared to the French standard, for example, the exposure to carbon dioxide of the cases using the ASHRAE standard is on average 40% higher, with higher or comparable heat losses (+16 to -8 %). Similarly, the ventilation heat loss in 4 cases using the Dutch standard is on average 20 % higher than that in the cases with the French standard for higher or comparable carbon dioxide exposure (+10 to -5%). In the flat, the heat losses using the French standard were comparable to those using the Dutch standard (+4%) with lower exposure to carbon dioxide (-26%).

2.4.5 Conclusions for Exhaust Ventilation

Sizing rules in residential ventilation standards lack uniformity in both methodology and resulting design flow rates. Mere comparison of design flow rates is case sensitive and, due to effects of infiltration, adventitious ventilation and occupancy, ill-suited to assess performance of a mechanical exhaust ventilation system with regard to the achieved indoor air quality and energy cost in terms of heat loss. A performance assessment of residential mechanical exhaust ventilation systems using five common dwelling typologies and the sizing rules put forward in the Belgian, British, Dutch, French and ASHRAE residential ventilation standards in multi-zone simulations with Monte-carlo based sensitivity analysis presented above showed that the performance of the different cases proved to be substantially different. An occurrence of poor perceived air quality in 5% or less of the occupation time for the Belgian, Dutch and French standard, and about 15% for the British and ASHRAE standard was found.

Except for the cases with the ASHRAE standard, the relative performance of the standards was consistent throughout the different building typologies. The spread observed in the performance of the cases using the ASHRAE standard can be attributed to the larger impact of geometrical parameters on the system design in this standard. In some cases, the relative performance of the standards was sensitive to the indoor air quality criterion used in the assessment, although the general trends could be observed with each of the criteria.

The total air change rate was close to or greater than the consensus value of 0.5 ACH in most cases, except in the cases using the British standard, where it was consistently about 40% lower. The cases using the Belgian and Dutch standards, with relatively large trickle ventilators, rendered the air change rates most sensitive to changes in boundary conditions. When the trade-off between indoor air quality and heat loss is considered, the cases with the Dutch and ASHRAE standard did not achieve pareto-optimal performance.

Considering the performance spread observed, harmonization of residential ventilation standards is to be recommended. The design philosophy of the French standard proves to be a good basis for mechanical exhaust ventilation design with high occurrence of good perceived air quality, minimized ventilation heat loss and robust performance. It’s combination of moderately high exhaust flow rates, large transfer devices and small trickle ventilators should explored further when new, more uniform standards are developed.

Table 2.9 Median, first quartile and third quartile air change rate from the 100 simulations in the monte-carlo analysis for all 5 standards in all 5 geometries.

		Belgium	France	Netherlands	UK	ASHRAE
flat	median	0.58	0.69	0.66	0.42	0.80
	Q1	0.57	0.68	0.65	0.41	0.79
	Q3	0.59	0.69	0.67	0.42	0.80
terraced	median	0.50	0.51	0.54	0.31	0.50
	Q1	0.46	0.51	0.50	0.31	0.50
	Q3	0.68	0.52	0.72	0.32	0.50
semi-detached	median	0.47	0.48	0.54	0.29	0.53
	Q1	0.42	0.47	0.48	0.29	0.53
	Q3	0.66	0.50	0.72	0.32	0.54
detached	median	0.64	0.59	0.72	0.39	0.55
	Q1	0.54	0.57	0.61	0.36	0.53
	Q3	0.85	0.67	0.96	0.45	0.61
bungalow	median	0.55	0.53	0.61	0.36	0.59
	Q1	0.47	0.51	0.54	0.33	0.57
	Q3	0.75	0.61	0.80	0.45	0.66

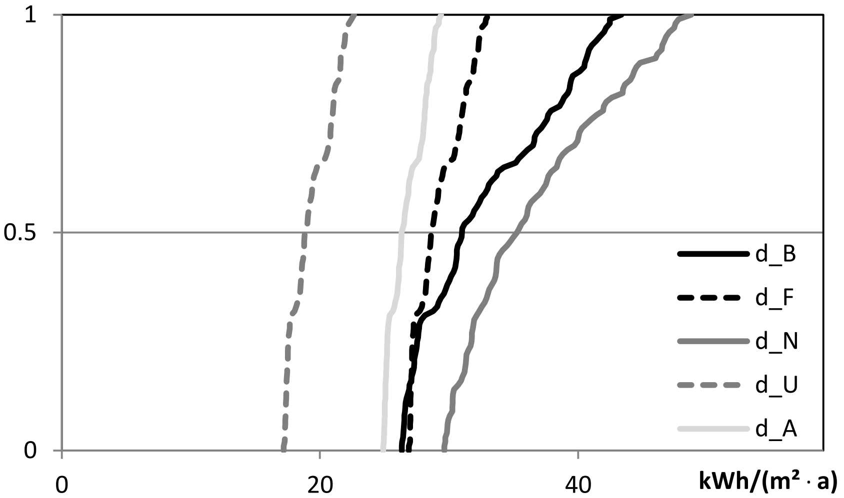

Figure 2.14 Cumulative distribution of ventilation heat loss in the detached dwelling for all 5 standards.

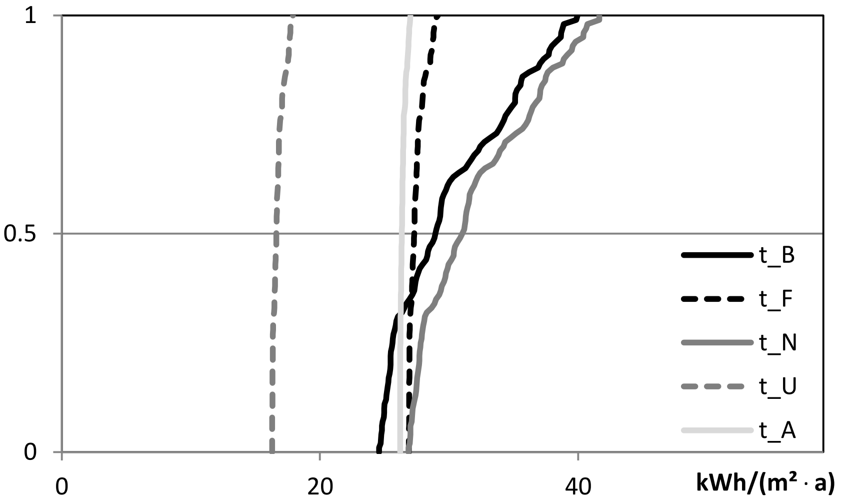

Figure 2.15 Cumulative distribution of ventilation heat loss in the terraced dwelling for all 5 standards.

2.5 Natural and Mechanical Ventilation

As was stated in the introduction of the chapter, the analysis presented above will be extended to natural and mechanical supply and exhaust systems in this section. The ASHRAE standard is not studied further; both because of the poor performance found in the previous section for systems sized in accordance with it and because it offers virtually no information on the design of ventilation systems other than mechanical exhaust ventilation. In its stead, the assessment of the design rules included in the German ventilation standard are included.

2.5.1 Modelling Assumptions

As the performance of mechanical exhaust ventilation systems was found to depend more on the sizing rules than on the geometry of the dwelling, the analysis of the performance of the natural and mechanical supply and exhaust systems is limited to the detached dwelling in this section. This type of dwelling is the most widespread typology for single family houses. In this specific segment of the residential construction market, ventilation system sizing, if it occurs at all, is most likely to be executed by simple application of the sizing rules in the relevant standard, because the project size and associated budget constraints usually prohibit more detailed engineering.

In contrast to the approach chosen in the previous section, local climate data were selected for each standard and envelope leakage was varied. Envelope leakage levels taken into account were 0.6, 3 and 6 $m^3/(h{\cdot}m^2)$ @ 50 Pa. For each Country, the climate file for the capital was chosen as a reference, while for the larger countries, a second city was selected to include the effect of climate variations within the countries. These additional cities are Lyon, Munich and Aberdeen for France, Germany and the UK respectively.

Sizing rules in the standards are usually not 100% consistent. The sizing adopted for the different standards is interpreted to be as realistic as possible. The most relevant adaptation is that, for mechanical supply and exhaust, the supply or exhaust flows, depending on which of both had the smallest total flow rate, were proportionally increased to make sure that the system operated with balanced flow rates. For the French standard, the exhaust flow rates were lowered to correspond to the flow rates usually selected in practice due to a loophole in the text that allows to reduce these rates when purge ventilation (eg. a cooker hood) is available. For the UK, the flow rates specified in the last update of the standard were used.

2.5.2 Results

In this section, the results for the different countries and the different system configurations will be reported. First, the ventilation heat loss associated with the different sizing options and system approaches is presented. Then the impact on the achieved indoor air quality is discussed.

Table 2.10 Design flow rates in l/s at 1 Pa for system components in the reference dwelling. Mechanical flow rates (l/s) are marked in italics

		Belgium			The Netherlands			France			Germany			UK		
		Nat.	Exh.	Mech.	Nat.	Exh.	Mech.	Nat.	Exh.	Mech.	Nat.	Exh.	Mech.	Nat.	Exh.	Mech.
Supply	Living room	25.2	25.2	*35.7*	32.1	32.1	*32.1*	3.9	1.3	*5.8*	2.7	4.4	*12.5*	27.6	2	*17.7*
	Study	5.7	5.7	*8*	7.2	7.2	*7.2*	2.6	1.3	*5.8*	1.4	2.3	*6.4*	11.8	2	*4*
	Bedroom 1	12	12	*17*	15.3	15.3	*15.3*	2.6	1.3	*5.8*	1.9	2.9	*8.3*	11.8	2	*8.4*
	Bedroom 2	12.9	12.9	*18.2*	16.4	16.4	*16.4*	2.6	1.3	*5.8*	1.9	2.9	*8.3*	11.8	2	*9*
	Bedroom 3	12.9	12.9	*18.3*	16.5	16.5	*16.5*	2.6	1.3	*5.8*	1.9	2.9	*8.3*	11.8	2	*9.1*
transfer	Living room	4.9	4.9	4.9	32.1	32.1	32.1	6.2	6.2	6.2	11.9	10.2	5.4	6	6	6
	Study	4.9	4.9	4.9	7.2	7.2	7.2	6.2	6.2	6.2	5.8	5.2	3.6	6	6	6
	Bedroom 1	4.9	4.9	4.9	15.3	15.3	15.3	6.2	6.2	6.2	8.1	6.8	5.4	6	6	6
	Bedroom 2	4.9	4.9	4.9	16.4	16.4	16.4	6.2	6.2	6.2	8.1	6.8	5.4	6	6	6
	Bedroom 3	4.9	4.9	4.9	16.5	16.5	16.5	6.2	6.2	6.2	8.1	6.8	5.4	6	6	6
	Kitchen	9.8	9.8	9.8	21	21	21	12.4	12.4	12.4	13.3	11.6	6.4	6	6	6
	Service room	4.9	4.9	4.9	7	7	7	6.2	6.2	6.2	7.5	6.4	6.4	6	6	6
	Bathroom	4.9	4.9	4.9	14	14	14	6.2	6.2	6.2	13.3	11.6	6.4	6	6	6
	Toilet	4.9	4.9	4.9	7	7	7	6.2	6.2	6.2	7.5	6.4	6.4	6	6	6
Exhaust	Kitchen	9.8	*14*	*14*	21	*21*	*21*	3.9	*12.5*	*12.5*	11.8	*13.1*	*14.2*	9.4	*18*	*18*
	Service room	9.8	*14*	*14*	7	*7*	*7*	1.3	*5*	*5*	11.8	*7.2*	*7.8*	9.4	*11*	*11*
	Bathroom	9.8	*14*	*14*	14	*14*	*14*	2.6	*9*	*9*	11.8	*13.1*	*14.2*	9.4	*11*	*11*
	Toilet	4.9	*7*	*7*	7	*7*	*7*	1.3	*5*	*5*	11.8	*7.2*	*7.8*	9.4	*8*	*8*

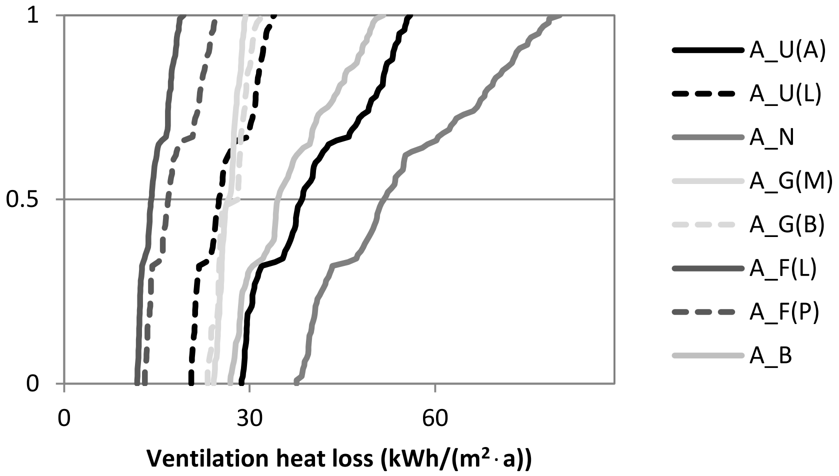

Figure 2.16 Cumulative distribution of ventilation heat loss for natural ventilation sized in accordance with 5 European residential ventilation standards (8 weather conditions) with an envelope leakage of 3 $m^3/(h{\cdot}m^2)$

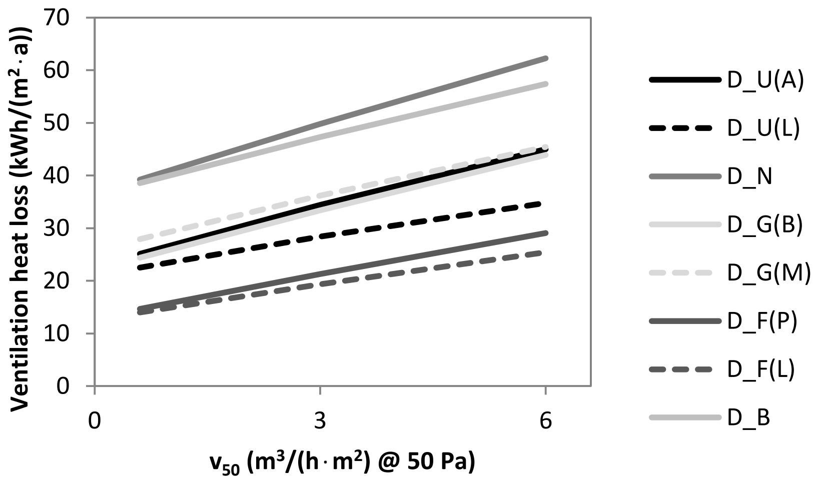

Figure 2.17 Average ventilation heat loss for mechanical supply and exhaust sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

Ventilation Heat Loss

The ventilation heat loss for natural ventilation and mechanical supply and exhaust for the different standards and climates is shown in Figure 2.16 and Figure 2.17 respectively for the three leakage levels proposed in the introduction. The ventilation heat loss is increased more or less linearly with increasing leakage due to growing infiltration of outdoor air associated with it. The order of the ventilation heat loss follows the order of the total design flow rate very closely for the mechanical supply and exhaust systems. For natural ventilation systems, however, a lot more variability is observed, due to the larger impact of the weather on the driving forces of the ventilation (wind pressure and buoyancy). Internal flow resistances will have a much larger impact on the total flow in natural ventilation, causing different slopes in the relation between ventilation heat loss and leakage for different standards or the large difference in ventilation heat loss between the Dutch and Belgian natural ventilation systems where it is more or less equal for mechanical supply and exhaust systems.

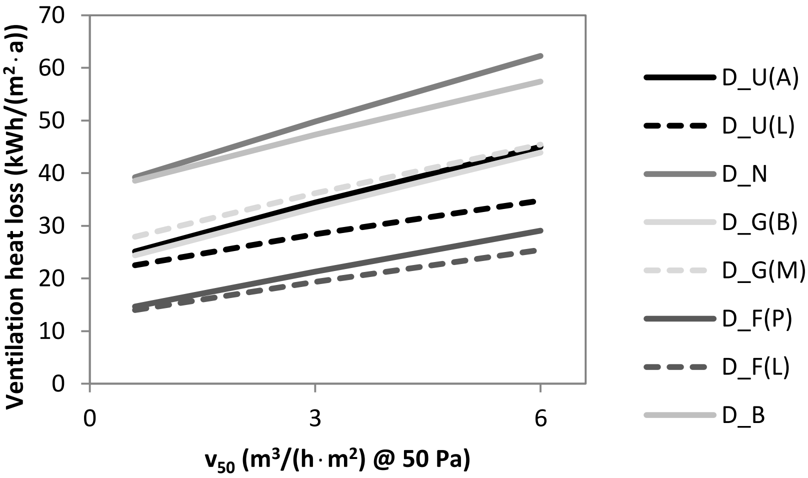

Figure 2.18 Ventilation heat loss for mechanical exhaust ventilation sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

The more or less linear increase of ventilation heat loss with increasing leakage is not observed in about half of the mechanical exhaust ventilation cases (Figure 2.18). In these cases, the trickle ventilators are relatively small with respect to the exhaust flow rate. Since the latter is invariable, lower leakage levels lead to increased under pressure in the dwelling which in turn reduces the amount of infiltration heat loss on top or the mechanical ventilation heat loss, as defined in chapter 2. The Dutch standard generates the largest ventilation heat losses for all systems, the modified French standard leads to the lowest losses.

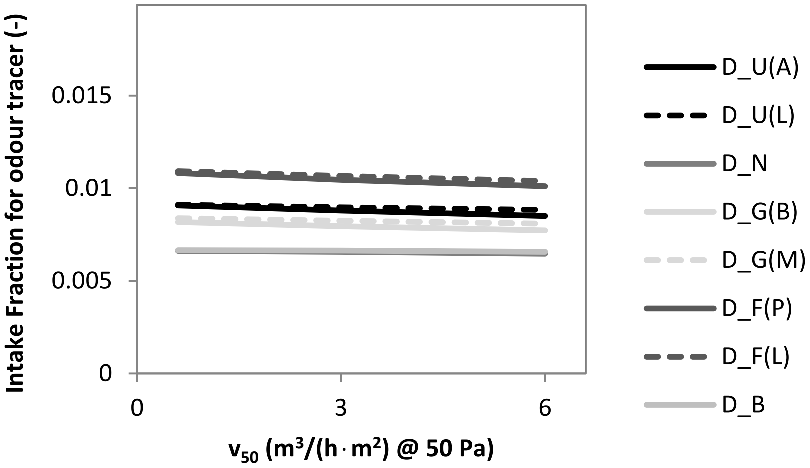

Figure 2.19 Odour tracer exposure for natural ventilation sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

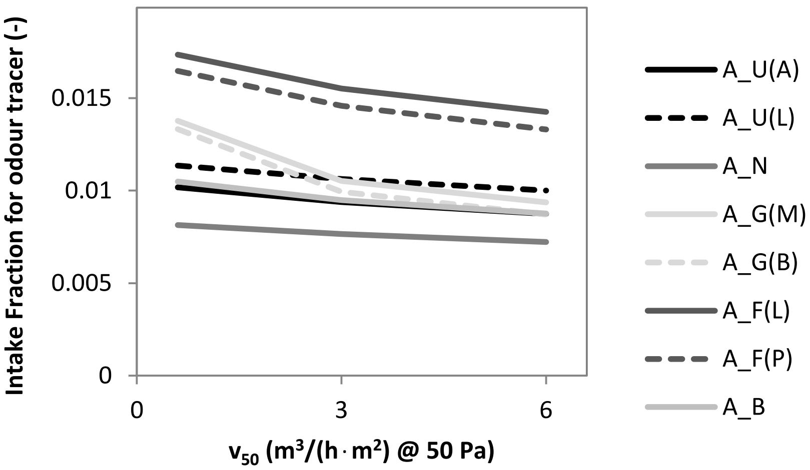

Figure 2.20 Odour tracer exposure for mechanical supply and exhaust sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

Indoor Air Quality

Similar effects can be seen in the results for exposure to indoor air pollutants. The exposure of the occupants to odours produced in the toilet (Figure 2.19and Figure 2.20) is substantially less elevated in mechanical supply and exhaust cases than in natural ventilation cases, regardless of the leakage level. Additionally, the results are far less sensitive to the leakage level since it is highly unlikely that the flow direction will be reversed in mechanical supply and exhaust systems, while in natural ventilation systems, the interaction between different driving forces causes such situations regularly.

The mean carbon dioxide concentration to which the occupants are exposed during occupancy follows the theoretical exponential decrease with increasing infiltration and ventilation, as can be seen in Figure 2.22 and Figure 2.23. Nonetheless, natural ventilation systems demonstrate sub optimal performance since they achieve a lower indoor air quality (higher mean concentration) at higher ventilation heat losses compared to the mechanical supply and exhaust systems. Again, this is easily explained by the higher variability of the flow rates in natural ventilation systems, leading to the occurrence of relatively large peak exposures. Figure 2.21 demonstrates this higher variability for the Belgian case by showing the average and $P_{0,95}$ concentration to which occupants are exposed.

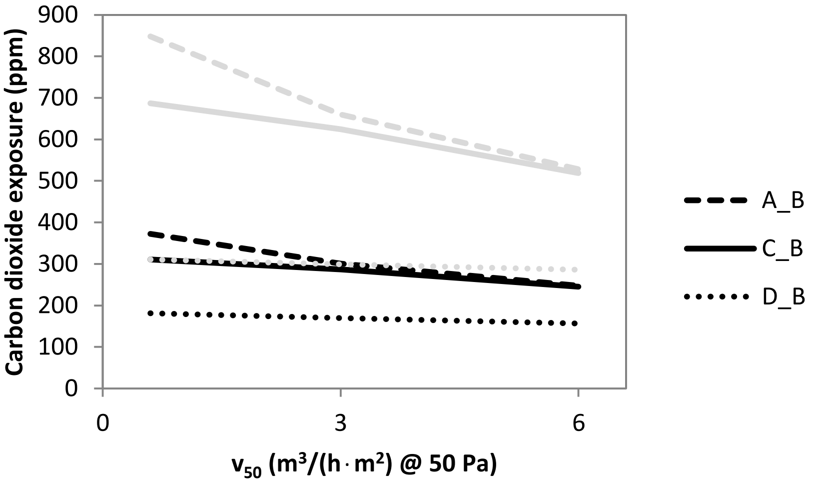

Figure 2.21 Average (black) and $P_{0,95}$ (grey) exposure to carbon dioxide for natural, mechanical exhaust and mechanical supply and exhaust ventilation sized in accordance with the Belgian residential ventilation standard at different envelope leakage levels

The latter is also very clear from the peak exposure results shown in Figure 2.24 and Figure 2.25. The exposures shown are heating season cumulated doses of exposure to carbon dioxide at concentrations exceeding 1000 ppm above the outdoor concentration. The peak exposures in natural ventilation cases rapidly increase with a tightening building envelope for natural ventilation systems, while they are virtually absent for mechanical supply and exhaust.

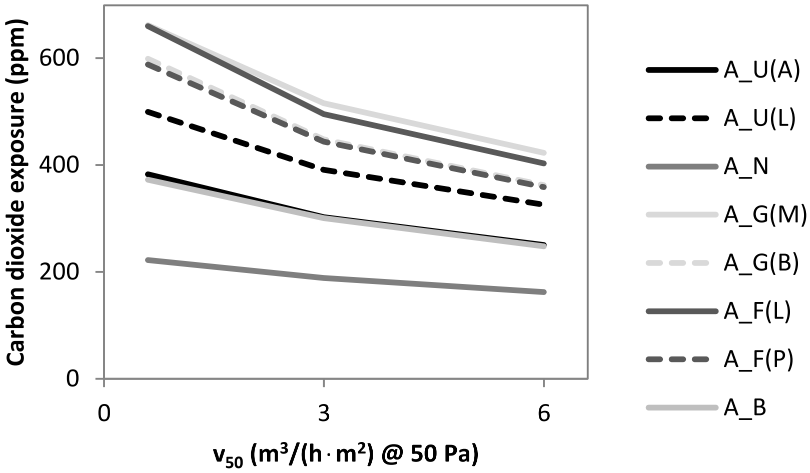

Figure 2.22 Average exposure to carbon dioxide for natural ventilation sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

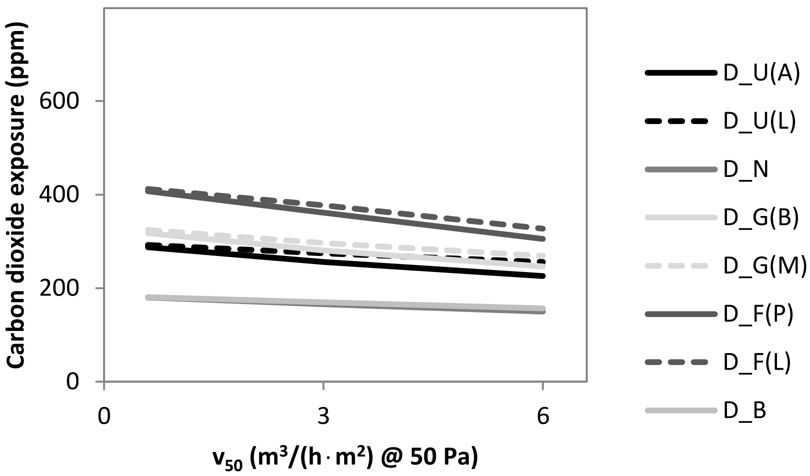

Figure 2.23 Average exposure to carbon dioxide for mechanical supply and exhaust sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

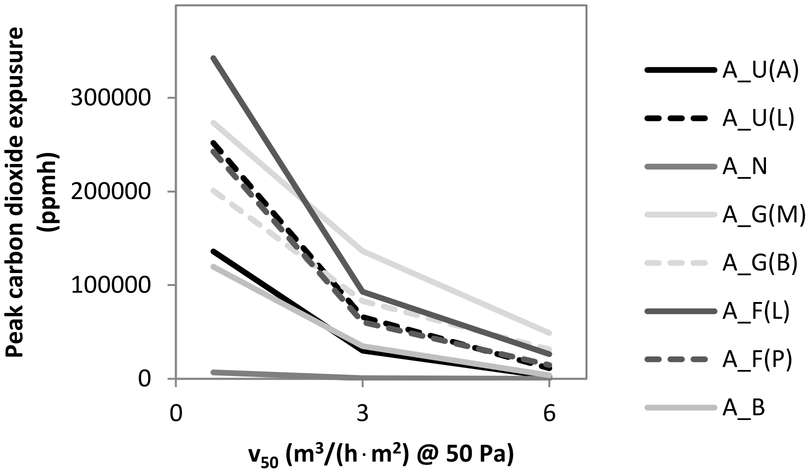

Figure 2.24 Peak exposure to carbon dioxide for natural ventilation sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

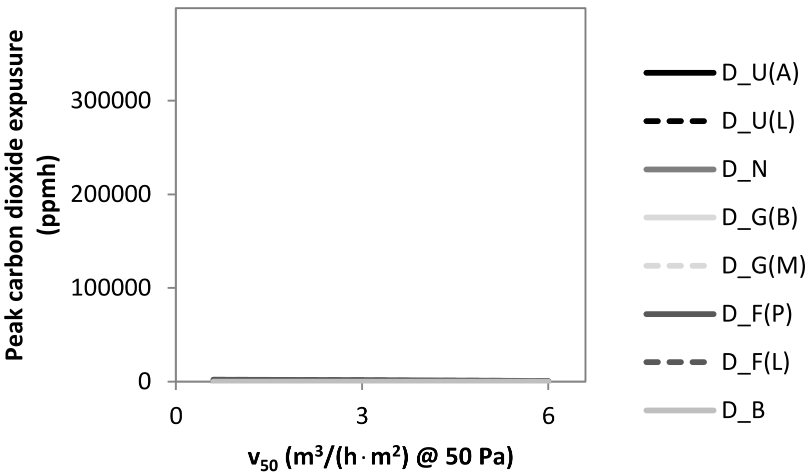

Figure 2.25 Peak exposure to carbon dioxide for mechanical supply and exhaust sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

Mechanical exhaust ventilation displays interesting behaviour (Figure 2.26) with increasing air tightness. Although total air flows and ventilation heat losses decrease with lower leakage levels, the peak exposure is also reduced. This can be explained by the under pressure generated by the exhaust fan. In tight buildings, the under pressure will be larger, cancelling out other, more variable, driving forces, and therefore the flows within the dwelling will be more stable. These more stable flows are aligned with the design intentions, causing fresh air to enter the densely occupied spaces (living room and bedrooms).

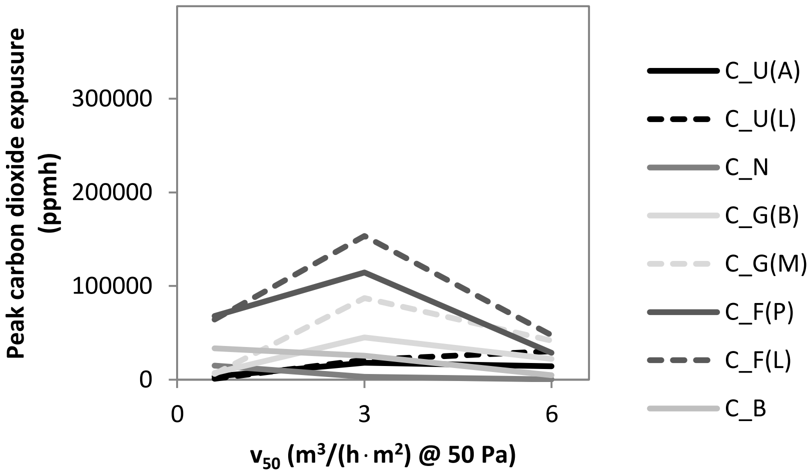

Figure 2.26 Peak exposure to carbon dioxide for mechanical exhaust ventilation sized in accordance with 5 European residential ventilation standards (8 weather conditions) with different envelope leakage levels

In leaky conditions, wind and buoyancy are the dominant driving forces, causing air to exit the building through trickle ventilators located on the leeward side or on the second floor (above the neutral pressure plane) of the dwelling. These rooms are then ventilated with air potentially polluted in other spaces of the building, thereby increasing the exposure of the occupants. In between those two extremes, maximum exposure is reached when mechanical and climate related driving forces cancel each other out, especially for spaces situated on the second floor. The leakage level that corresponds to this point is determined by the relative magnitude of the exhaust rates and the trickle ventilators. For the French and German standard, this is at 3 $m^3/(h \cdot m^2)$ @ 50 Pa, whereas for the British standard this point was already reached at 6 $m^3/(h \cdot m^2)$ @ 50 Pa. For the Dutch standard, it is not even reached at 1 $m^3/(h \cdot m^2)$ @ 50 Pa. Similar but less pronounced effects are seen considering average exposure.

In comparison, the ventilation heat loss as well as the mean and $P_{0,95}$ carbon dioxide concentrations for the British and Dutch standard, at both ends of the sizing spectrum, are shown in Figure 2.27 and Figure 2.28.

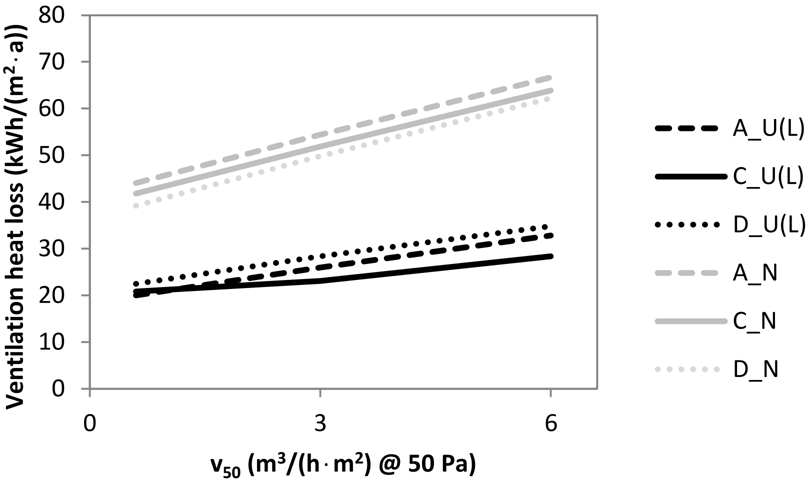

Figure 2.27 Ventilation heat loss for natural, mechanical exhaust and mechanical supply and exhaust ventilation sized in accordance with the Dutch and British residential ventilation standard at different envelope leakage levels

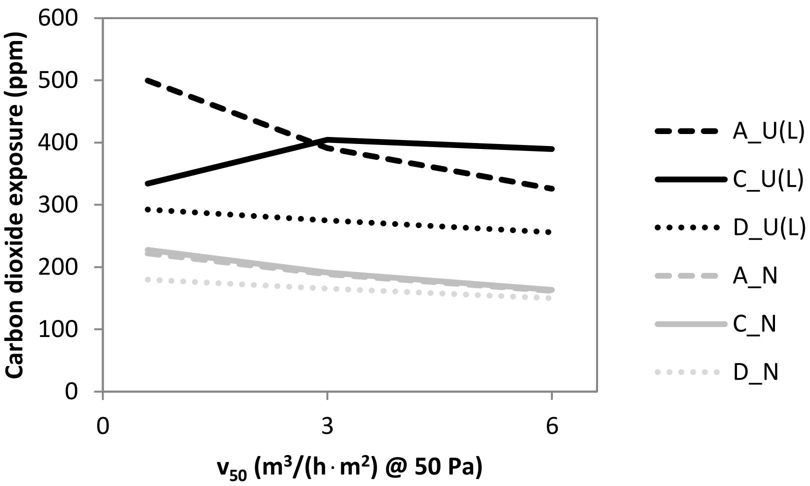

Figure 2.28 Average exposure to carbon dioxide for natural, mechanical exhaust and mechanical supply and exhaust ventilation sized in accordance with the Dutch and British residential ventilation standard at different envelope leakage levels

2.6 Conclusions

The objective of this chapter was to assess the performance of ventilation systems sized in accordance with the disparate sizing rules included in contemporary residential ventilation standards, thereby establishing a ‘state of the art’ performance level.

Since mere comparison of design flow rates is case sensitive and, due to effects of infiltration, adventitious ventilation and occupancy, ill-suited to assess performance of residential ventilation systems with regard to the achieved indoor air quality and energy cost in terms of heat loss, a more detailed methodology was required. A multi-zone simulation model supplemented with Monte Carlo sensitivity analysis was presented and proposed as a tool for this analysis.

With the proposed methodology, mechanical exhaust ventilation systems were dealt with first, since their system layout is rather straightforward and therefore well described in most ventilation standards. This performance assessment of residential mechanical exhaust ventilation systems used five common dwelling typologies and the sizing rules put forward in the Belgian, British, Dutch, French and ASHRAE residential ventilation standards. The performance of the different cases proved to be substantially different, with an occurrence of poor perceived air quality in 5% or less of the occupation time for the Belgian, Dutch and French standard, and about 15% for the British and ASHRAE standard.

The analysis was then expanded to that of both natural and mechanical supply and exhaust. In this part, the ASHRAE standard was replaced by the German standard because of both the poor performance of its design rules found in the first part of the chapter and the lack of well described system layout for natural and mechanical supply and exhaust systems. Envelope leakage and local weather were introduced as supplementary variables in the analysis. Similar differences in performance between the standards were found. Mechanical supply and exhaust was shown to be more robust than natural or mechanical exhaust ventilation across all standards. A clear interaction between the performance of mechanical exhaust ventilation and envelope leakage was found.

3

Achievable Performance: Sizing

The disparity in the performance of ventilation systems sized in accordance with different standards found in chapter 2 suggests that a range design alternatives are available with equivalent, so called Pareto performance while other design options fail to achieve this level of performance. This in turn raises the question whether the performance of ventilation systems can be improved by selecting design approaches that differ from the ones suggested in current standards, as well as which general design rules render the most robust results. This chapter addresses these questions by presenting an optimization study based on an adaptation of the models presented in the previous chapter.

The potential systemic differences in performance of natural, mechanical exhaust and mechanical supply and exhaust ventilation in dwellings is the object of relentless debate among scientists, industry and policy makers, although comparisons found in literature often fail to compare the different systems on an equal basis. Presenting the results from a multi zone simulation based optimization study of residential ventilation design flow rates and sizing of the system components published in Building and Environment, this chapter aims to provide a benchmark for achievable performance for the different systems for moderate climate regions (eg. Western Europe), as well as point to possible sizing strategies for future standards.

3.1 Introduction

The air flow rates required in residential ventilation standards all over the Western countries, as well as the sizing required for non-mechanical system components such as trickle ventilators and transfer grilles [399], can vary quite considerably, with whole building air change rates ranging from 0.3 to 1 [231] for dwellings.

In addition, as was explained in the first chapter, due to differences in climate, boundary conditions, occupant behaviour and consumer preference, three main types of residential ventilation systems are dominant in the Northern, Western and Southern part of Europe respectively. Balanced mechanical heat recovery ventilation is the most common system in Scandinavia [182]. In the moderate climate region, mechanical exhaust systems are more popular [317], while natural ventilation is widespread in Southern Europe.

The performance of the different systems [282, 400, 401] and approaches to sizing of their components that are put forward [363] is the object of relentless debate. Based on multi zone simulations, an optimization study of residential ventilation design flow rates and sizing of the system components is presented in this chapter. The first section covers the used methodology, while the results section outlines the achievable performance for the different systems for moderate climate regions (e.g. Western Europe), as well as points to possible sizing strategies for future developments in standards. The results of this chapter have been published in Building and Environment [402].

3.2 Methods

The results presented in this chapter are based on the same simulation approach as was presented in chapter 2. Due to the large number of cases to be considered, as will be explained more in detail in the subsequent sections, computational time per simulation had to be limited to keep the total simulation time within reasonable limits. Therefore simulations were not run for the whole heating season, but for the 3rd week of December, with an average outdoor temperature (7.2 °C), wind speed (4.9 m/s), wind direction (210°) and humidity ratio (5.6) that is representative for the typical operating conditions of ventilation systems in a moderate climate region. This week provided the best fit with the average values for the whole heating season of all weeks in the heating season of the test reference year considering any combination of the following meteorological parameters: wind speed, wind direction, humidity ratio and temperature. A full week is chosen to assure sufficient variation in the climate as well as to include both week and weekend days in the occupancy. Table 3.1. lists the means of these characteristics for the selected week and the full heating season.

Table 3.1 average climate data parameters for the selected week and full heating season

	3rd Week of december	Heating Season
Wind speed (m/s)	4.9	5.1
Wind direction (°)	210	209
Temperature (°C)	7.2	6.4
Humidity ratio (g/kg)	5.6	5.2

3.2.1 Building Model

The detached and flat geometries presented in the previous chapter were used. In the results presented, a uniform specific airleakage (v_{50}) of 1 and 12 m³/h/m² is adopted for the envelope of the detached dwelling, so the impact of leakage on achievable performance can be assessed. For the flat, the specific leakage is set to 3 m³/h/m², corresponding to the best quartile of measured airtightness values in a measurement campaign in Belgium in the late 90's [298]. A recent measurement campaign [386], along with results from other countries [296], shows a tendency towards this level of airtightness in newly built dwellings. At this leakage level, the flat therefore represents the current and near future building practice.

Ventilation System Design and Model

The ventilation scheme used in both dwelling geometries is based on the sizing rules put forward in the Belgian residential ventilation standard [364]. This standard is chosen because it contains clear and simple sizing rules for natural, mechanical exhaust and mechanical supply and exhaust ventilation and all relevant components in them.

The Belgian standard requires a design fresh air flow rate of 1 l/(s·m^2) for each occupied space. For kitchens, bathrooms and service rooms, a minimum design flow rate of 14 l/s should be taken into account. The design flow rate for a toilet is 7 l/s. The occupied spaces and the wet spaces should be connected to each other or via circulation spaces by transfer grilles sized at 7 l/s at 2 Pa pressure difference, which corresponds to 70 cm², except for the kitchen, in which the transfer grille should be sized twice as large. For natural and mechanical exhaust ventilation systems, supply trickle ventilators and exhaust grilles should be sized at the design flow rate at 2 Pa pressure difference.

In the simulation study, the design flow rates for supply, transfer and exhaust as mentioned above, are varied from 10 % to 200 % of the original design flow rates in 10 % steps in order to assess the optimal performance of the 3 main ventilation system approaches discussed in the introduction. Since the flow rates proposed in the Belgian standard are moderate to high in comparison to other residential ventilation standards [231, 399], this covers a broad range of sizing options that are both realistic and within ranges used in existing standards.

All mechanical vents were modelled as constant volume flow rate components in the respective zone node, while transfer grilles and trickle ventilators were modelled with single direction power law flow components with a flow exponent of 0.5 [270].

All systems were modelled with windows and internal doors closed, in order to simulate the performance of the systems as such, without user interaction. A summary of the sizing and parameterization of the different components is given in Table 3.2.

3.2.2 Assessment Parameters

From the parameters suggested in chapter 2, two were selected for use in this chapter, namely the heating season average CO_2 concentration above the outdoor concentration (excess CO_2) to which an occupant is exposed and the dose of CO_2 over 1000 ppm excess CO_2. Exposure to concentrations in excess of 1000 ppm excess CO_2 is considered to correspond to poor perceived indoor air quality and is therefore a good indicator for the ability of the system to protect the occupants against pollution peaks.

The exposure to elevated humidity levels is an indicator for exposure to pollutants related to chemical and biological activity in the building. In literature the association of growth of fungae [111] and offgassing of building materials [103, 105, 403] with elevated humidity levels is extensively discussed. In this chapter, the average relative humidity to which the occupants are exposed is preferred over the comfort zone defined in the previous chapter because, for optimisation purposes, a single numerical value is easier to work with.

Again, the exposure to building material emissions themselves is not assessed since it is best controlled by either source control or intensive local ventilation. The ability of a sizing option to contain nuissant contaminants produced in the 'wet' spaces, such as odours, within these spaces and prevent their spreading through the dwelling is considered through the intake fraction of an odour tracer, as was described in chapter 2.

The total convective heat loss through the combination of mechanical ventilation, adventitious ventilation and infiltration is used to assess the energy performance of the different sizing options. Fan power was not taken into account because it is very system specific. Since the simulations are only run over a single week instead of the complete heating season, but with similar average temperatures, the ventilation heat loss is scaled proportional to time to the complete heating season, so that the obtained values can be compared to those in the other chapters.

3.3 Results

In the following paragraphs, the results of the simulation study are presented. In the first section, the Pareto optimal performance of natural, mechanical exhaust and mechanical supply and exhaust ventilation is presented with respect to different performance parameters.

The subsequent section addresses the impact of leakage on the performance of the ventilation system, while the last section explores general sizing rules to achieve optimal or close to optimal performance.

Table 3.2 Summary of airflow link parameters

		Natural (A)	Mechanical Exhaust (C)	Mechanical Supply and Exhaust (D)
Envelope Leakage Detached ($m^3/h/m^2$ @ 50 Pa)	1 and 12			
Envelope Leakage Flat ($m^3/h/m^2$ @ 50 Pa)	3			
Interior wall leakage ($m^3/h/m^2$ @ 50 Pa)	2			
Interior door leakage (m^3/h @ 50 Pa)	45			
Supply	1-20	$l/(s \cdot m^2)$ @ 2 Pa	$l/(s \cdot m^2)$ @ 2 Pa	$l/(s \cdot m^2)$
Transfer	0.7-14	l/s @ 2 Pa	l/s @ 2 Pa	l/s @ 2 Pa
Transfer (Kitchen)	1.4-28	l/s @ 2 Pa	l/s @ 2 Pa	l/s @ 2 Pa
Exhaust	1.4-28	l/s @ 2 Pa	l/s	l/s
Exhaust (Toilet)	0.7-14	l/s @ 2 Pa	l/s	l/s

3.3.1 Achievable Performance

As was discussed above, the sizing of supply, transfer and exhaust components was, each in 20 steps, varied within a realistic range. This amounts to $8 \cdot 10^3$ cases for every system concept. From these sizing cases, the achievable performance is assessed based on the proposed criteria.

Ventilation is always faced with a trade-off between indoor air quality and associated ventilation heat loss. Therefore, the achievable performance is defined as the set of Pareto optimal sizing cases for a specific indoor air quality criterion. Pareto optimal cases are cases where none of the other cases achieve better results on both indoor air quality and heat loss. Any case for which another case exists that has lower exposure and lower ventilation heat loss is called a dominated case and is considered to be sub optimal, since an unambiguously better solution exists. It is therefore rejected and not included in the Pareto front depicted in the graphs. The collection of cases shown in the graphs thus corresponds to the lower left most boundary of the cloud of results found for all cases.

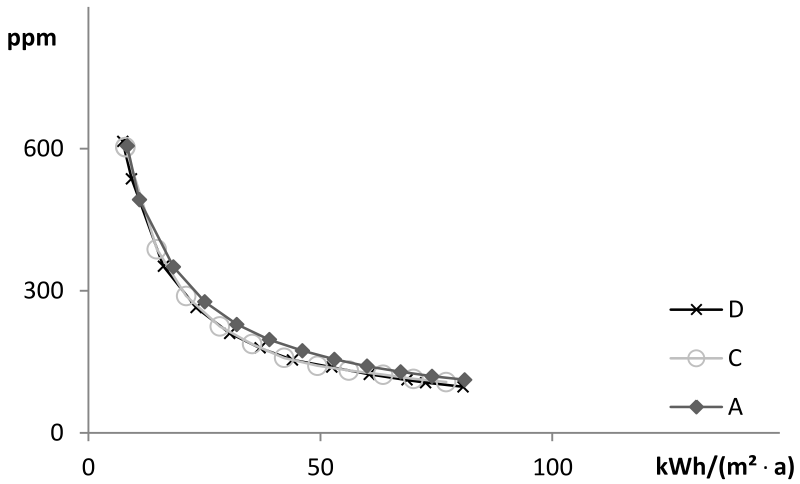

Figure 3.1 Total ventilation heat loss and average exposure to carbon dioxide of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling

Exposure to Carbon Dioxide

Figure 3.1 - Figure 3.3 show the Pareto fronts formed by the Pareto optimal cases for the airtight detached house with natural ventilation, mechanical exhaust ventilation and mechanical supply and exhaust ventilation, using occupant exposure to carbon dioxide as indoor air quality criterion. In this and all subsequent pictures, for readability, only 10-20 cases on the front are indicated with a marker, while the line traces the contour of the front, usually composed of several hundred cases.

As was mentioned in the methods section, two separate parameters were initially selected to characterize the exposure: the average concentration to which occupants are exposed and the total dose over 1000 ppm excess carbon dioxide. When the former is considered, the Pareto optimal solutions for the different installation concepts are rather similar (Figure 3.1).

Considering the latter, however, two striking aspects are observed: in a large number of cases, the occupants are not exposed to excess carbon dioxide concentrations higher than 1000 ppm, regardless of the system concept, while the optimal cases of the different system concepts demonstrate larger discrepancies (Figure 3.3). The exposure to high carbon dioxide concentrations for an optimal case of the natural ventilation system is on average 4.5 times higher than that for an optimal case of the mechanical supply and exhaust system at the same level of heat loss, while this increase was only about 15% when considering average exposure. The fact that the dose is zero for a large number of cases prohibits the use of this criterion for optimization purposes. Therefore, an alternative criterion, the dose over 600 ppm excess carbon dioxide concentration, corresponding to the next IDA class, IDA 3, is calculated. Although considerably smaller than with the original criterion for the natural ventilation system, the number of mechanical exhaust and mechanical supply and exhaust ventilation cases with zero dose remains quite high (Figure 3.2).

The performance of the systems is alternatively compared for equal exposure. The heat loss for the average optimal case of the natural ventilation system is 13, 19 and 26 % higher than the heat loss for the average optimal case of the mechanical supply and exhaust system at equal exposure, considering average exposure, dose over 600 ppm excess carbon dioxide concentration and dose over 1000 ppm excess carbon dioxide concentration, respectively.

The differences between the spread in optimal performance found with the different criteria is readily explained by the higher variability of the air flow in the natural system. This increases the exposure to peak concentrations, while the average is less affected.

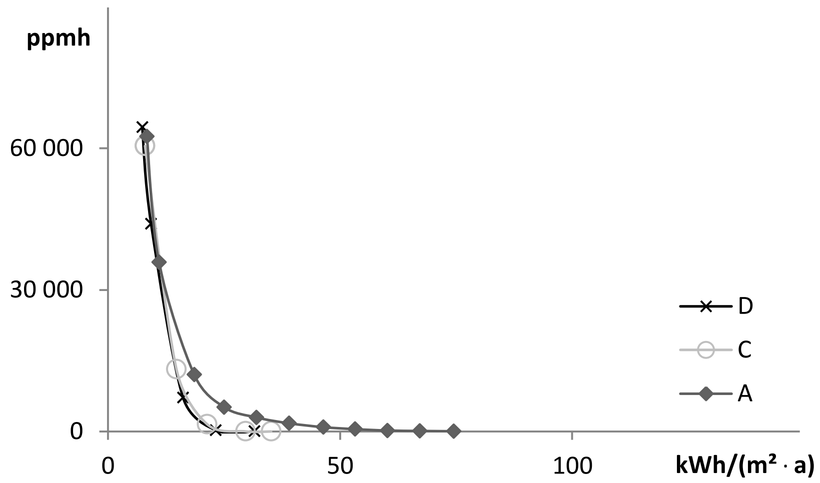

Figure 3.2 Total ventilation heat loss and exposure to carbon dioxide concentrations in excess of 600 ppm above the outdoor concentration of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling

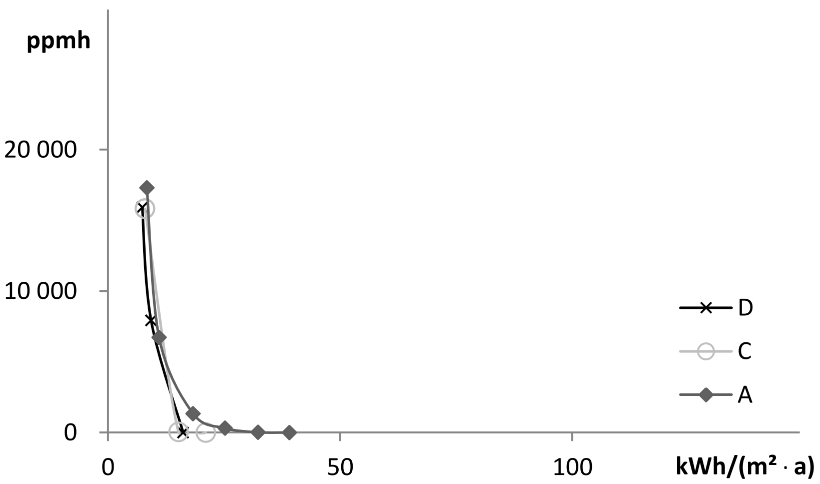

Figure 3.3 Total ventilation heat loss and exposure to carbon dioxide concentrations in excess of 1000 ppm above the outdoor concentration of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling

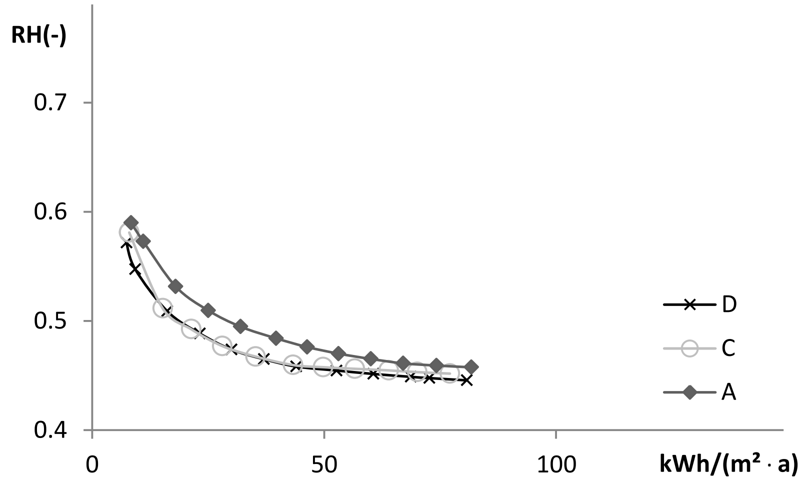

Figure 3.4 Total ventilation heat loss and average exposure to relative humidity of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling

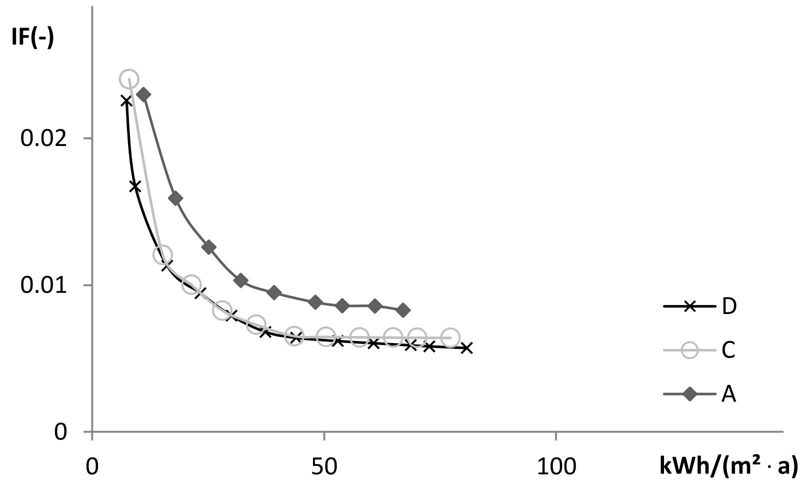

Figure 3.5 Total ventilation heat loss and intake fractions of odour tracer of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling

Humidity and Odours

In contrast to the relatively equal performance of the different system concepts observed when considering exposure to metabolic carbon dioxide, the performance of the natural ventilation concept is considerably inferior to that of the mechanical exhaust and the mechanical supply and exhaust ventilation concept when exposure to relative humidity (Figure 3.4) and to odours (Figure 3.5) is considered. Again, this is readily explained by the higher variability of the air flow in the natural ventilation system. Not only will the magnitude of the flow rate fluctuate more, the sense of the flow through the wet spaces of the dwelling also depends on the changing boundary conditions. Humidity and odour production are concentrated in these spaces. The occurrence of reversed flow from the wet spaces to the living spaces, known as backdraft, will increase the concentration of these pollutants in the living spaces, with, considering the long exposure time in these spaces, higher exposure as a consequence.

3.3.2 Leakage

Figure 3.6 and Figure 3.7 show the Pareto optimal cases for the airtight and the leaky detached dwelling with natural and mechanical supply and exhaust ventilation, respectively. For both system concepts, the optimal performance of the airtight cases is superior to that of the leaky cases, although the difference is more marked for the mechanical supply and exhaust system. Airtight dwellings with larger size ventilation systems (right side of the Pareto front) achieve similar airflow rates as leaky dwellings with smaller sized systems (left side of the Pareto front). The Pareto optimal airtight cases however achieve, at the right side of the Pareto front, the same average exposure to carbon dioxide at 10-32% and 12-25% lower ventilation heat losses compared to the Pareto optimal leaky cases, at the left end of that Pareto front, for the mechanical supply and exhaust and natural ventilation systems, respectively.

With mechanical supply and exhaust ventilation in a leaky building some of the supply air exfiltrates through the building envelope in the living spaces, while some of the exhaust air has infiltrated the wet spaces. In consequence, the total fresh air flow rate increases while the air change rate of each individual room, and therefore the exposure to carbon dioxide in the living spaces – which dominates the total exposure –, remains constant. This effect is less important in a dwelling with natural ventilation. Additionally, the mechanical ventilation ceases to be the dominant driving force for air flow, reducing the spread in performance with natural ventilation, in a leaky dwelling. Therefore the advantage of improved air tightness is higher for mechanical supply and exhaust ventilation.

The results presented above confirm the call for the adoption of airtight construction, supplemented with the implementation of a suitable dedicated ventilation system as a more efficient strategy to achieve high indoor air quality, that has been promoted by the Air Infiltration and Ventilation Center (AIVC), a permanent annex of the IEA/ECBCS program, since its foundation, summing it up in their 'Build Tight, Ventilate Right' baseline.

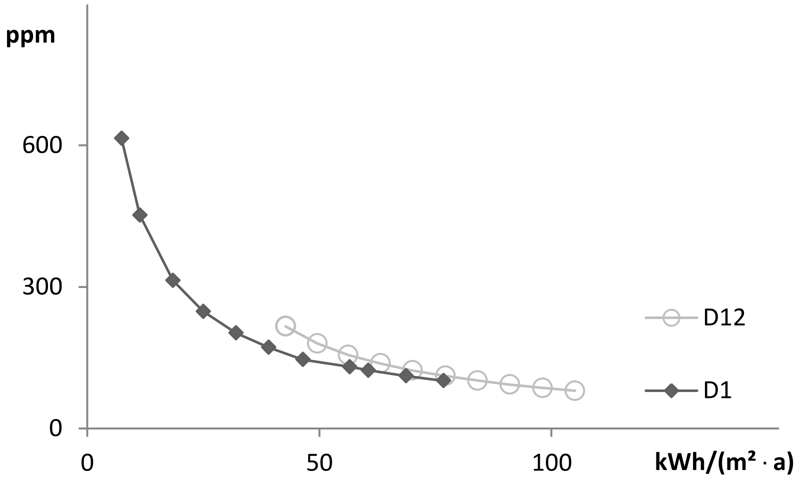

Figure 3.6 Total ventilation heat loss and average exposure to carbon dioxide of the pareto optimal configurations for mechanical (D) ventilation systems in the airtight (D1) and leaky (D12) detached dwelling

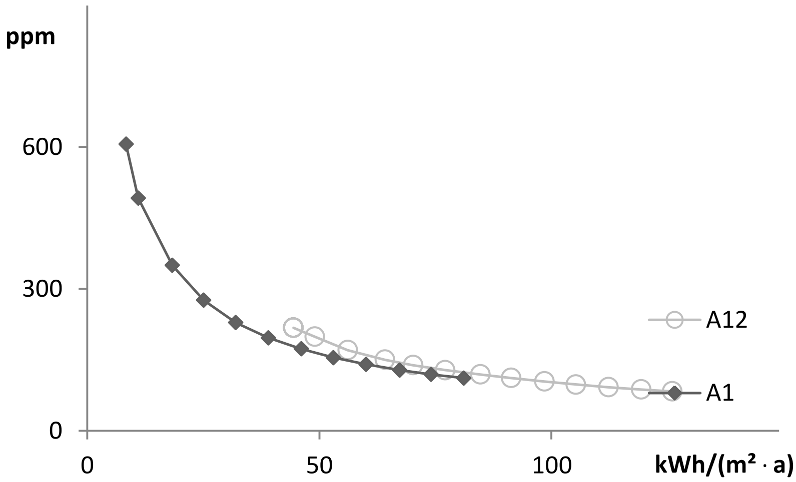

Figure 3.7 Total ventilation heat loss and average exposure to carbon dioxide of the pareto optimal configurations for natural (A) ventilation systems in the airtight (A1) and leaky (A12) detached dwelling

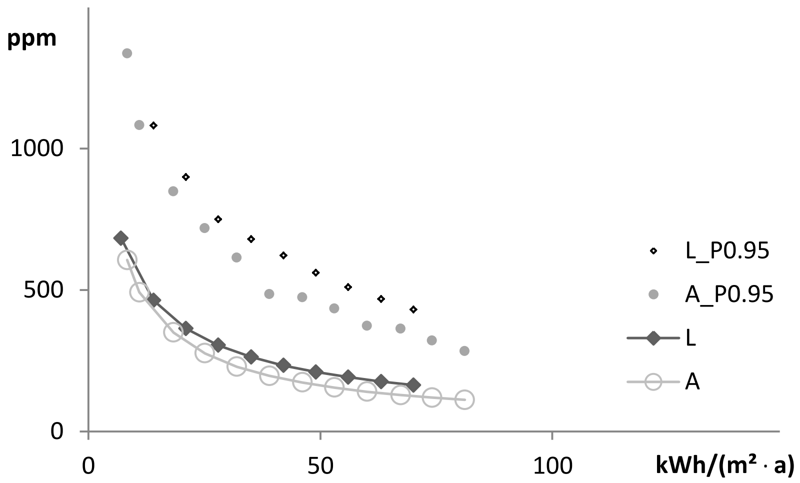

Figure 3.8 Average and peak ($P_{0.95}$) excess carbon dioxide exposure during occupancy for the cases with uniform envelope leakage as the exclusive means of ventilation (L) and the Pareto optimal sizing options with dedicated natural ventilation (A) systems

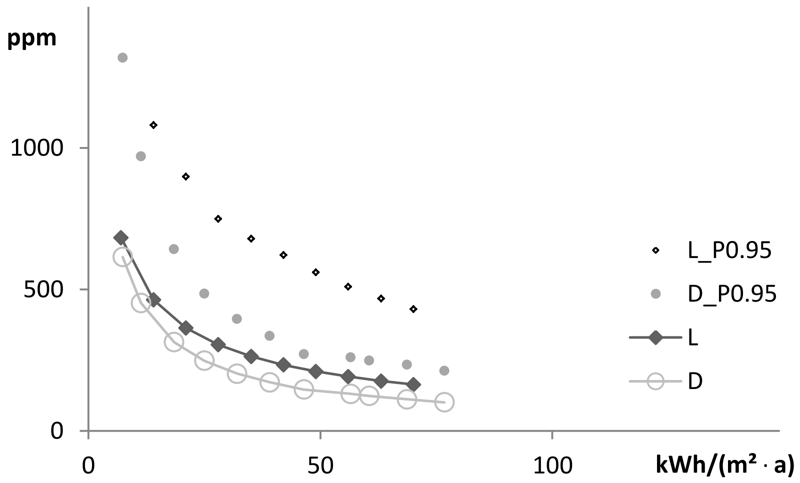

Figure 3.9 Average and peak ($P_{0.95}$) excess carbon dioxide exposure during occupancy for the cases with uniform envelope leakage as the exclusive means of ventilation (L) and the Pareto optimal sizing options with dedicated mechanical supply and exhaust (D) systems

Air infiltration through building leakage, however, is the most common form of ventilation in the existing building stock. The concept of airtight construction also meets a lot of resistance with home-owners, loathing the double investment in airtight construction and ventilation system it requires, while leakage is perceived as 'just as good'. Weatherization is even often not executed or only partly executed for fear of lowering building leakage too far, thereby reducing ventilation rates and deteriorating indoor air quality, and the installation of dedicated ventilation systems in building refurbishments is scarce.

Figure 3.8 and Figure 3.9 show the average exposure of the occupants to carbon dioxide during occupancy for the cases with only uniformly distributed envelope leakage as a means of ventilation and for the cases with a natural and mechanical supply and exhaust system, respectively, as well as the 95th percentile of these exposures. The latter is preferred over the dose over a threshold as metric for peak exposure here, because it is expressed in the same unit as the average exposure. For the cases with a dedicated ventilation system, the results for an envelope leakage of 1 $m^3/(h·m^2)$ are shown. Additionally, for these cases, only the Pareto optimal sizing options, considering average exposure and ventilation heat loss, are shown. From these results, it is clear that dedicated ventilation systems achieve better indoor air quality at lower ventilation heat losses, with the mechanical supply and exhaust system achieving the best performance.

The performance of the different ventilation approaches converges at low flow rates (low ventilation heat loss). However, exposure to concentrations in excess of 1000 ppm excess CO_2 is considered to correspond to poor perceived indoor air quality and is therefore classified as the lowest class of indoor air (IDA) quality, IDA 4 in European ventilation standards [230]. On the other hand, the indoor air quality results show little improvement with increasing flow rates above 60 $kWh/(m^2·a)$. Additionally, systems in this range on average achieve excellent indoor air quality (IDA 1 < 400 ppm) and even peak exposures are well within acceptable indoor air quality limits (IDA 2 < 600 ppm). Therefore, the analysis of the results can be limited to the midrange of sizing options between 20 and 60 $kWh/(m^2·a)$. Within this range, the average exposure of the occupants to carbon dioxide in the envelope leakage cases is 23 % higher than that in the cases with a natural ventilation system and 40% higher than that in the cases with a mechanical supply and exhaust system at equal ventilation heat loss.

Considering peak exposure, represented by the 95th percentile ($P_{0,95}$) to which the occupants are exposed during occupancy in the graph, this increases to 29 and 88 %, respectively. Comparing the results at equal indoor air quality, the respective differences are 32 and 51% for average exposure and 52 and 106 % for peak exposure. Especially the substantially lower peak exposure achieved by the mechanical supply and exhaust system is noteworthy.

3.3.3 Sizing Options

Although the achievable performance of all 3 ventilation system types was found to be very similar when considering average exposure to carbon dioxide, the impact of suboptimal sizing on the performance of the system is very different for each system type. The natural ventilation system is more sensitive to inadequate sizing than mechanical supply and exhaust ventilation, as is clear from the much larger spread in results of the natural ventilation cases in Figure 3.10, while intermediate sensitivity is observed for mechanical exhaust ventilation. Obviously, the cases with smaller design flow rates of the components are associated with lower ventilation heat loss and higher exposure, while those with large design flow rates of - for the mechanical supply and exhaust and mechanical exhaust systems at least one type of mechanical - components achieve better indoor air quality at a larger ventilation heat loss.

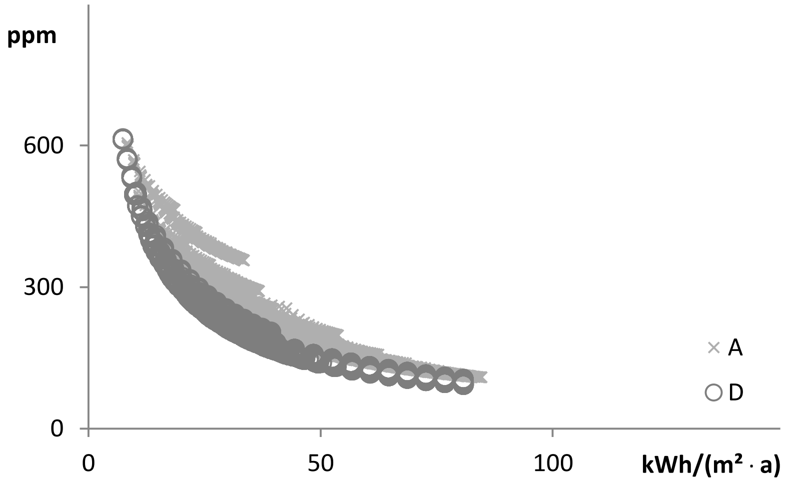

Figure 3.10 Total ventilation heat loss and average exposure to carbon dioxide of all configurations for natural (A) and mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling

Natural Ventilation

Figure 3.11 and Figure 3.12 depict the performance of the Pareto optimal cases in the flat with natural, mechanical exhaust and mechanical supply and exhaust ventilation using the average carbon dioxide exposure as criterion for indoor air quality. In the midrange, a marked increase in exposure compared to the optimal cases for mechanical supply and exhaust with equal heat loss is seen for the mechanical exhaust and natural ventilation cases. The performance of these systems is mainly limited by the size of the transfer grilles. If the transfer grilles are sized in accordance to the design supply or exhaust flow rate of the space instead of to a fixed value, the optimal performance of the natural and mechanical exhaust ventilation improves and becomes virtually equal to that of the mechanical supply and exhaust ventilation system. The optimal performance of the latter is not affected by the change in sizing of the transfer grilles. The more marked nature of this effect compared to the detached dwelling is caused by the specific aspects of the flat. The higher stack decreases the variability of the flow rate in the cases with natural ventilation, reducing the difference in optimal performance of this system with that of mechanical exhaust ventilation.

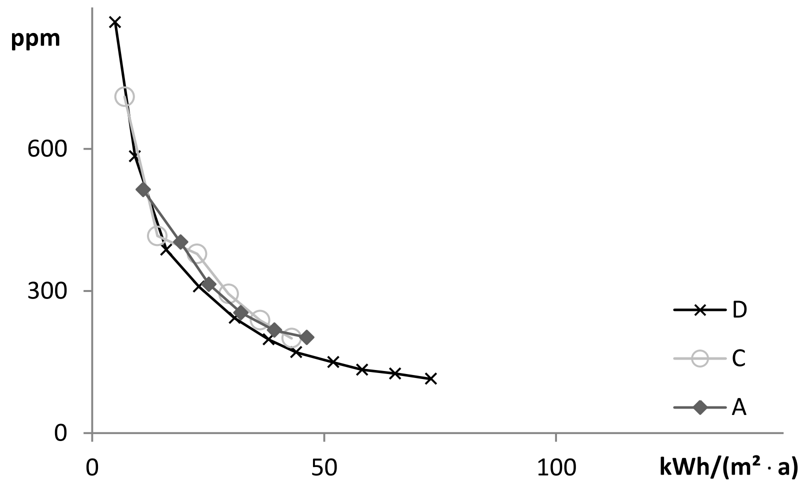

Figure 3.11 Total ventilation heat loss and average exposure to carbon dioxide of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the flat

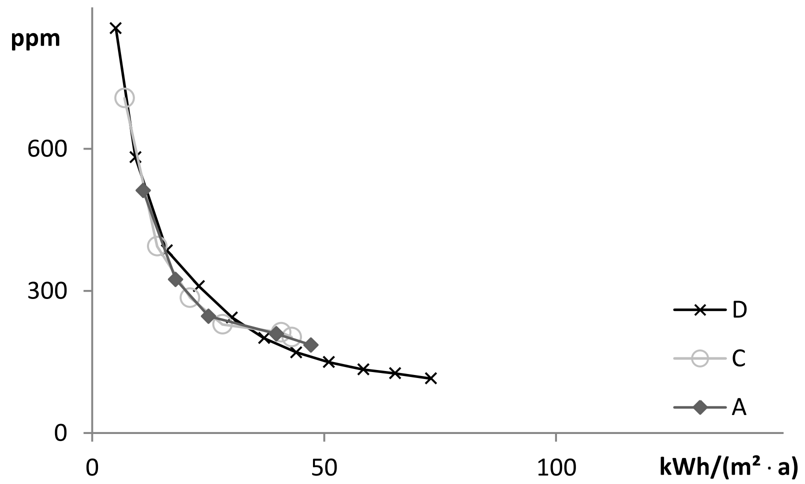

Figure 3.12 Total ventilation heat loss and average exposure to carbon dioxide of the pareto optimal configurations for natural (A), mechanical exhaust (C) and mechanical supply and exhaust (D) ventilation systems in the flat with adapted transfer device sizing

The bulk of the exposure is situated in the living spaces. Due to the limited building envelope surface, infiltration and adventitious ventilation are limited and the fresh air flow rate in these spaces is mainly governed by the exhaust in the wet spaces and the transfer grilles. The size of the transfer devices has a similar effect on the performance as the overall airtightness of the dwelling. The fraction of exfiltrating supply air and infiltrated exhaust air will decrease along with a decreasing ratio of the flow resistance of the transfer devices and that of the building envelope. From this observation follows that decreasing the internal flow resistance by increasing the size of the transfer grilles or choosing a more open plan layout increases the performance of natural and mechanical exhaust ventilation, especially in flats. This may, however, be limited by practical issues. The popular method to create transfer by a simple slit under or above the internal doors is aesthetically unacceptable for higher design flow rates. Large transfer grilles also disrupt acoustic privacy.

Mechanical Exhaust Ventilation

In literature [399], the sizing rules introduced by the French ventilation standard were found to provide Pareto optimal performance for mechanical exhaust ventilation when compared to 4 other mechanical exhaust ventilation standards. What sets the French standard apart from the other standards that were assessed, are its small trickle ventilators, combined with moderately high exhaust flow rates and relatively large transfer devices.

The effectiveness of this design approach is obvious in Figure 3.13. It shows, for the airtight detached dwelling, the performance of all mechanical exhaust ventilation cases as well as those cases where the design flow rate, relative to that of the Belgian standard, for the trickle ventilators is smaller than that for the transfer devices and for the exhaust flow rate. 90% of the Pareto optimal solutions satisfied these criteria.

Mechanical Supply and Exhaust Ventilation

The performance of the mechanical supply and exhaust ventilation system is best when slightly higher flow rates are selected for supply than for exhaust in airtight buildings. Figure 3.14 shows both all mechanical supply and exhaust ventilation cases and those that have a supply air flow that is between 1 and 3 times the exhaust flow rate for the air tight detached dwelling. 80% of the Pareto optimal cases are in this group. The good performance of these cases, with an odds ratio of Pareto optimality of 13 versus the other cases, is explained by the dominance of the exposure in the living spaces in the total exposure to carbon dioxide.

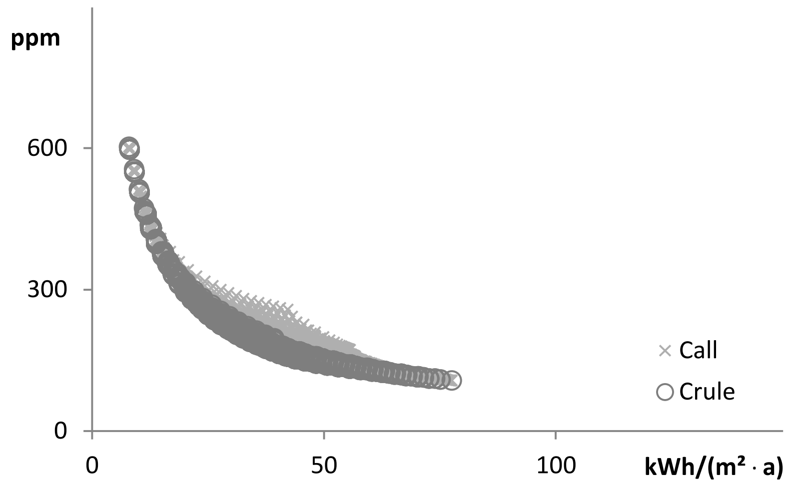

Figure 3.13 Total ventilation heat loss and average exposure to carbon dioxide of all configurations for mechanical exhaust (C) ventilation systems in the airtight detached dwelling (x) and those where the trickle ventilators are smaller than the transfer devices and the exhaust flow rate (O)

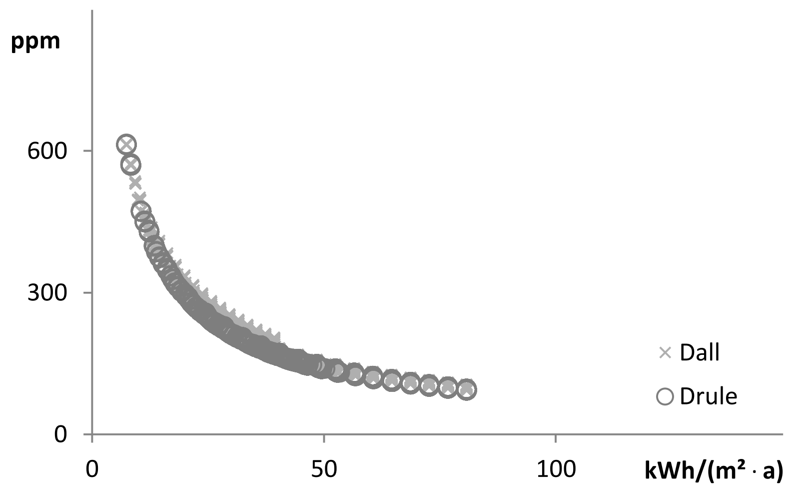

Figure 3.14 Total ventilation heat loss and average exposure to carbon dioxide of all configurations for mechanical supply and exhaust (D) ventilation systems in the airtight detached dwelling (x) and those where the supply flow rate is equal to 1-3 times the exhaust flow rate.

3.4 Conclusions

In this chapter, both the differences in optimal performance of the natural ventilation, the mechanical exhaust ventilation and the mechanical supply and exhaust concept and the opportunity to frame optimal general sizing rules for these concepts were assessed with numerical simulations. Optimal performance was defined as the best achievable indoor air quality for a given ventilations heat loss or vice versa. Exposure to carbon dioxide, to humidity and to odour were used as indoor air quality assessment parameters. Heat loss through mechanical flow rate, adventitious ventilation and infiltration were considered part of the total ventilation heat loss.

Considering average exposure to carbon dioxide, only slightly better performance of the optimized mechanical supply and exhaust concept compared to mechanical exhaust ventilation was observed, while the latter demonstrated slightly better performance than natural ventilation. The spread in optimal performance increased when exposure to peak concentrations was considered instead of average exposure. Nevertheless, the differences remained moderate.

A more substantial spread in optimal performance of natural ventilation cases with optimal mechanical exhaust and mechanical supply and exhaust cases was found considering exposure to humidity and odours, due to the more frequent occurrence of backdraft from the service spaces to living spaces.

With respect to the quality of the building envelope, all system concepts showed superior optimal performance in airtight than in leaky conditions. Additionally, dedicated ventilation systems achieved 30-40% lower exposure at equal ventilation heat loss than uniformly distributed leakage as a means of ventilation.

For mechanical exhaust ventilation cases, smaller trickle ventilators are more likely to provide optimal performance, while in mechanical supply and exhaust cases, supply flow rates slightly larger than exhaust flow rates prove to provide the best chance for optimal performance. In all system concepts, larger transfer devices increased the performance, although this increase was more marked for natural and mechanical exhaust ventilation systems.

These basic sizing rules should be used for the development of future design guidelines and standards. More specifically, one general recommendation is the specification of a required maximum leakage level of the building envelope, especially for mechanical supply and exhaust ventilation. Another is specifying design flow rates for trickle ventilators at pressure differences higher than 2 Pa. Transfer devices should be specified at low pressure differences (< 2 Pa) and at the design flow rates of the spaces they connect.

4

Achievable Performance: Demand Control

As was mentioned in Chapter 1, roughly two approaches exist to further reduce ventilation heat loss in well-designed systems: demand controlled operation and heat recovery technology. In this chapter, the energy saving potential of demand controlled ventilation is investigated. Starting from the 'state of the art' with an assessment of the performance of 4 control approaches similar to systems available on the Belgian market (results published in Building and environment [347]), the analysis expands to the impact of the local context created by ventilation standards or climate, by implementing 2 control strategies on the mechanical exhaust ventilation systems presented in the last section of chapter 2, and to Pareto optimal performance. The latter is up to 60% better than that of continuous flow systems. In between, the Belgian methodology for the assessment of demand controlled ventilation systems within the framework of the Energy Performance of Buildings Directive is discussed. Finally, results from case study measurements on demand controlled systems in Belgium are presented.

4.1 Demand controlled: types and performance

Two main strategies have been developed to reconcile these seemingly opposing interests: heat recovery and demand controlled ventilation. In the moderate climate zone of Western Europe, especially in the Netherlands, France, the UK and Belgium, with about 2500-3000 heating degree days [404, 405], the payback time for investments in heat recovery ventilation is long, especially in buildings with relatively low air change rates such as dwellings. Due to its competitive price setting as well as due to reports in popular media and scientific literature about possible health risks associated with heat recovery systems [406] simple mechanical exhaust ventilation dominates the residential ventilation market in this region [317, 361]. In light of this exhaust ventilation tradition, home owners tend to prefer demand controlled mechanical exhaust ventilation over heat recovery systems to comply with tightening energy performance legislations. However, little information is available in literature on the performance that can be achieved with different approaches to demand controlled exhaust ventilation.

This section focuses on the energy saving potential of demand controlled mechanical exhaust ventilation in residences and on the influence such systems may have on the indoor air quality to which the occupants of the dwellings are exposed. The conclusions are based on simulations done with a multi-zone airflow model of a detached house that is statistically representative for the average Belgian dwelling. Four approaches to demand based control are tested and reported. In this chapter exposure to carbon dioxide and to a tracer gas are used as indicators for indoor air quality. Both energy demand and exposures are reported and compared to the results for a standard, building code compliant, mechanical exhaust system, operating at continuous flow rates.

Residential ventilation systems are usually bought as a complete package with a set of standard components and are therefore far less tailor made than large HVAC systems. In a competitive market, reliable performance assessment and evaluation of these ventilation systems is essential, although the operating conditions (building geometry, wind conditions) can vary largely between dwellings and occupancy is susceptible to change over the lifetime of an installation. Therefore, the sensitivity of the control strategies to environmental and user variations is tested using Monte-Carlo techniques. Under the conditions that were applied, reductions on the ventilation heat loss of 25 to 60% are found, depending on the chosen control strategy (with the exclusion of adventitious ventilation and infiltration). The presented performance and sensitivity results can be used to understand and design appropriate residential demand controlled mechanical exhaust ventilation.

4.1.1 Demand Controlled Ventilation in Literature

Available literature on demand controlled systems is mainly focussed on two aspects: single-zone, large air change rate situations [335, 407-409] on the one hand and on optimal set point or control algorithm development [346, 410] on the other. Although few papers focus on the residential context [411], time use reports indicating that 70 % of our time is spent at home and 50% of that time is spent alone [357] clearly show the potential for demand control in dwellings.

In contrast to the dedicated air handling systems for large spaces such as open plan offices, conference halls and theatres, fresh air supply and exhaust in residential ventilation are usually decoupled in space. Fresh air is introduced in the living room and bedrooms whereas polluted air is extracted from the dwelling in 'wet' spaces such as kitchens, toilets and bathrooms. Transfer devices in doors allow air to flow from the dry spaces to the wet spaces through hallways and staircases. This particular configuration requires a performance assessment on a multi-zone (system) level in order to account for inter zone interaction.
In addition to that, the rating of the indoor environment is a complex, multi-layered problem [227]. The long list of indoor air quality performance indicators for residential ventilation systems proposed in the EN 15665 standard [229] clearly demonstrates that no consensus exists on how to rate ventilation system performance. Nonetheless, the choice of a specific performance criterion has a large influence on assessment results [412].

4.1.2 Methods

Four different demand control strategies were implemented on a basic mechanical exhaust ventilation system sized in accordance with the Belgian standard. The general simulation methodology presented in chapter 2, including the Monte Carlo sensitivity analysis, is used to assess the performance of these 4 demand control strategies for the detached reference house (also see chapter 2) All of the strategies reduce the flow rates when ventilation need is limited in terms of perceived indoor air quality, relative humidity or presence of occupants. Three of those strategies interact with a single system component (trickle ventilator, vent hole or fan), whereas the fourth interacts with these system components simultaneously. All strategies are abstractions of commercially available systems. Table 4.1 lists a summary of all strategies.

Table 4.1 Selected demand control strategies

Strategy	component	set point	band
1. relative humidity	vent hole	70 %	5 pp
2. occupant presence	fan	-	20 min
3. CO_2	trickle	1000 ppm	-
4. all	vh / f / t	1000 ppm	20 min

The first control strategy interacts with a valve in the vent hole of each 'wet' room (kitchen, toilet, service room and bathroom) and is based on the relative humidity measured in the extracted air. A minimal opening area of 10 % of the design opening area for each vent hole is maintained at all times. The opening area of the vent hole is increased to the design opening area if the measured relative humidity is higher than 70 % and is reduced to the minimal flow rate again when it drops below 65 %. Note that the fan is not directly affected by this control strategy. The 70 % set point is chosen because it is a marker for elevated mould risk on typical thermal bridges [107, 111]. An effective moisture penetration depth (EMPD) model [413] is used to simulate moisture buffering in the spaces. The second strategy interacts only on the exhaust fan and is triggered by presence in either bathroom, toilet or kitchen. With the detection of presence in any of these rooms, the exhaust flow rate is increased to the total design flow rate for exhaust again. The total exhaust flow rate is reduced to 10 % of the design flow rate after 20 minutes of absence in all of these rooms.

The third strategy interacts with the trickle ventilators (supply) and reduces their opening size according to the CO_2 concentration in the room where the trickle ventilator is situated. If the CO_2 concentration is below the set point of 1000 ppm, the opening area is reduced to 10 % of the original size. The 1000 ppm setpoint is a popular value in demand controlled systems on the market [348]. It also corresponds quite well with the concentration which can be expected when an airflow rate of 36 m^3/h of fresh air is provided for every occupant in a room. This corresponds to the upper limit of the IDA 2 in EN 13779 [230], which is the basis of the design flow rates imposed in the Belgian standard. In this case, extraction flow rates are constant.

The last strategy interacts with all of the components manipulated in the first 3 strategies. The trickle ventilators are manipulated according to the CO_2 concentration in the same way as is used in the third strategy. The vent holes and the exhaust fan are manipulated according to occupant presence like in the second strategy.

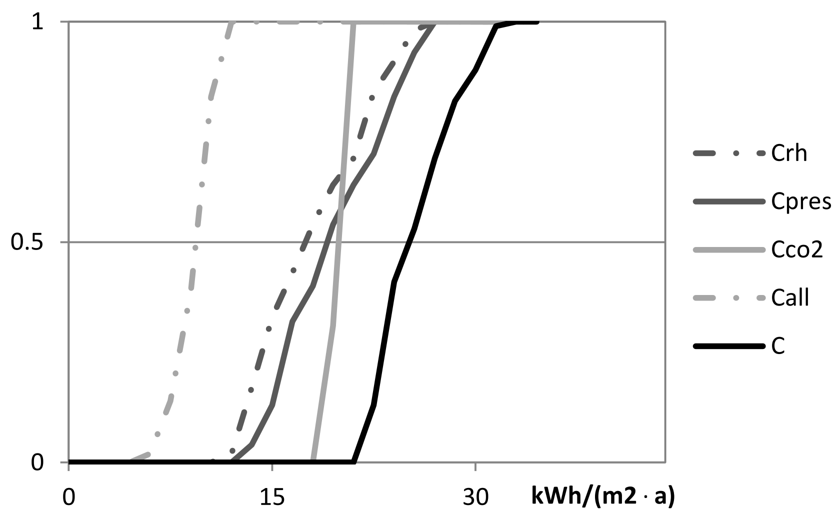

Figure 4.1 Distribution of the mean heat loss ($P_B\{E(Q_{v,tot})<x|B\}$) associated with fresh air supply to the dwelling of the 100 cases selected in the Monte Carlo analysis for the base case (mechanical exhaust ventilation) and the 4 selected demand control strategies

4.1.3 Results

As was detailed in section 2.2.4, 100 simulations were carried out for each of the strategies that were discussed in section 4.1.2. In this section, the results of these simulations will be presented.

All simulations were run over the heating season only, in this case between September 28th and April 15th, using the Test Reference Year for Ukkel climate data [414]. In section 2.2.3, three assessment parameters were proposed. Figure 4.1, Figure 4.2 and Figure 4.3 show the distribution of the results total ventilation heat loss, the perceived indoor air quality (average excess CO_2-concentration) and the extraction efficiency (intake fraction), for each of the 4 demand control strategies. Additionally, the results for the standard exhaust system are included in the figures as a reference.

Energy Saving Potential

As can be seen in Figure 4.1, the control strategies that only act on the exhaust air flow, either through manipulation of the vent hole (Crh) or by manipulation of the exhaust fan (Cpres), have about the same energy saving potential. However, in comparison with the base case with constant flow mechanical exhaust ventilation, the heat loss is slightly more sensitive to changes in the environmental parameters. The control strategy where the trickle ventilators are manipulated (Cco2) has a comparable average saving potential as the first two strategies, but in contrast to them, makes the ventilation heat loss less sensitive to the climate, which can be observed in the much steeper curve in Figure 4.1.

This can be explained by the flow dynamics in the dwelling. Two driving forces determine the flow pattern through the model: mechanically induced forced ventilation flow on the one hand, climate induced flow caused by the buoyancy effect and wind pressure [415, 416] on the other. While the mechanical flow is constant and well controlled, the climate induced flow is governed by fluctuating environmental parameters and therefore more variable. The control strategies that reduce or manipulate the exhaust flow will reduce the relative influence of the mechanical flow and their performance will therefore be more sensitive to changes in environmental parameters. By manipulating the trickle ventilators, however, the natural convection component is reduced and the under pressure created by the mechanical exhaust is increased, considerably increasing the relative influence of the forced flow and reducing the sensitivity to environmental parameters.

Since the 4th strategy reduces both the mechanical exhaust flow rate and the opening size of the trickle ventilators (Call), the heat loss results for this configuration have about the same robustness as the base case.

This effect is also visible in Figure 4.4, where the results for the energy indicator are shown for simulations with the 1st data set of the Monte Carlo algorithm for different levels of airtightness. In this dataset, all parameters are set to their average value. While all other strategies, like the base case, are linearly related to the airtightness level, the strategy that manipulates the trickle ventilators levels off to a constant performance in the more airtight range. Note that the projected intercept of the curves with the Y-axis in Figure 4.4 defines the ventilation heat loss in absence of infiltration and adventitious ventilation.

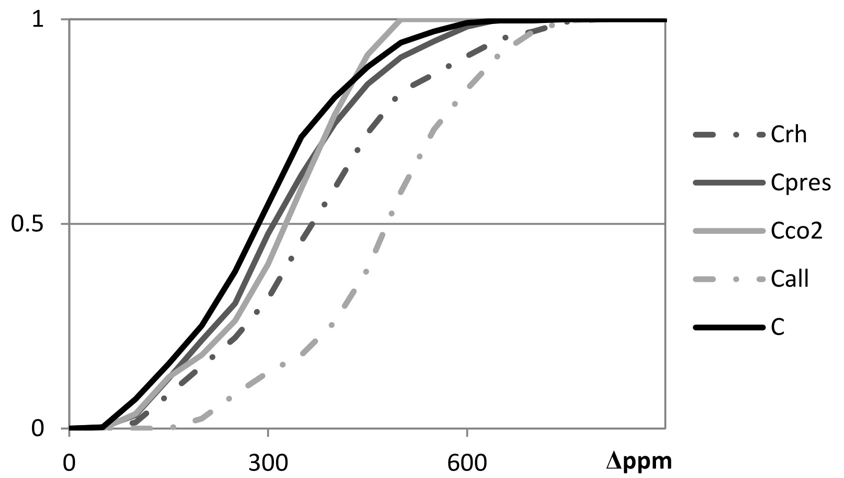

Figure 4.2 Distribution of the mean exposure ($P_O\{E(C_{exp}$-$C_{amb})<x|O\}$) to excess CO_2 concentration (ppm above the outdoor concentration: Δppm) of the 334 occupants in the 100 cases selected in the Monte Carlo analysis for the base case (mechanical exhaust ventilation) and the 4 selected demand control strategies

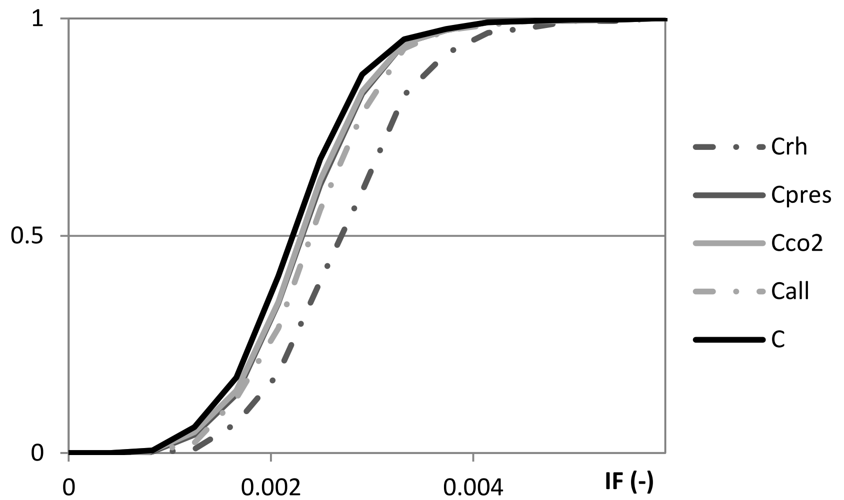

Figure 4.3 Distribution of the individual intake fraction for the odour tracer gas of the 334 occupants in the 100 cases selected in the Monte Carlo analysis for the base case (mechanical exhaust ventilation) and the 4 selected demand control strategies

Indoor Air Quality

From Figure 4.2. we can conclude that both the control strategy based on presence detection and the one based on CO_2 detection render a perceived indoor air quality level that is comparable to that of the original constant flow system. The CO_2 based strategy is slightly more robust, while the control strategy based on relative humidity has a negative impact on the perceived indoor air quality level and increases the sensitivity to climate conditions. This can be explained by the fact that relative humidity is influenced by a lot of factors, such as hygroscopic buffering, ventilation rate, and outdoor climate, whereas the performance criterion, exposure to CO_2, is only function of occupancy.

The control strategy with simultaneous manipulation of the trickle ventilators, vent holes and exhaust fan considerably increases the mean exposure to CO_2 compared to the base case. When, however, instead of the distribution of the mean exposure (as in Figure 4.2), the distribution of all instantaneous concentrations to which the occupants are exposed in the simulations are plotted, like in Figure 4.5, we can see that the increased mean exposure is mainly due to an increase in exposure to concentrations below 600 ppm above the outdoor concentration, while the increase in exposure to higher concentrations is negligible. Since the EN 13779 standard [230] considers concentrations below 600 ppm above outdoor concentration to indicate good indoor air quality (IDA II), the increased exposure in the 4^{th} control strategy is still acceptable. Note that the CO_2-based strategy, with constant exhaust rates, drastically reduces exposure to high carbon dioxide concentrations.

The results for extraction efficiency of the tracer gas in the wet spaces (Figure 4.3) are similar to those for perceived indoor air quality. Again, the control strategy based on relative humidity results in a less efficient system because of the intricate relation between elevated relative humidity and ventilation demand. The combined control strategy now produces results in the same range as the constant flow base case and the presence and CO_2 based control strategies.

Table 4.2. shows the results for the indicators based on a single simulation where all parameters from the Monte Carlo algorithm are set to their average value. These results are compared to the median value of the Monte Carlo results. Although a lot of information is lost when only one simulation is done, the results prove to be a good estimate for the relative performance of the control strategy on all indicators. For the energy and perceived indoor air quality indicators, they even coincide well with the average results of the Monte Carlo analysis.

4.1.4 Summary

In this section, the energy saving potential and the repercussions on the indoor air quality of 4 different demand control strategies for residential ventilation were investigated. Two performance indicators were proposed to assess the indoor air quality. The first deals with perceived indoor air quality and the second dealing with the efficiency of extraction of specific pollution from ‘wet’ areas in the dwelling.

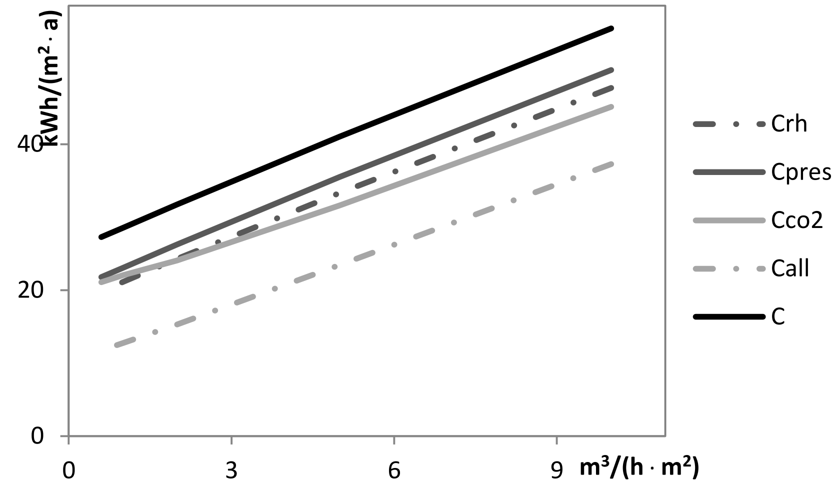

Figure 4.4 Mean total heat loss associated with fresh air supply to the dwelling at different airtightness levels (v50) for the base case (mechanical exhaust ventilation) and the 4 selected demand control strategies

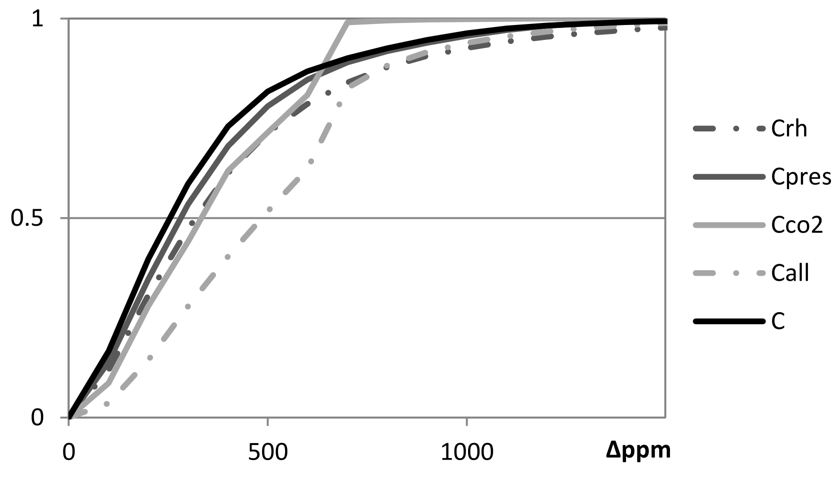

Figure 4.5 Distribution of exposure of all the occupants ($E_O(P\{C_{exp}-C_{amb}<x|O\})$ to CO_2 concentrations above outdoor concentration (Δppm) for the base case (mechanical exhaust ventilation) and the 4 selected demand control strategies

Monte Carlo analysis was used to assess the sensitivity of the control strategies to changes in occupancy, environmental boundary conditions and dimensional parameters. The 3 demand control strategies that only manipulate 1 system component rendered an energy saving potential of about 25 %, whereas the strategy with combined manipulation of trickle ventilators, vent and exhaust fan had an energy saving potential of 60 %. Strategies that manipulate trickle ventilators proved to be more robust, with performances that are less sensitive to variable conditions such as climate and occupancy.

The control strategy based on relative humidity was least suited to maintain the indoor air quality at the level of the base case with constant exhaust flow, although the indoor air quality level was within the same range for all of the proposed strategies. At first glance, for the combined strategy the perceived indoor air quality seemed significantly worse than in the base case. Further analysis, however, revealed that the exposure to elevated carbon dioxide levels was comparable to that of the other strategies. The strategy that manipulates the trickle ventilators proved to drastically reduce peak exposure to metabolism related pollutants.

The results obtained with a single simulation with average input values proved to be a good indicator for the relative performance of a control strategy although they do not provide any information on its robustness. They can therefore be useful in optimization exercises, but in an overall assessment of a new control strategy, a sensitivity analysis should be included.

Table 4.2 single simulation results (a) - median of Monte-Carlo results (b)

Strategy	Energy (kWh)		CO_2 (ppm)		Tracer IF (-)	
	(a)	(b)	(a)	(b)	(a)	(b)
0. Standard system	27.30	26.37	299	292	$4.2 \cdot 10^{-3}$	$2.6 \cdot 10^{-3}$
1. relative humidity	19.83	18.60	378	364	$4.5 \cdot 10^{-3}$	$2.8 \cdot 10^{-3}$
2. presence	21.15	21.18	355	379	$4.4 \cdot 10^{-3}$	$2.7 \cdot 10^{-3}$
3. CO_2	21.78	20.25	331	316	$4.4 \cdot 10^{-3}$	$2.7 \cdot 10^{-3}$
4. all	11.73	10.84	486	489	$4.7 \cdot 10^{-3}$	$3.1 \cdot 10^{-3}$

4.2 Assessment Method in Belgium

4.2.1 Belgian Context

In the competence structure of the different state levels in the federal state Belgium, the implementation of the EPB-directive is a regional competence. The Belgian residential ventilation requirements are set forward in the Belgian Standard NBN D 50-001, which is annexed to the EPB-decrees of the different regions. This standard dates back from 1991 and in it, the ventilation systems are presented in a descriptive manner. Four standard systems are described, ranging from natural ventilation (system A), over mechanical exhaust (system C) or supply ventilation (system B) to fully mechanical ventilation (system D). In the market, systems A, C and D are dominant, while system B is virtually inexistent.

The standard requires the air supply and return components of the systems to be sized according to the function of the room in which they are located. The specifics of these sizing rules have been discussed in Chapter 2.

Since the standard only mentions demand control as a possible extension of the reference systems without any detail as to how this demand control should be achieved, an equivalence approach is used to rate the performance of demand controlled ventilation systems. This assessment is done by UBAtc, a technical approval agency in Belgium and is valid for all the regions. In order to be deemed equivalent to the reference systems in the standard and therefore acceptable under the building code, the performance of the demand controlled system cannot be inferior to the worst performance obtained by the application of each of the reference systems.

4.2.2 Assessment Methodology

A demand controlled residential ventilation system is assessed through numerical simulations with the multi-zone airflow model Contam, following the methods proposed in chapter 2, for the detached building geometry presented there. Both the system under review and the 3 reference systems that have a reasonable market share are assessed.

The performance of the demand controlled system is assessed on 3 IAQ parameters, namely humidity levels, exposure to odours (tracer) and perceived indoor air quality (carbon dioxide). If performance of the system under review is equal to or better than that of the worst performing reference system for each of these parameters, it is accepted as equivalent and an energy saving coefficient is determined.

Since heat loss and exposure reductions are opposing interests, they have to be traded off against each other [234, 388]. Several authors have proposed using weighted sums of these different criteria [389, 390]. The definition of these weighting coefficients, however, lacks a robust scientific basis. Instead of weighting both criteria, demand controlled systems should achieve better results on both indoor air quality and heat loss. By interpolation between their respective indoor air quality and heat loss performance, the reference systems (natural ventilation, mechanical exhaust ventilation and mechanical supply and exhaust) define the minimum performance level. The level of indoor air quality that is attained by the application of the demand controlled system is taken into account to determine its performance coefficient. The coefficient is defined as the ratio of the heating season integrated ventilation heat loss with the exclusion of infiltration losses and a reference. This reference is the heat loss of the interpolated minimal performance limit at the same indoor air quality level. The process for the determination of the reference and the performance coefficient for a system ‘X’ demonstrated in Figure 4.6.

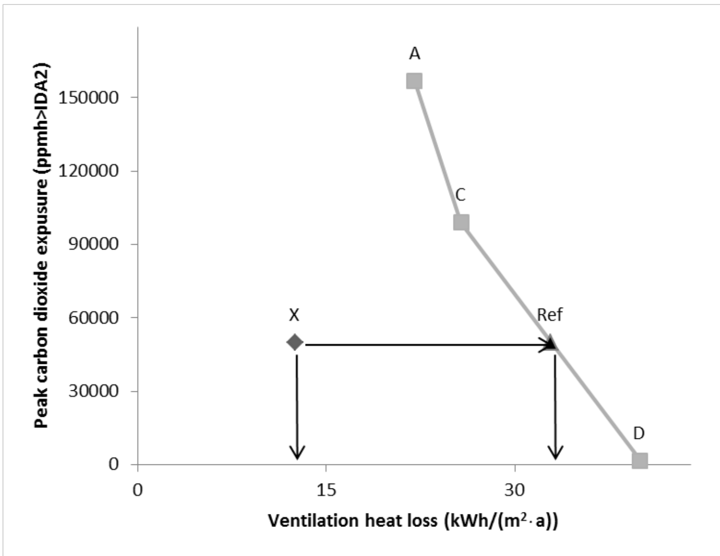

Figure 4.6 Reference and energy saving coefficient calculation for a ventilation system X

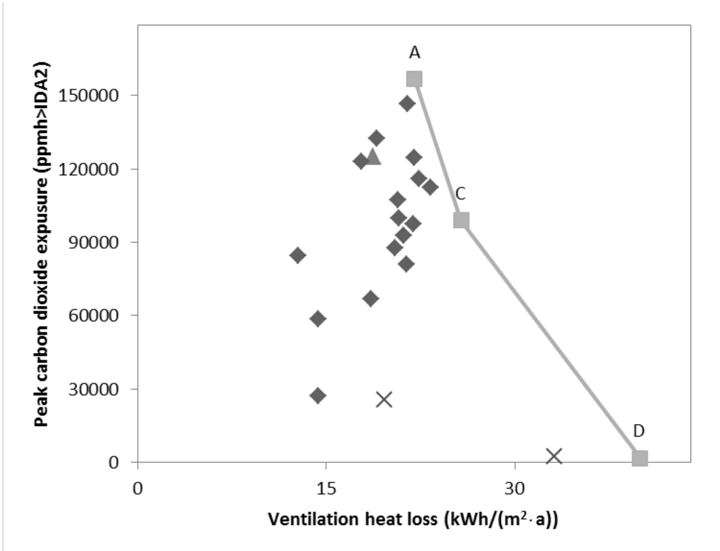

Figure 4.7 Performance of certified natural (Δ), mechanical exhaust (◊) and mechanical supply and exhaust (X) demand controlled residential ventilation systems on the Belgian market

The infiltration heat losses are treated separately in the EPB-calculation method. Nevertheless, should the building leakage to infiltration heat loss ratio obtained under operation of the demand controlled system be different from that of the reference systems, an additional correction coefficient for this is calculated. So far, however, no system reviewed presented such behaviour.

4.2.3 Application to Systems on the Market

Of the systems that are available on the market in Belgium, 20 have filed a request to be assessed with this methodology and have received certification. They include 17 mechanical exhaust ventilation systems, 2 mechanical supply and exhaust ventilation systems and 1 natural ventilation system. The results are shown in Figure 4.7. Although mechanical supply and exhaust systems render the best indoor air quality, the best mechanical exhaust ventilation systems perform very well, achieving an energy saving coefficient of 0.4, while in a number of more simplified demand control approaches for exhaust systems, the reduction in ventilation heat loss is mainly achieved by reducing the indoor air quality, resulting in an energy saving coefficient of up to 0.95. The best demand controlled mechanical exhaust systems either have supplementary exhaust vents in the main spaces of the dwelling or actively control the trickle ventilators.

4.3 Sizing and climate

In this section, the simulation based analysis of the performance of demand controlled mechanical exhaust ventilation systems presented in the first section of the chapter is expanded to a broader range of boundary conditions, using the Belgian trade-off methodology presented above and two of the systems assessed with this methodology. The 2 selected strategies are the first and the best rated system on the Belgian market respectively. Two aspects are studied: the change in assessed performance under different sizing of the reference systems and the impact of climate. This is done by implementing the 2 selected demand control strategies in the models for continuous flow mechanical exhaust ventilation from different Western European countries presented in section 2.5 and using the results presented in that section as reference levels to obtain the energy saving coefficient.

4.3.1 Description of Demand Control Strategies

The first control strategy interacts with an economiser in the vent of each 'wet' room (kitchen, toilet, service room and bathroom) and is based on the relative humidity measured in the extracted air. A minimal flow rate of 15 % of the design flow rate for each vent hole is maintained at all times. The flow through the vent hole is increased to the design flow rate if the measured relative humidity is higher than 80 % or if an increase in relative humidity of more than 2 % is observed over 5 minutes. The fan is electronically controlled to operate at constant pressure difference, within the range of its flow capacity. The 70 % set point is chosen because it is a marker for elevated mould risk on typical thermal bridges [107, 111]. An EMPD model [413] is used to simulate moisture buffering in the spaces.

The second strategy has exactly the same features, with additional exhaust vent holes and dampers in each of the bedrooms, spaces with trickle ventilators that normally do not dispose of a vent hole. The flow rate through these is determined by the CO_2 concentration measured in the extracted air. If the CO_2 concentration is below the set point of 450 ppm above the outdoor concentration, the opening size is reduced to 10 % of the original size. Between 450 and 550 ppm, the flow rate is linearly increased to the design flow rate of 30 m^3/h, aiming to keep the indoor air quality within the limits of IDA 2 as defined in EN 13779 [230].

4.3.2 Results

The cumulated ventilation heat loss for the heating season for the Belgian sizing is shown in Figure 4.8. Mechanical supply and exhaust ventilation, mechanical exhaust ventilation and the first demand control strategy (DC.1) have a very similar sensitivity to the envelope leakage level, while the second demand control strategy (DC.2) manages to slightly limit excessive leakage losses and natural ventilation is more sensitive to the leakage level. For all other countries and climates, virtually the same ranking of systems and qualitative sensitivities are found.

For the reference systems, the magnitude of the average flow rate is inversely correlated to the average exposure of the occupants to excess carbon dioxide. As was demonstrated in Chapter 2, this effect is amplified by considering peak exposures. This can be seen by comparing Figure 4.9 to Figure 4.8. The same parameter as proposed in Chapter 2, the dose of exposure to carbon dioxide concentrations exceeding 1000 ppm above the background concentration, is used here. The order of the flow rates of the reference systems in Germany is the same as in Belgium. The demand controlled systems achieve better indoor air quality than the natural ventilation system at lower flow rates. They therefore effectively address the heat loss – exposure trade off associated with ventilation. The very simple first demand control strategy fails in that respect, since the energy gain causes a significant increase in exposure. Due to its more advanced control strategy, the second demand control option is very effective at limiting peak exposure to carbon dioxide, with results close to those achieved by mechanical supply and exhaust for all cases.

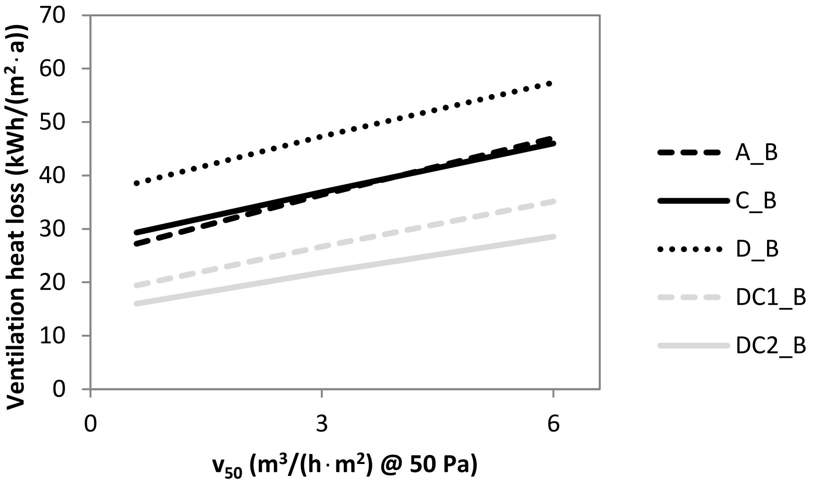

Figure 4.8 Ventilation heat loss for the 3 reference systems and 2 demand controlled systems in Brussels under different leakage levels

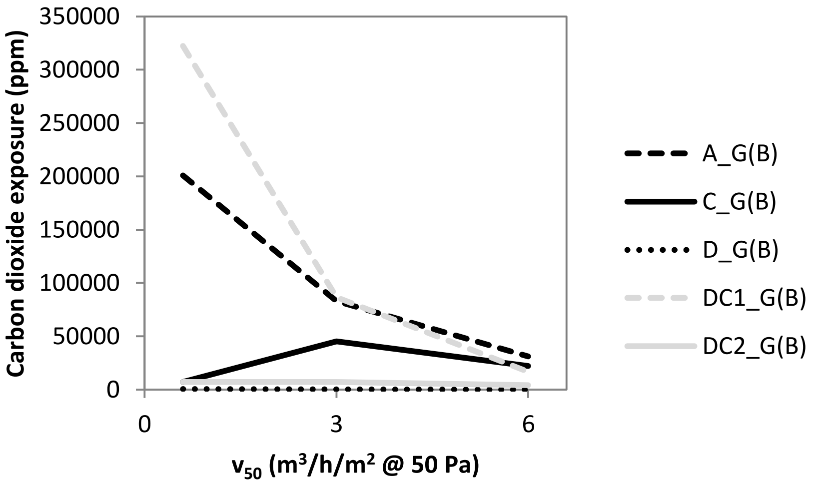

Figure 4.9 Peak exposure to excess carbon dioxide (dose > 1000 ppm) during occupancy for the 3 reference systems and 2 demand controlled systems in Berlin under different leakage levels

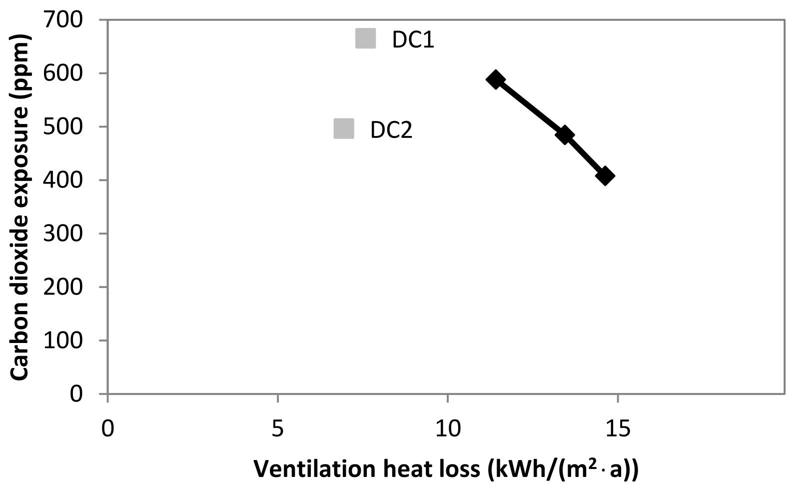

Figure 4.10 Average excess carbon dioxide exposure of the occupants vs. ventilation heat loss for the 3 reference systems and 2 demand controlled systems in Paris for an envelope leakage of 1 $m^3/h/m^2$

The energy saving coefficient of both demand control options was calculated according to the Belgian methodology explained above for each of the cases, for 3 $m^3/h/m^2$ of envelope leakage and for both average and peak exposure to carbon dioxide. The trade-off between ventilation heat loss and exposure for the Paris case is shown in Figure 4.10 as an example.

Table 4.3 Energy saving coefficient for both demand control options for all cases for an envelope leakage of 3 $m^3/h/m^2$ based on average (A) and peak (P) exposure to carbon dioxide

Case	DC.1A	DC.1P	DC.2A	DC.2P
Brussels	0.73	0.60	0.59	0.46
Paris	0.68	0.68	0.57	0.52
Lyon	0.59	0.66	0.54	0.49
Berlin	0.60	0.61	0.53	0.43
Munich	0.61	0.60	0.50	0.39
Amsterdam	0.87	0.82	0.71	0.67
London	-	-	0.42	0.39
Aberdeen	-	-	0.65	0.45

All results are listed in Table 4.3. The average coefficient for the first option is 0.68 and 0.66 based on both average and peak exposure. For the second option, this is 0.56 and 0.48 respectively. For London and Aberdeen, the coefficient could not be calculated for the first demand control option due to the peculiar sizing definitions for natural ventilation in the British standard.

4.4 Achievable Performance

The results of each of the previous sections suggests that substantial energy savings can be achieved by demand controlled ventilation systems without compromising on indoor air quality, regardless of sizing and climate, although the latter two, as well as the approach taken to demand control, strongly determine the saving potential. A question that remains to be tackled is if demand controlled systems can obtain better Pareto optimal performance than continuous flow systems. This section deals with just that.

4.4.1 Methods

The performance of a wide range of system layouts with demand control, implemented in the models presented in Chapter 3, is used to investigate the Pareto optimal performance of demand controlled systems. The different layouts are composed of components from the 3 basic demand control strategies discussed in the first sections of this chapter. As in Chapter 3, each of the system configurations discussed in this section follows the same layout rationale, found in most residential ventilation standards (e.g. [269]). Fresh air is supplied directly to the occupied zones in the dwelling (living room, bedrooms …), transferred through the circulation spaces and extracted from the 'wet' zones (toilet, bathroom, kitchen …). All system components and flow rates are sized to a specific air flow rate (e.g. 10 l/sm^2) or a function related flow rate (e.g. 15 l/s for a kitchen). Trickle ventilators are sized to reach the design flow rate at a 2 Pa pressure difference.

For the optimization of the system layout, two strategies are implemented, directly related to the number of possible configurations. For the constant flow rate systems presented in Chapter 3, either natural, mechanical exhaust or mechanical supply and exhaust, only the sizing of the 3 main system components, supply, transfer and exhaust, were modified proportionally in 20 steps. This rendered 20^3 possible configurations per system type. For the demand controlled systems, a control is added for both supply and exhaust, each with 3 possible control strategies (CO_2 based, Humidity based or Presence based) and 3 set points. This renders about $20^3 \cdot 10^2$ possibilities.

This set is too large to compute within acceptable time limits. Therefore a selection is made based on a genetic optimization algorithm (e.g. [370]). In such algorithm, the unique parameters of a specific configuration are compiled to form a 'genome', represented in Table 4.4. Again, as with the Monte Carlo, the number of parameters is relatively small, so a genepool of 100 configurations is considered sufficient. The configurations are then simulated as described above and the results are compared. A configuration is considered dominated by another configuration if that last configuration scores better on all of the performance indicators selected. Only configurations that are not dominated by any of the other configurations are considered Pareto optimal. After evaluation, the Pareto optimal configurations are kept and a new generation is bred by combining the genome of 2 configurations into a new genome. The probability of selection is inverse to the number of times the configuration is dominated. This method is continued for 100 generations, since preruns have demonstrated that this provided an acceptable number of Pareto optimal solutions.

Table 4.4 Possible configurations for demand controlled systems

parameter	value
size supply	0.1 - 2 l/sm (step = 0.1)
size transfer	0.1 - 2 l/sm (step = 0.1)
size exhaust	0.1 - 2 l/sm (step = 0.1)
control supply	RH - CO2 - PRES
set point supply	60,70,80 % - 400,600,1000 ppm - 0,1
delay supply	0 - 5 - 10 minutes
control exhaust	RH - CO2 - PRES
set point exhaust	60,70,80 % - 400,600,1000 ppm - 0,1
delay exhaust	0 - 5 - 10 minutes

4.4.2 Results

Figure 4.11 and Figure 4.12 depict the Pareto optimal system configurations with a leakage level of 1 $m^3/h/m^2$ for the demand controlled mechanical exhaust and mechanical supply and exhaust systems respectively. The average and the 95th percentile ($P_{0,95}$) concentration of carbon dioxide during occupancy are shown for both the optimized demand controlled systems and the optimized constant flow systems. Mutatis mutandis, similar results were found for leaky buildings and for exposure to humidity and odours.

Detection of CO_2 in the occupied zones (living and bedroom area) is almost invariably combined with occupancy detection in the wet zones for the Pareto optimal solutions for both the exposure to metabolic CO_2 and humidity. In the configurations with the lowest energy loss, the occupancy detection in the wet zones is sometimes replaced with CO_2 detection since this will only activate the system after a certain time of continued presence in the room.

From Figure 4.11 and Figure 4.12, one can conclude that demand controlled systems and constant flow rate systems achieve similar performances at the extremes of the spectrum. Both minimal exposure and minimal heat loss are comparable. In the midrange spectrum however, where heat loss and exposure are traded off against each other, the demand controlled systems have a clear advantage. Their performance dominates that of the constant flow rate systems , both considering average and peak exposure. A performance gain by heat loss reductions of up to 60% can be achieved with similar exposure for both types of systems. In absolute terms, like was found in Chapter 3, the mechanical supply and exhaust ventilation systems have a slight advantage over the mechanical exhaust ventilation systems.

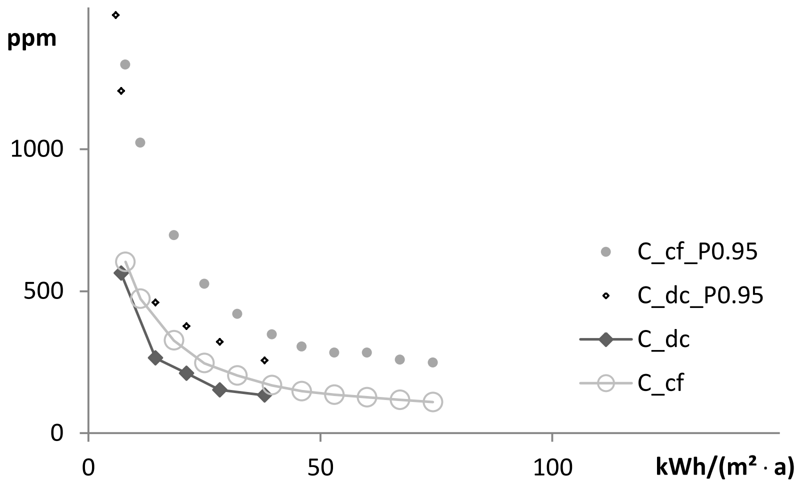

Figure 4.11 Average and peak ($P_{0.95}$) excess carbon dioxide exposure during occupancy for the Pareto optimal sizing options with constant flow mechanical exhaust ventilation (C_cf) systems and demand controlled mechanical exhaust ventilation (C_dc) systems

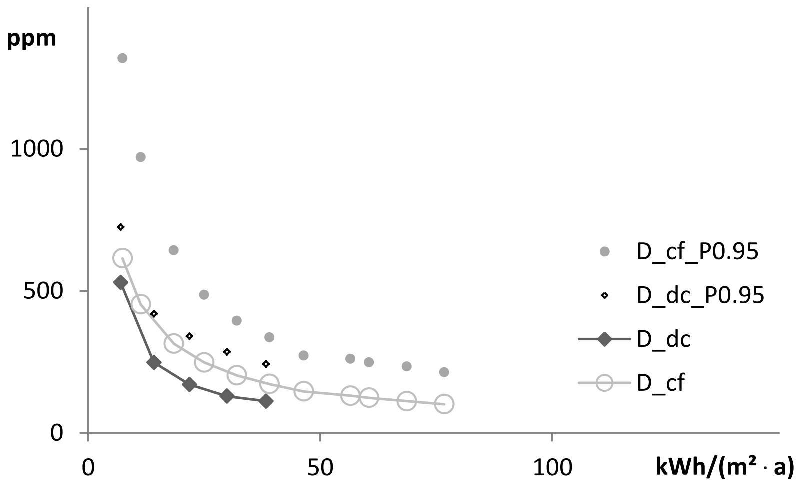

Figure 4.12 Average and peak ($P_{0.95}$) excess carbon dioxide exposure during occupancy for the Pareto optimal sizing options with constant flow mechanical supply and exhaust ventilation (D_cf) systems and demand controlled mechanical supply and exhaust ventilation (D_dc) systems

4.5 Case Studies

Time use studies indicating that 70 % of our time is spent at home and 50% of that time is spent alone [357] clearly show the potential for demand control in dwellings. This is confirmed by the results published on simulation studies [347, 417-419] and in situ measurements [334, 411] report energy saving potentials of about 45% on average and up to 70% in some cases.

In the project presented in this section, the performance of four approaches to demand controlled residential mechanical exhaust ventilation systems which are commercially available on the market was monitored in real life implementations. Their performance was assessed both on the level of heat loss through ventilation and on the level of indoor environmental quality.

For the heat loss performance, the fan power was continuously measured to know the exhaust flow. For the indoor environmental quality, the carbon dioxide concentration, the temperature and the humidity level in each of the rooms in the houses were monitored. The systems and the houses they were integrated in were modelled in multi zone simulation software and the predicted performance was compared to the monitoring results. Information on the interactions of the occupants with the system was gathered by questionnaires about the occupants' habits.

The results also show that the performance of the monitored systems, as far as ventilation heat loss is considered relative to that of a constant air volume mechanical exhaust system, is severely limited by adventitious ventilation and infiltration and is largely dependent on the occupancy and other specifics of the particular case. This indicates that the systems react appropriately to changes in pollution source strength, achieving a better exposure to heat loss trade off than constant volume flow systems.

4.5.1 Cases

As was mentioned in the introduction, 4 cases located in Belgium were studied. In each of the cases a different approach to demand controlled mechanical exhaust ventilation was implemented in a recently built dwelling. All systems used in the cases presented are commercially available as residential ventilation systems on the Belgian market and their energy saving potential has been assessed using the equivalence procedure proposed in the building code [401]. This assessment approach was discussed in a previous section of this chapter.

The ventilation scheme for each of the cases is based on the Belgian ventilation code [364], which was discussed in Chapter 2. This implies that trickle ventilators are installed as natural fresh air supplies in the main living spaces and transfer devices in interior doors allow the air to flow to the service areas (kitchen, service rooms, bathrooms and toilets) where it is exhausted trough exhaust grilles. Although this is not required by the standard, in all cases, ductwork connects all grilles to a single exhaust fan. The flow rates or sizing required by the standard for each of these components is shown in table 1. The design pressure difference for non-mechanical components is 2 Pa.

Table 4.5 Require flow rates in NBN D 50-001 (m^3/h)

Supply	$m^3/(h{\cdot}m^2)$	Exhaust	m^3/h
Living room	3.6	(Open) Kitchen	(75) 50
Study	3.6	Bathroom	50
Bedroom	3.6	Toilet	25
		Service room	50

Case 1: RH and Presence Controlled Exhaust Grilles

The first case is a 2 story semi-detached single family dwelling. It was built in 2008 and has 4 occupants, of which 2 adults and 2 children. The adults both have fulltime jobs and both the children are in school. Each of the parents works from home 1 full day a week. The dwelling has an open kitchen area in the main living space. An open staircase also connects this space to the upstairs landing.

The ventilation system [420] in the dwelling was installed during the main construction period. The trickle ventilators are self-regulating in order to provide the design flow rate, within certain boundaries, at positive pressure differences over 2 Pa. Under the 2 Pa threshold and at negative pressure differences, they behave as regular openings with a flow exponent of 0.5. The system is equipped with relative humidity or presence controlled exhaust grilles. In the kitchen, service room and bathroom, the exhaust grilles assume a fully open position if relative humidity is higher than 80%, and close in proportion to relative humidity to a minimum of 15% of the fully opened position at about 30% relative humidity. Since this is a mechanical process, the grilles are tailor made to produce the desired humidity/flow curves at 100 Pa. The actual curve may vary with the pressure distribution in the ventilation system.

The exhaust grille in the kitchen, bathroom and toilet is equipped with a battery operated infrared presence detection and an electric motor that switches the grille between a fully opened position when presence is detected. This position is maintained for 30 minutes after the last detection of presence, after which a minimum position of about 15% of that fully opened position is assumed.

The user can interact with the system using a 3 position frequency control switch for the exhaust fan, located in the main living space. The opening and closing of the exhaust grilles does not affect the fan speed and will only impact the pressure loss in the system. This will in turn change the system curve and operating point of the fan.

Case 2: RH and Presence Controlled Dampers

The second case is a 2 story terraced dwelling built in 2001. There is only one occupant, working full time. It has a garage that is integrated in the heated volume, an open kitchen area connected to the main living space and a separate stair case.

The ventilation system [421] was installed in 2010. The trickle ventilators are of the same self-regulating type as was described in case 1. The exhaust flow is modulated using electronically controlled dampers installed just downstream of the fan. Each damper is connected to a single exhaust grille. The exhaust fan is equipped with constant pressure controls.

The dampers connected to the kitchen, service room and bathroom exhaust grilles are controlled by a relative humidity sensor integrated in the damper. A numeric algorithm is used to compensate for the change in relative humidity in the exhausted air due to temperature changes between the exhaust grille and the damper. The damper has 2 positions, fully open and 15% of the latter. The setpoint to switch between the 2 positions is 80% relative humidity with a hysteresis band of 10 percent point.

The dampers connected to the bathroom and toilet exhaust grilles are controlled by a wireless battery operated infrared presence detection in the space, switching the damper between the fully opened and minimum position, with a 30 minute delay on assuming the minimum position after the last detection of presence.

Case 3: CO_2 Controlled Trickle Ventilators, RH and Presence controlled fan

This is a 2 story terraced house built in the 1950's. The dwelling was fully renovated in 2009 and now has 5 occupants, 2 adults with fulltime jobs and 3 school aged children. Both the kitchen and the staircase are separated from the main living space.

The ventilation system [422] was installed beginning of 2010 as a final phase of the 2009 general renovation. The trickle ventilators are self-regulating as was described in the first case, but are additionally equipped with electro motors. They are controlled by carbon dioxide sensors in the space they provide with fresh air. They assume a fully open position corresponding to the design flow rate at concentrations higher than 1000 ppm and linearly close to a minimum position of 10% of the fully opened position with decreasing concentrations over 150 ppm range, offsetted by a 150 ppm hysteresis band.

The exhaust fan is frequency controlled by relative humidity sensors in the kitchen, bathroom and service room, and by presence detection in the bathroom and toilet. In function of the highest humidity level measured, a numeric algorithm selects one of 4 fan flow rates: 12.5, 33, 66 or 100 % of the design flow rate. If presence is detected, the highest speed is selected for at least 30 minutes. Additionally, the occupant can overwrite the control algorithm by selecting one of flow rates on a control panel in the main living space.

Case 4: CO_2, RH and Presence Controlled Dampers

The last case is a 2 story detached dwelling built in 2010. It has 3 occupants, 2 adults with full time jobs and 1 school aged child. It has an open kitchen area in the main living area and a separate stair case to the second floor.

The ventilation system [423] was installed during the main construction phase. It is equipped with the same type of self-regulating trickle ventilators as the 3 previous cases. In contrast to the other cases, all but one trickle ventilators face to the side of the dwelling.

The ventilation system has additional exhaust grilles in all living spaces in the dwelling. All exhaust grilles, those in the service spaces as well as those in the living spaces, are connected to a single exhaust plenum, situated just upstream of the fan, with an electronically controlled damper. Each damper is connected to a single grille. The exhaust fan is equipped with constant pressure controls.

The dampers are controlled by a single carbon dioxide sensor situated in the plenum using a PID control loop. A numerical algorithm is used to calculate the concentration in each individual space. The set point for the control loop is 1200 ppm. In addition, the dampers are fully opened if presence is detected in the toilet or bathroom. For the dampers connected to a service space, a minimum flow rate of 10% of the design flow rate is maintained at all times.

4.5.2 Monitoring: Strategy

A monitoring campaign in each of the cases described in the previous sections was carried out to characterize the indoor climate produced by the different approaches to demand control. In this section, the methods used in the monitoring campaign are presented. It has to be stressed that the goal of the monitoring period was not to compare the results for the different systems to one another in order to rank them. Due to the obvious dissimilarity of the boundary conditions, this is not possible.

Measurements

In each of the cases, a pressurization test in conformity with EN 13829 [288] was executed. The dwellings were tested to a pressure difference of 100 Pa, or a maximum leakage flow rate of 2500 m3/h, in steps of 10 Pa.

The outdoor climate was characterized by monitoring temperature and relative humidity on site for each of the cases. Temperature and relative humidity were also measured in every room of the dwelling over the whole measurement period in 5 minute intervals. In the main living spaces and occupied bedrooms, carbon dioxide concentrations were monitored to assess indoor air quality, again at 5 minute intervals.

The fan power was continuously monitored at 1 minute intervals. The exhaust flow rate at each individual exhaust grille was measured once with the fan operating at the design flow rate, using a calibrated orifice.

Questionnaires

In order to gather information about other boundary conditions such as occupancy and occupancy interaction with the ventilation system, the occupants were asked to complete a questionnaire at the beginning of the monitoring period.

Operating Conditions

Each of the cases described in the previous section was monitored under three different operating conditions for at least a week.

In the first test period, the ventilation system was run according to its demand controlled algorithm as was described above. The occupants were asked not to interact with the system as usual. This case allows the performance of the system to be analyzed under ‘factory conditions’.

In a second test period, the controls of the system were disabled and the system was run at its design flow rate continuously. In this case, the occupants were asked not to interact with the system. This case allows the saving potential of the demand control approach in this particular implementation to be characterized and also allows the indoor air quality achieved by the demand controlled system to be compared to that attained under the requirements of the standard.

During the last test period, the controls of the system were adapted so that only a 3 position frequency control switch for the fan was available in the main living area. All other controls were disabled. The occupants were asked to manually select a flow rate (minimum, intermediate, design) and change it whenever they felt the need to do so. This case conforms to the control possibilities available to the majority of occupants of dwellings equipped with mechanical exhaust ventilation systems [424]. This test period allowed the indoor air quality typically found in dwellings with mechanical exhaust ventilation systems to be assessed and is used as a second benchmark for the indoor air quality achieved by the demand controlled system.

4.5.3 Monitoring: Results

In this section the results of the monitoring will be discussed. The fan power measurements revealed that occupants rarely change the position of a 3 position switch once they have selected a position. This confirms data reported based on questionnaire data [424] and in building surveys [425]. In cases 1 and 2, the intermediate flow rate was chosen as the default flow rate, in case 3 and 4 lowest flow rate was the primary selection. In cases 1, 2 and 4 the highest flow rate was selected in 1.5, 0.5 and 3.5 % of the test period respectively. The occupants report that they only select this flow rate upon acute problems such as accidentally burning food. Only in case 3 was the switch used more frequently. The lowest flow rate was selected in 64%, the intermediate flow rate in 13% and the highest flow rate in the remaining 13% of the time. The average of the measured fan power under the three operating conditions is shown in Figure 4.13 for all 4 cases. This amounts to fan energy savings of in the demand controlled condition of 21, 85, 75 and 86 %, respectively when compared to continuous operation at the design flow rate. This corresponds to flow rate reductions of about 57, 63, 54 and 70 %. These numbers are less accurate than the fan power numbers since flow rate was not measured continuously. The lower reductions in flow rate in case 1 and 3 are due to the fact that fan speed is not changed in the former and to the more stringent set point used in the latter.

The results for the leakage tests are shown in Table 4.6, while the indoor air quality results will be presented case by case. Note that only case 4 has a good airtightness. The 3 other cases fall within the leakiest quartile of recently built Belgian residences [386]. The results for indoor air quality and fan power are presented for each of the test periods. Any particular adaptations to the system that were made to achieve the 3 operating conditions mentioned above are also reported.

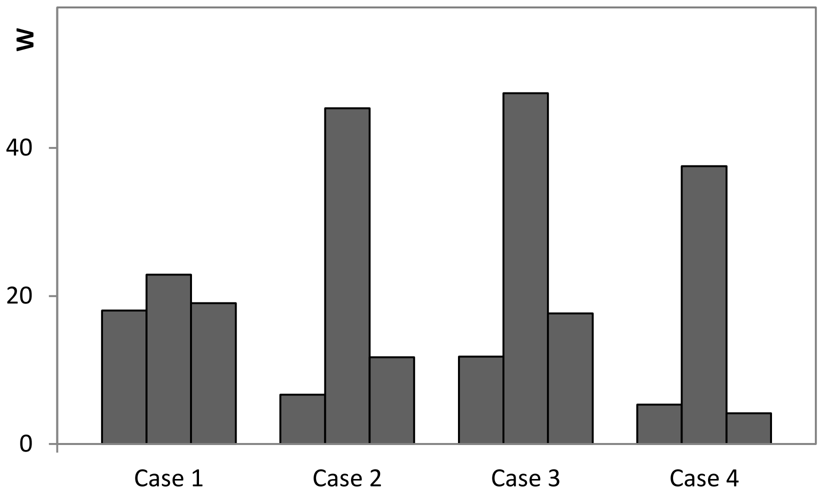

Figure 4.13 Average fan power in all cases under demand controlled, design flow rate and manually selected flow rate operating conditions (W)

The measured carbon dioxide concentrations were categorized according to the IDA classification proposed in EN 13779 [230]. IDA 1, 2 and 3 classes corresponds to concentrations lower than 400, 600 and 1000 ppm over the outdoor concentration respectively. IDA 4 comprises al higher concentrations and indoor air in this class is considered to be of poor indoor air quality. Although information about the mean occupancy schedule of the families was gathered by means of the questionnaires, it was not possible to isolate the periods during the measurements when someone was present in each room. Therefore the indoor air quality classification shown in this section based on the full dataset, including periods of absence. The fraction of time where good excellent air quality is found will therefore be greater than that to which the occupants are exposed.

Table 4.6 Leakage air change rate at 50 Pa pressure difference

Case	n50 (ACH)
1	9.3
2	10
3	13
4	2.8

Case 1

In the first case, the relative humidity or presence controlled exhaust grilles were removed during the second and third test period. Subsequently, the maximum fan speed was adjusted to conform to the design flow rate again, as the removal of the grilles changed the pressure drop and operating point of the system.

The measured flow rates at full fan speed were uniformly found to be about 45 m^3/h in all service spaces. For the bathroom and service room, this is more or less in agreement with the 50 m^3/h design flow rate required in the standard. For the open kitchen and toilet, however, the standard requires 75 and 25 m^3/h respectively.

As can be seen in Figure 4.14, the demand controlled configuration produced the best indoor air quality. However, the difference with the air quality achieved by the other operating conditions is not significant, especially if the fact that the boundary conditions, for example occupancy time and weather conditions, were not identical in all three test periods. These non-equal boundary conditions might account for the fact that the indoor air quality was lowest in the second test period, which none the less operated at the highest (design) flow rate. Additionally, the primary flow rate in the third test period was, as was mentioned before, the intermediate flow rate. This flow rate was still about 90% of the design flow rate, selected in the third period, further adding to the small difference in the obtained results for both periods.

It is, in any case, worth noting that the overall air quality is excellent and that the impact of the ventilation system on is minimal. This is mainly attributable to the high leakage level, and the infiltration flows associated with it.

Case 2

The flow rates measured in the second case were in good agreement with the design flow rates specified by the standard, except in the bathroom, where 19 m^3/h was measured instead of the required 50 m^3/h.

The differences between the first and last test period in the living room (Figure 4.15) could be explained by better matching of flow rate and emission. The virtually identical air quality in the bedroom is most likely due to low air exchange of this room with the rest of the dwelling and the high infiltration rates. This will be discussed further in simulation section. The reduction of air quality in the demand controlled test (although not significant) indicates a mismatch between control parameter (humidity) and exposure to the measured pollutant (carbon dioxide) for this specific environment (bedroom).

Case 3

Again the low and sometimes counter intuitive impact of the operating condition of the ventilation system is worth noting. In this case again, the high leakage level of the building is the most likely cause. The decrease in air quality observed in bedroom 3 during the second test period, that has the highest flow rate, could be explained by the fact that the trickle ventilator in this room is located on the leeward side of the prevailing wind direction. Increased exhaust flow rate will decrease the under pressure over trickle, decreasing net flow rate.

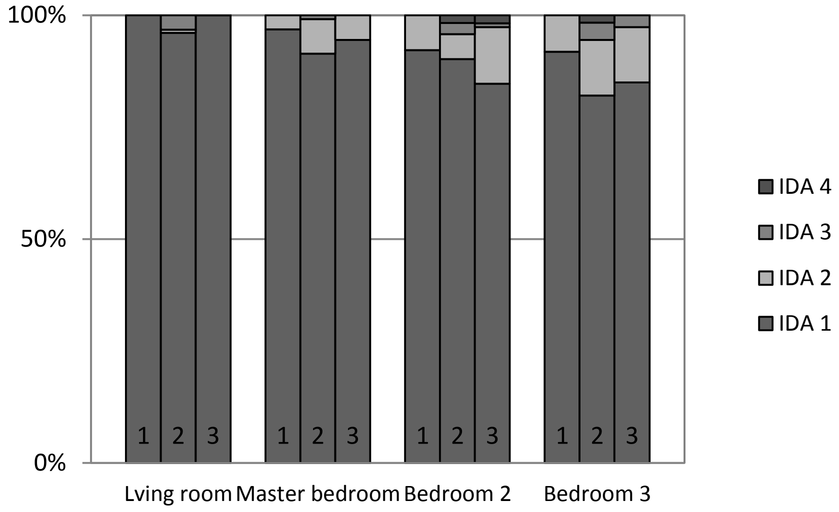

Figure 4.14 Indoor air quality in living room and bedrooms, classified by IDA classes for each of the 3 monitored operating conditions for case 1.

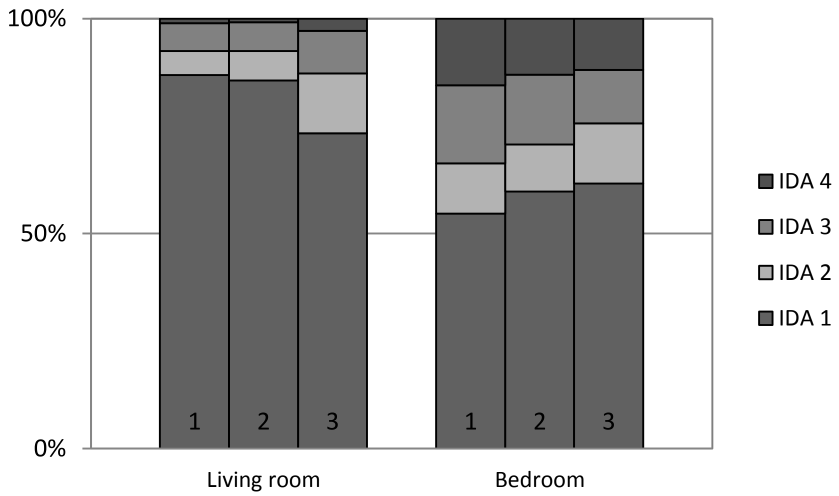

Figure 4.15 Indoor air quality in living room and bedrooms, classified by IDA classes for each of the 3 monitored operating conditions for case 2.

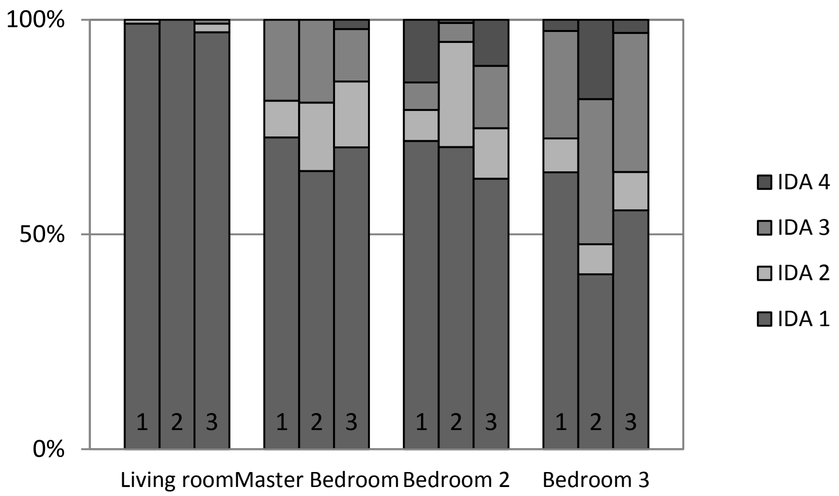

Figure 4.16 Indoor air quality in living room and bedrooms, classified by IDA classes for each of the 3 monitored operating conditions for case 3.

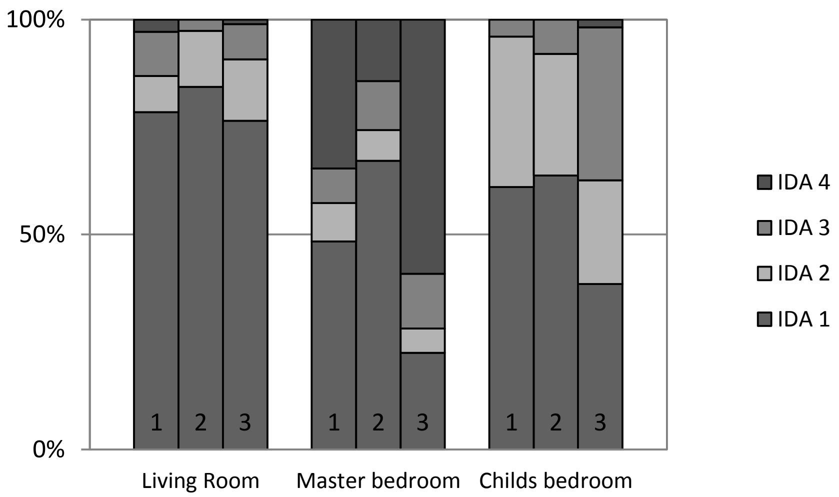

Figure 4.17 Indoor air quality in living room and bedrooms, classified by IDA classes for each of the 3 monitored operating conditions for case 4.

When test period 1 and 3 are compared (Figure 4.16), the demand controlled system provides consistently better air quality than that achieved by manual control, although the average flow rate in the latter is notably higher.

Case 4

As can be seen in Figure 4.17, the indoor air quality achieved with the demand control is significantly better than that achieved by manual control, especially in the bedrooms, although the flow rate is more or less the same.

Additionally, in contrast to the results seen in the first 3 cases, the tight building envelope ensures a direct correlation between the flow rate and achieved indoor air quality. Moreover, the overall indoor air quality is significantly lower than in the 3 other cases due to the absence of large infiltration rates.

4.5.4 Simulations: Setup

Due to adventitious ventilation and infiltration, the total heat loss by ventilation is higher than that by the total mechanically exhausted flow rate. Time dependent infiltration and adventitious ventilation, on the other hand, are extremely hard to measure. Time dependent data is needed, because the pressure differences generated by continuous change in flow rates due to the demand control.

Therefore, multi-zone models were developed for each of the cases. The models will allow the impact of adventitious ventilation and infiltration on the total heat loss savings that can be obtained by demand control to be estimated. The models were developed in the multi-zone simulation software Contam [375], developed by NIST. Each room in the dwelling was modeled as a separate zone. The airflow in this dwelling has been modelled through the introduction of system components and leakage, closely following the modelling approach presented in the second chapter.

The ventilation system was modelled in accordance with the real ducting layout in each case and flow rates were tuned to the measured flow rates for each room. As was mentioned before, each of the systems installed in the different cases was already assessed with the equivalence procedure required by the building code. The modelling of the control algorithms for the different systems were copied from the models that were developed for the equivalence procedure.

The occupancy schemes used in the model were based on the answers from the questionnaires filled out by the occupants. More details on the model, eg. on the modeling of the pollutant sources, were discussed in chapter 2. Temperature, humidity and wind boundary conditions were taken from the TRY weather file for uccle. In contrast to occupancy and manual flow rate selection, that are rather stable over time and can thus be measured in the relatively short test periods of 1 to 2 weeks, the heat loss associated with adventitious ventilation and infiltration is highly dependent on the climatic boundary conditions. Therefore, the simulations were run over the full length of the heating season (September to April).

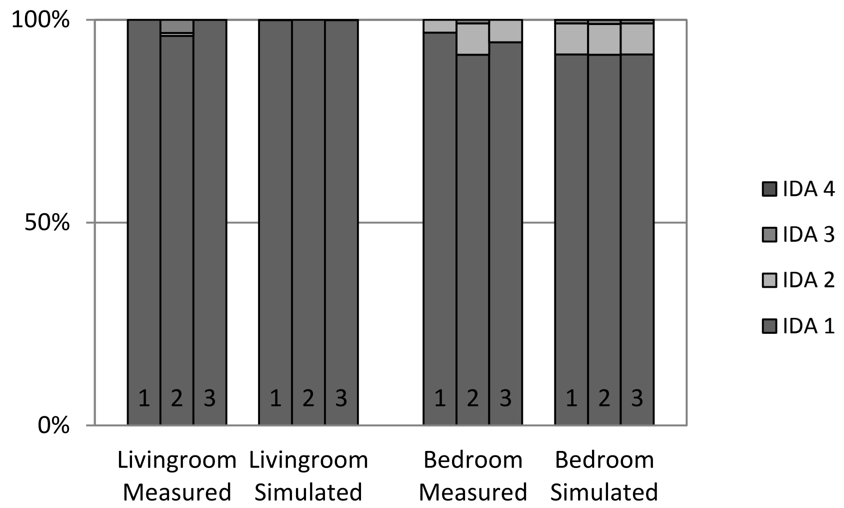

Figure 4.18 Measured and simulated indoor air quality in living room and master bedroom, classified by IDA classes for each of the 3 monitored operating conditions in case 1

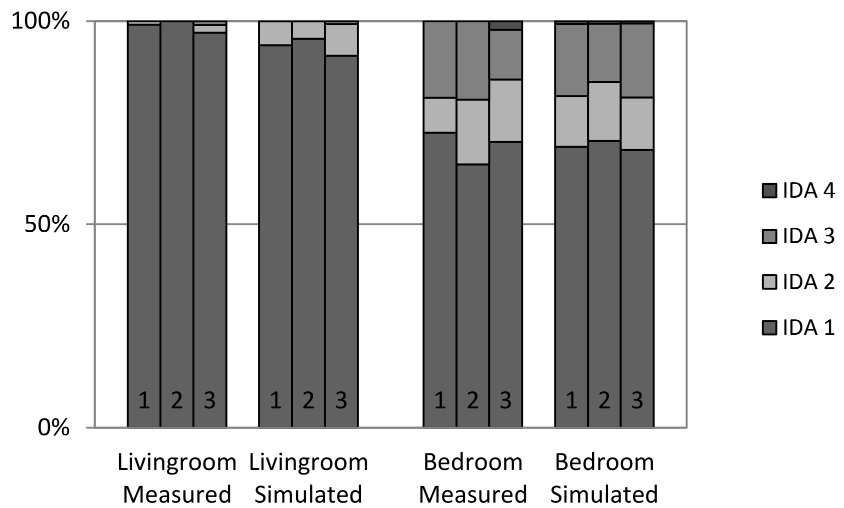

Figure 4.19 Measured and simulated indoor air quality in living room and master bedroom, classified by IDA classes for each of the 3 monitored operating conditions in case 3

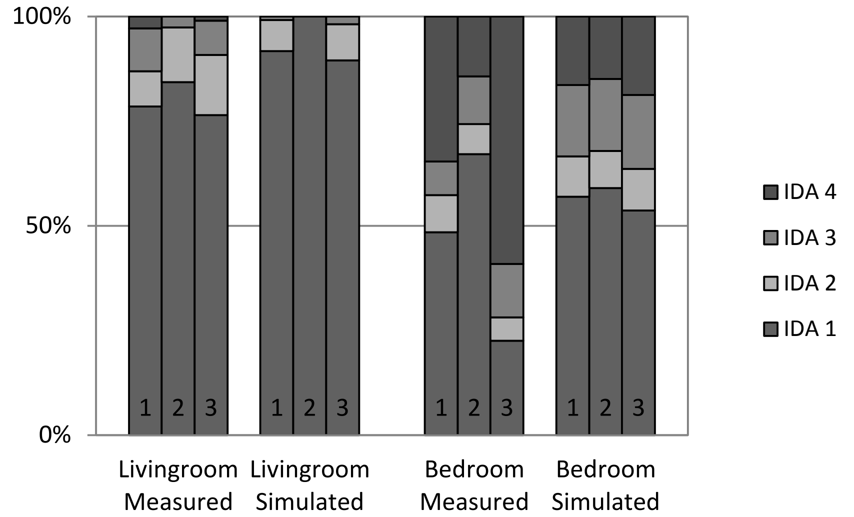

Figure 4.20 Measured and simulated indoor air quality in living room and master bedroom, classified by IDA classes for each of the 3 monitored operating conditions in case 4

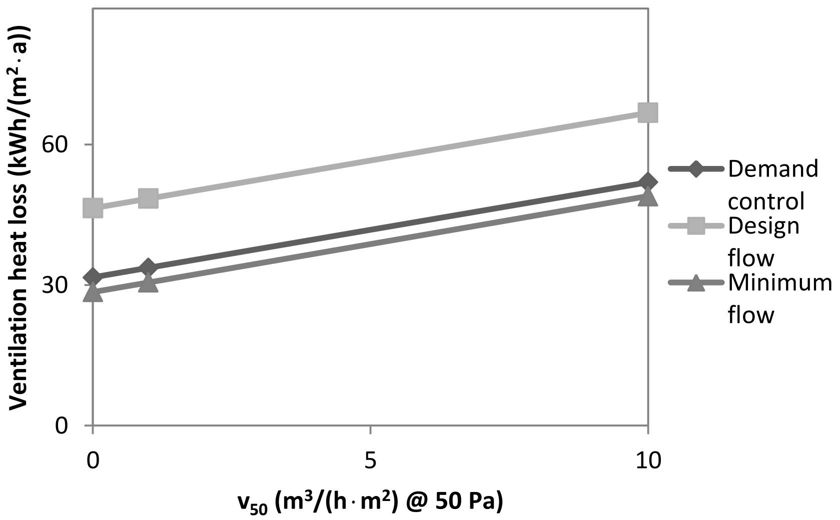

Figure 4.21 Extrapolation to 0 leakage level for case 2

For each of the cases, a total of 6 simulations were made. The first set of 3 simulations corresponds to the 3 test periods described above, with the exception of the last. Since to many assumptions would have to be made to simulate manual control, and the results from the measurements demonstrate that occupants tend to make very little use of the three position switch, this period was simulated with the ventilation system running continuously at the lowest flow rate. By comparing the results of the first and second simulation in this first set, an estimation of the real heat loss reduction, taking adventitious ventilation and infiltration into account, can be obtained.

The second set of 3 simulations is done under the same operating conditions, but with the leakage level of the houses reduced to 1 ACH at 50 Pa pressure difference. By making an extrapolation based on the results from both sets, the heat loss at the 0 leakage level can be obtained. This allows the influence of infiltration to be discerned from that of adventitious ventilation. While the former is a good indicator for the quality of the building envelope, the latter is purely caused by trickle ventilator sizing and the geometry of the dwelling. The thus obtained ventilation heat loss reduction excluding infiltration is then compared the performance of the demand controlled system to be taken in account in the energy performance assessment of buildings, assessed by the equivalence procedure, which uses this same metric.

4.5.5 Simulations: Results

In this section, the results obtained with the simulations will be discussed. They will be compared to the results obtained in the measurements in a qualitative manner. Quantitative comparison is not possible due to the fact that the boundary conditions used needed the simulations, especially occupancy and wind conditions, were not measured during the monitoring phase and could therefore not be used in the model. Additionally, the distribution of leakage over the building envelope is unknown and is, as was mentioned before, considered to be uniform, while in reality it is more likely to be concentrated at certain faulty joints [293, 426].

Figure 4.18 and Figure 4.19 show that the distribution of indoor is rather well predicted by the model for cases 1 and 3. For cases 2 and 4, the models underestimate the carbon dioxide concentrations, as can be seen in Figure 4.20 for case 4. An example of the extrapolation of the results to perfect air tightness is shown in Figure 4.21.

Table 4.7 lists the heat loss reductions taking into account only mechanical flow rates, mechanical flow rates and adventitious ventilation and mechanical flow rates, adventitious ventilation and infiltration, respectively. The influence of the latter two is clearly demonstrated, leading to heat loss reductions 2 to 4 times smaller than anticipated when only mechanically exhausted flow rates are considered.

Table 4.7 Heat loss reduction taking into account mechanical flow rates (MF), adventitious ventilation (AD) and infiltration (IN) as well as the reduction for mechanical flow rates and adventitious ventilation found with the equivalence procedure in the building code

Case	MF	MF + AD	MF + AD + IN	Code (MF + AD)
1	0.57	0.30	0.15	0.16
2	0.62	0.32	0.22	0.33
3	0.54	0.09	0.15	0.53
4	0.70	0.60	0.37	0.28

When the results are compared to those found with the equivalence procedure, only case 2 corresponds well to the performance as calculated by the procedure. This indicates the high impact of the boundary conditions of a specific case on the results. This high sensitivity of the results to the specifics of a particular implementation are also found in the results of the equivalence procedure with typical 95 % confidence intervals of 25 to 30 per cent points. In case 3 for example, the higher number of occupants, 5, instead of the average 3.34 in the equivalence procedure is a large contributor factor to its low reduction factor. The results for case 3 are also particularly interesting since this the only case where the demand control seems to detect and compensate for the fresh air coming in by infiltration.

4.5.6 Summary

This section presents measurement and simulation results for 4 cases with different approaches to demand controlled residential mechanical exhaust ventilation systems. The results presented include monitoring data over 3 consecutive periods of 1 to 2 weeks, representing different operating conditions of the systems, and whole heating season simulations for each of the cases.

Not surprisingly, very good indoor air quality was measured in the cases with very leaky envelopes. The monitoring results also showed that in these cases, the influence of the ventilation systems on the level or air quality that is achieved in the various spaces of the dwelling is very limited.

Fan power reductions found reached up to 85 %, but in a case without frequency controlled fan, this reduction was only 20 %. Exhaust flow rate reductions were between 55 and 70 %. Taking estimations for adventitious ventilation and infiltration from the simulations into account, however, the estimated heat loss reductions for ventilation were only 15 to 40 %. In comparison to the stochastically averaged results obtained using the equivalence procedure included in the building standard, rather large variability in estimated heat loss reductions were found, depending on occupancy, geometry and other elements specific to the particular cases. This large variability is in agreement with that found in the equivalence procedure.

4.6 Conclusions

In this chapter, the energy saving potential of demand controlled ventilation was investigated.

Again starting from the 'state of the art' based on a performance assessment of systems available on the Belgian market, a substantial energy saving potential for demand controlled residential ventilation systems was found. This potential was then confirmed in practice in a series of 4 cases with different approaches to demand controlled residential mechanical exhaust ventilation systems. The results presented included monitoring data over 3 consecutive periods of 1 to 2 weeks, representing different operating conditions of the systems, and whole heating season simulations for each of the cases. Fan power reductions found reached up to 85 %, but in a case without frequency controlled fan, this reduction was only 20 %. Exhaust flow rate reductions were between 55 and 70 %. Taking estimations for adventitious ventilation and infiltration from the simulations into account, however, the estimated heat loss reductions for ventilation were only 15 to 40 %.

The analysis was then expanded to the impact of the local context created by ventilation standards or climate confirming the potential that was found in the first sections of the chapter. The results demonstrated that the proposed definition for an energy reduction coefficient renders a relatively robust estimate for the energy saving potential of a specific demand control approach.

Finally, further analysis demonstrates that optimized demand controlled systems and constant flow rate systems achieve similar performances at the extremes of the spectrum. Both minimal exposure and minimal heat loss are comparable. In the midrange spectrum however, where heat loss and exposure are traded off against each other, the demand controlled systems have a clear advantage. Their performance dominates that of the constant flow rate systems. The performance gain is most notable in the airtight case, where heat loss reductions of up to 60 % can be achieved with similar exposure.

5

Achievable Performance: Heat Recovery Units

This chapter discusses the performance of heat recovery technology in mechanical supply and exhaust systems by addressing the inherent conversion problem that it presents. The application of heat recovery ventilation reduces the ventilation heat losses considerably, but requires additional fan energy to operate, therefore expending electrical power to save heat.

In the first section of the chapter, the impact of different conversion frameworks on the predicted performance of air to air heat recovery units for mechanical supply and exhaust systems is investigated. Since this technology cannot easily be applied to mechanical exhaust ventilation, other approaches such as exhaust air heat pumps have been proposed as an alternative heat recovery technology specifically for mechanical exhaust ventilation. Using the equivalent heat recovery effectiveness, introduced in the second part of the chapter, allows to compare the performance of both technologies. The results of this chapter have been published in Energy and Buildings [282].

5.1 Introduction

Since the results of the optimization study presented in chapter 3 have demonstrated that the optimal performance of appropriately sized balanced mechanical ventilation without heat recovery, mechanical exhaust ventilation and natural ventilation is comparable with regard to both heat loss and mean exposure to pollutants, no a priori towards one of them can be assumed.

Balanced mechanical supply and exhaust, on the other hand, has the advantage over the other system approaches that air-to-air heat exchangers can be added to the concept to achieve heat recovery between supply and exhaust air, thus considerably reducing ventilation heat losses. Nevertheless heat recovery ventilation operation is faced with a trade-off: an increased pressure drop due to the narrow passages in the heat exchanger unit for heat recovery ventilation companions the decrease in heat loss due to that same heat recovery. The increased pressure drop will in turn lead to a higher electric energy use for the fan, while the reduced heat loss will lower the space heating load. The balance between both is, beside by system characteristics, strongly affected by climate and by the conversion factor used to compare electricity consumption to fossil fuel consumption. Mechanical exhaust ventilation has the disadvantage that no broadly available technology allows for heat recovery on it, while it still needs electricity for fan operation. Nonetheless, it provides more stable flow conditions than natural ventilation and some of the energy in the exhaust air can be recovered by the implementation of heat pump technology for domestic hot water production or for low temperature heating systems [325-327].

Although numerous papers discuss the performance of heat recovery ventilation [331, 427], the sensitivity of this performance to variations in climatic conditions [318, 332, 428-431] or the impact of mechanical exhaust ventilation and natural ventilation on the indoor environment [432, 433], they are rarely traded off against each other. Dodoo [434], however, presented an analysis of their merits in a very specific context with district heating. Nevertheless, the number of climatic conditions considered is always rather limited.

Therefore, in this chapter, we investigate the trade-off between heat recovery and increased electricity load for the operating phase of heat recovery ventilation for the different climates in Europe by using a set of possible conversion factors for electricity, based on primary energy, carbon dioxide emission, operating cost and exergy frameworks respectively. In the first section of the chapter, the tradeoff is studied using fixed assumptions for the relevant system characteristics and boundary conditions. The sensitivity of the results with regard to these assumptions is then treated in the discussion section.

5.2 Analytical Assessment

The frameworks used to trade off heat loss and electricity consumption all have their specific limitations. When considering carbon dioxide emissions, for example, nuclear power is a positive contributor to emission reductions. This, however, completely neglects all other considerations regarding safety and waste management that are associated with that particular technology. Using four different frameworks allows to grasp more than one of these facets, but is still incomplete and should be supplemented with additional information.

The results presented only consider the operating energy under different trade-off conditions. It has to be stressed that the method proposed is not suited for the prediction of actual performance of a specific implementation of a system, but is merely focused on demonstrating the distribution and variability of the potential for heat recovery ventilation. The potentials, demonstrated in the figures shown in the results section, are valuable resources for fixing stimulus policies and during conceptual design phases. A more complete feasibility assessment for a specific project should be based on precise system characteristics and climate data and should include investment costs, building specific elements such as acoustic nuisance, draft risk or the need to filter contaminated outdoor air and additional operating costs such as maintenance and component replacement (e.g. fans and filters!).

The assessment of heat recovery ventilation made in this chapter only takes into account the intended ventilation. The ventilation systems are all assumed to run at a constant rate, all year long. This is a valid assumption since, although occupants tend to open windows during summer [308], thus increasing the total air flow rate, the system is rarely shut down. The ability to shut the system down is even forbidden in some ventilation standards [364]. For all of the coefficients used, averaged values should be handled with care.

5.2.1 Climate and Recovered Heat

To characterize the different climate conditions, the heating degree day [405] data from Eurostat [404] is used. The data was averaged over a ten-year period from 2000-2009.

This data conforms to the NUTS 2 level as defined by Eurostat [435] which corresponds to a subnational, regional scale for EU Member States, Norway, Turkey, Croatia, Switzerland, Liechtenstein and Iceland. For some regions the data is available on NUTS 3 (city) scale. The data was used in its finest geographical form available.

In accordance with the Eurostat definition of heating degree days (HDD) [436], which assumes a heating threshold of 15 °C and an indoor temperature of 18 °C, the number of heating degree days for any given day is defined as 18 °C minus the daily mean temperature, whenever that daily mean temperature is below 15 °C. The daily mean temperature is defined as the mathematical average of the minimum and maximum temperature of that day. Based on this definition, the average number of HDD for the EU is 3000. The distribution of HDD within the EU is shown in Figure 5.1.

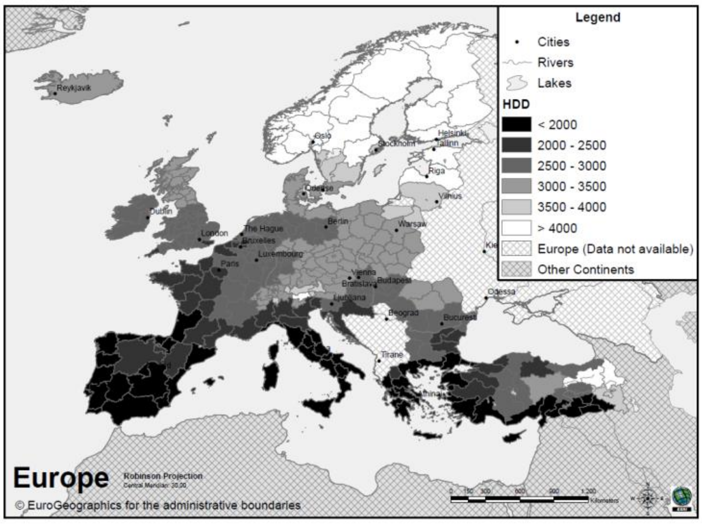

Figure 5.1 HDD in the EU (Kday) [404]

The total heat recovered annually by a heat recovery unit (HRU) is calculated from the heat content of the ventilation air:

$$Q_{HR} = \int_a \rho(p,T) \cdot c(T) \cdot \varepsilon(\dot{g},c) \cdot \dot{g}(t) \cdot \Delta T(t) \cdot dt, \qquad (5.1)$$

with:

Q_{HR}	total annual heat recovered	[J]
ρ	density	[kg/m^3]
c	specific heat capacity	[J/(kg·K)]
p	pressure	[Pa]
ε	effectiveness of heat recovery unit	[-]
$\dot{g}$	flow rate	[m^3/s]
t	time	[s]
T	temperature	[K]
ΔT	indoor/outdoor temperature difference	[K].

Assuming density, specific heat capacity, flow rate and effectiveness to be constant over the whole heating season and assuming 1224 J/m³K to be the volumetric heat capacity of air, normalized per m^3/h, this can be transformed to:

$$q_{HR} = 24 \cdot \rho \cdot c \cdot \varepsilon \cdot HDD = 29376 \cdot \varepsilon \cdot HDD, \qquad (5.2)$$

with:

q_{HR}	total annual heat recovered per m³/h	[Jh/m³]
ρ	density	[kg/m³]
c	specific heat capacity	[J/(kg ·K)]
ε	thermal effectiveness of heat recovery unit	[-]
HDD	number of heating degree days	[Kday].

Balanced heat recovery ventilation can only recover heat from the mechanically supplied/exhausted fraction of the total outdoor air exchange rate. On the other hand, it also only uses electrical energy for that particular fraction. Therefore, infiltration is not accounted for in the method presented here. Multi-zone simulation results [437] demonstrate that the type of ventilation system (balanced mechanical, mechanical exhaust or natural) has only a minor effect on the total infiltration. Although mechanical exhaust ventilation depressurizes the building, thus potentially reducing infiltration, the sizing of trickle ventilators usually decreases this depressurization below the prevailing wind and buoyancy pressure differentials, largely cancelling this effect [363]. Only in the rare cases where an extreme airtightness level and small trickle ventilator sizing are combined should the consequent reduction of infiltration due to the mechanical exhaust ventilation be taken into account. Again this falls out of the scope of this chapter.

5.2.2 Fan Power

The (increase in) electric load for fan operation in the heat recovery ventilation system is highly dependent on fan and ducting characteristics. In addition, fan power will typically increase with higher HRU effectiveness, due to the increased pressure drop.

$$Q_f = \int_a \eta_f(\dot{g})^{-1} \cdot \Delta p(\dot{g}) \cdot \dot{g}(t) \cdot dt, \qquad (5.3)$$

with:

Q_f	total annual fan energy	[J]
Δp	pressure drop	[Pa]
η_f	fan efficiency	[-]
$\dot{g}$	flow rate	[m³/s]
t	time	[s].

Nevertheless, the use of specific fan power (SFP), defined as the ratio of the measured fan power and flow rate, allows for a straightforward approach to it. The European standard EN 13779 [230] specifies SFP 3-4, 750-2000 J/m³ per fan, as default values for heat recovery systems and SFP 2, 500-750 J/m³, for mechanical exhaust systems. For the results presented below, the boundary between SFP 3 and SFP 4, 1250 J/m³ per fan, is used as a reference specific fan power for heat recovery systems, while the upper limit of SFP 2, 750 J/m^3, is used for mechanical exhaust systems.

5.2.3 Conversion Factors

Four different frameworks were used to calculate conversion factors that allow to convert heat loss in systems without heat recovery and electricity consumption in heat recovery systems to a single metric: primary energy, carbon dioxide emission, household consumer price and exergy. The equivalent of the recovered heat in the systems without heat recovery is assumed to be generated with a 100% efficient natural gas combustion system. Although this is a simplification, the performance of modern gas heaters is very close to that value.

To calculate the primary energy factors, denominated f, the 2008 country energy balances reported by the International Energy Agency (IEA) [278, 279] were used. The primary energy factor was calculated by dividing the sum of the primary energy input of electricity plants and CHP plants by the net produced electricity. Since we assume the ventilation systems considered to continuously run at a fixed rate all year long, average factors were preferred to peak load factors. Due to the constraints of the first law of thermodynamics, this factor cannot be inferior to one.

The availability of the data limits the resolution of the calculated factors to country scale. Since, in spite of efforts of the EU to integrate its electricity market, electricity production is still mostly organized on a national scale, this is also the most logical scale within the primary energy framework.

The carbon dioxide emissions related to electricity production were also calculated based on IEA data [280]. Here too, the availability of the data limits the resolution of the calculated factors to country scale. Again, because of the organization of the electricity production in the EU, this is also the most logical scale for this framework.

For the countries where data was available, the prices of gas and electricity for household consumers reported by Eurostat [404] were used to calculate operating costs for non-industrial consumers. Maintenance and additional costs such as filter replacement were not taken into account. The availability of the data limits the resolution of the calculated factors to country scale.

Although energy content of a system may be considerable, it is only relevant to us when it can be transformed into a useful effect, work. Exergy is, in thermodynamics, the amount of work that can be produced by a system. For a heat flux between 2 systems at different temperatures it is limited, according to the second law of thermodynamics, by the Carnot efficiency. If we consider the outdoor temperature to be the temperature of the cold reservoir, that is the reference temperature for exergy calculation, the exergy content of the heat loss in a building is found by the product of that heat loss and the Carnot efficiency, while electricity is pure work/exergy. The exergy content of the heat loss is a measure to assess the efficiency of using electricity to operate a system that compensates this heat loss. Therefore, the Carnot efficiency is a measure of the quality, the usefulness of the energy flows in systems and its inverse can be used as an exergy conversion factor. Additionally, in a heat pump cycle, the inverse of the Carnot efficiency is the limit of the COP that can be obtained between 2 temperatures, the limit of the amount of heat that can be produced per W of electricity input in a fully reversible heat pump cycle.

To calculate the Carnot efficiency, the indoor temperature (T_H) is assumed to be 18 °C, corresponding to the indoor temperature for the calculation of the heating degree days [436] and to the average indoor temperature measured in dwellings [298]. The average outdoor temperature during the heating season is calculated back from the number of heating degree days. Due to the constraints of the second law of thermodynamics, unity is the under limit for this factor. Since in most countries, heating degree day data was available on region or even city scale, the resolution of the calculated factors was fitted to the smallest available data scale.

5.2.4 Minimal Heat Recovery Unit Effectiveness

Based on the calculated total heat recovered annually by the heat recovery unit, fan power difference, the selected conversion factor and assuming continuous operation of the system as well as 100 % efficiency for a condensing gas boiler, the minimal heat recovery unit effectiveness for break-even operation of heat recovery ventilation can be calculated by:

$$\varepsilon = \frac{24 \cdot 365 \cdot \Delta SFP \cdot f}{24 \cdot \eta_h^{-1} \cdot \rho \cdot c \cdot HDD} = \frac{\Delta SFP \cdot f}{3.35 \cdot HDD}, \tag{5.4}$$

with:

ε	effectiveness of heat recovery unit	[-]
HDD	number of heating degree days	[Kday]
ΔSFP	increase in specific fan power	[J/m³]
ρ	density	[kg/m³]
c	specific heat capacity	[J/(kg ·K)]
η_h	efficiency of the heating system	[-]
f	electricity/heat conversion factor	[-].

Obviously, heat recovery operation will only be net positive for the total performance of the ventilation system under the framework considered if the real effectiveness is higher than the minimal effectiveness thus calculated.

Table 5.1 heating degree days (HDD) and conversion factors for 1 J of electricity to 1 J of gas fired heating with primary energy, carbon dioxide emission, household consumer price and exergy ratios for the EU 27 and three typical cases in Europe

	HDD	Primary energy	CO_2 emissions	Consumer price	Exergy ratio
EU 27	3066	2.74	1.72	2.80	22
France	2326	3.21	0.41	1.98	25
Norway	5392	1.14	0.03	-	15
Spain	1773	2.31	1.94	2.55	30

5.2.5 Equivalent Heat Recovery Effectiveness

Alternatively, again based on the calculated total heat recovered annually by the heat recovery unit, associated fan power difference and conversion factors, the equivalent heat recovery unit effectiveness can be calculated by:

$$\varepsilon_{eq} = \frac{\varepsilon_t \cdot \eta_h^{-1} \cdot q_v - \Delta SFP \cdot f \cdot 24 \cdot 365}{\eta_h^{-1} \cdot q_v}, \tag{5.5}$$

with:

ε_t	air to air effectiveness of heat recovery unit	[-]
q_V	flow rate specific ventilation heat loss	[Jh/m³]
HDD	number of heating degree days	[Kday]
ΔSFP	increase in specific fan power	[J/m³]
η_h	efficiency of the heating system	[-]
f	electricity/heat conversion factor	[-].

To assess the performance of exhaust air heat pump (EAHP) based domestic hot water production as a ventilation heat recovery measure, compared to air-to-air heat exchanger technology, a similar definition of equivalent heat recovery effectiveness ε_{eq} is proposed. ε_{eq} is defined as the ratio of the energy savings achieved by the exhaust air heat pump system in comparison to a reference system, converted to heat, and the ventilation heat losses:

$$\varepsilon_{eq} = \frac{\eta_{DHW}^{-1} \cdot Q_{DHW} - f \cdot SPF^{-1} \cdot Q_{DHW}}{\eta_h^{-1} \cdot Q_v}, \tag{5.6}$$

with:

Q_{DHW}	annual energy for domestic hot water	[J]
Q_v	annual ventilation heat loss	[J]
SPF	seasonal performance factor EAHP	[-]
η_{DHW}	efficiency of a gas fired DHW boiler	[-].
η_h	efficiency of the heating system	[-]
f	electricity/heat conversion factor	[-].

By comparing the equivalent heat recovery effectiveness of both technologies under a specific set of boundary conditions, the most appropriate technology can be selected. In a feasibility study for a specific project, the different investment and maintenance costs will have to be taken into account.

5.3 Results

The different conversion factors for three typical countries, along with the heating degree days in Europe are shown in Table 5.1. Norway has the lowest numbers across the board (although household consumer price data for this country are not available). For the exergy factor, this is due to the low outdoor temperature, also reflected in the high number of heating degree days. The low values for primary energy and carbon dioxide emissions are due to the virtually exclusive (98 %) use of hydro power for electricity production in Norway. Although France also has a low carbon dioxide emission factor, this is due to an entirely different energy mix, for electricity production in France is dominated (88 %) nuclear power, supplemented by coal plants. Spain, at last, has primary energy, carbon dioxide and household consumer price conversion factors that are very close to the EU 27 average, but, due to its warmer climate, faces a low recovery potential and a high exergy conversion factor.

The minimal HRU effectiveness for the different regions in the EU when no conversion factor is considered ($f = 1$) is shown in Figure 5.2. This figure gives an overview of the contribution of climate differences to the spread in minimal HRU effectiveness.

Figure 5.4. shows the minimal effectiveness of the heat recovery unit for break-even operation of heat recovery ventilation considering primary energy as a framework to trade off electricity use and heat loss for the different regions in Europe. The regions are colour-coded in grayscale according to the minimal HRU effectiveness in 20 % steps from white to black. Regions with minimal HRU effectiveness over 100% are hatched. The same representation in used in Figure 5.5 to Figure 5.11 for the other conversion factors.

Based on primary energy, a line north of which heat recovery ventilation under realistic conditions allows for net savings compared to natural (ΔSFP = 2500 J/m^3) can be traced between Paris and the Black Sea, roughly corresponding to 2500 HDD. Below this line, unrealistic HRU effectiveness (even higher than 1) would be needed to compensate for the rise in electricity load.

Nonetheless, some interesting exceptions to this general conclusion can be seen in Switzerland, Austria and Northern Spain. Due to a higher fraction of renewable energy sources in their electric energy mix along with the colder alpine climate, these regions perform relatively better than the rest of Southern Europe.

When heat recovery ventilation is traded off against mechanical exhaust ventilation (ΔSFP = 1750 J/m^3, Figure 5.5), profitability is extended southward to the latitude of Corsica.

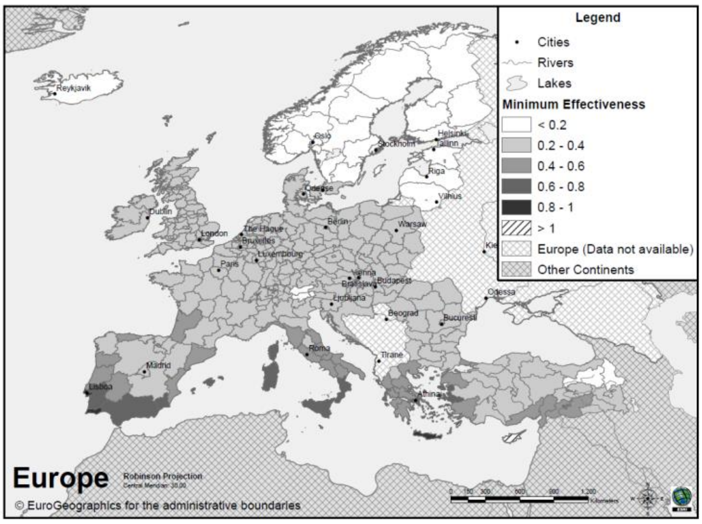

Figure 5.2 Minimal HRU effectiveness for break-even operation with Δ SFP 2500 J/m³ calculated with conversion factor f =1

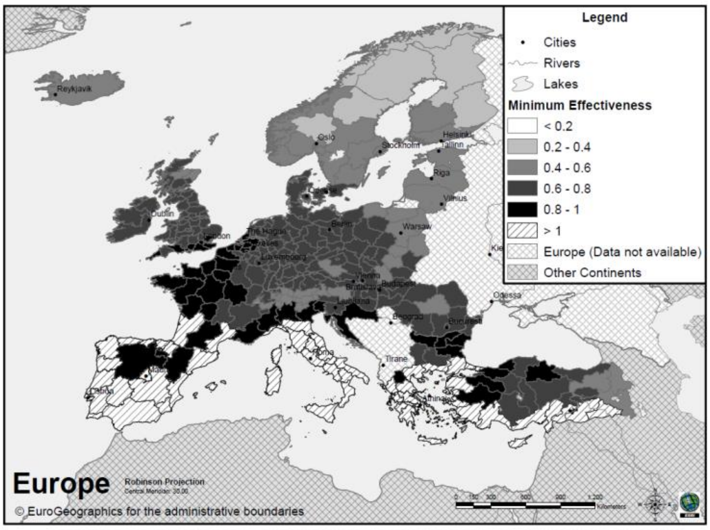

Figure 5.3 Minimal HRU effectiveness for break-even operation with Δ SFP 2500 J/m³ calculated with the average EU27 primary energy conversion factor

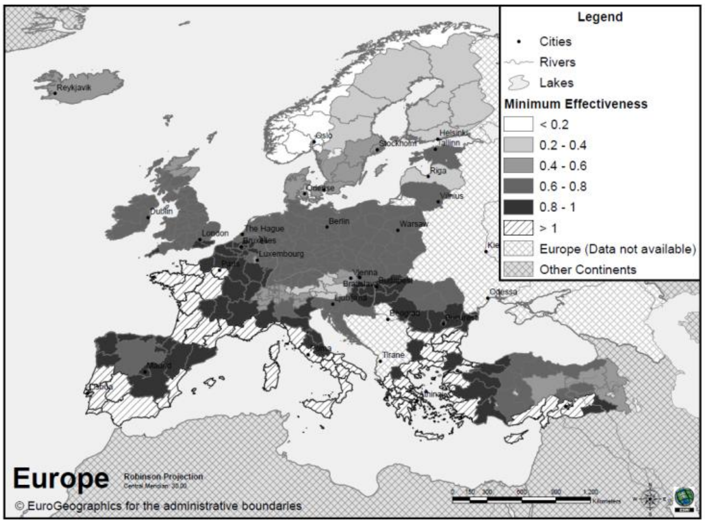

Figure 5.4 Minimal HRU effectiveness for break-even operation with Δ SFP 2500 J/m³ calculated with national primary energy conversion factor

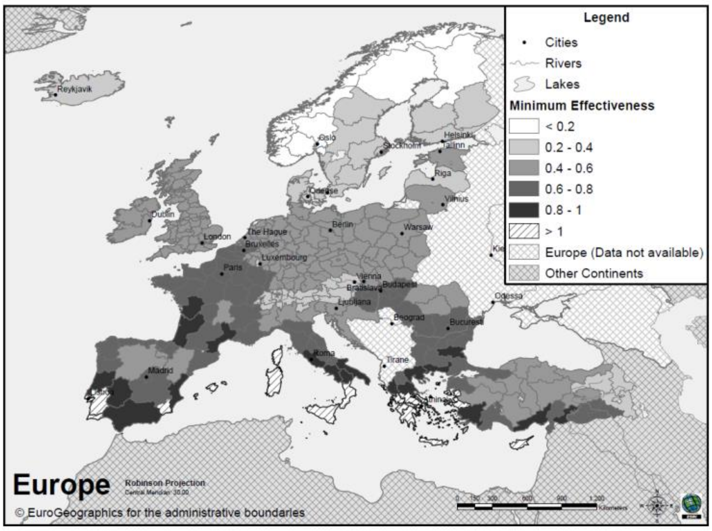

Figure 5.5 Minimal HRU effectiveness for break-even operation with Δ SFP 1750 J/m³ calculated with national primary energy conversion factor

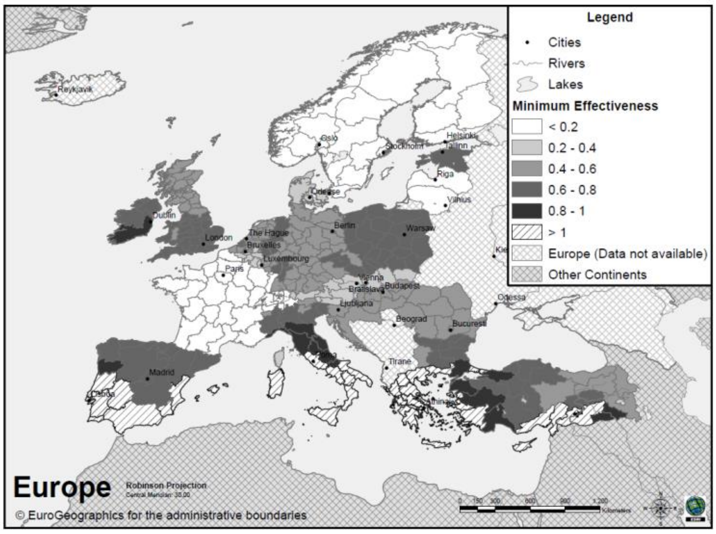

Figure 5.6 Minimal HRU effectiveness for break-even operation with Δ SFP 2500 J/m^3
calculated with CO_2 *conversion factor*

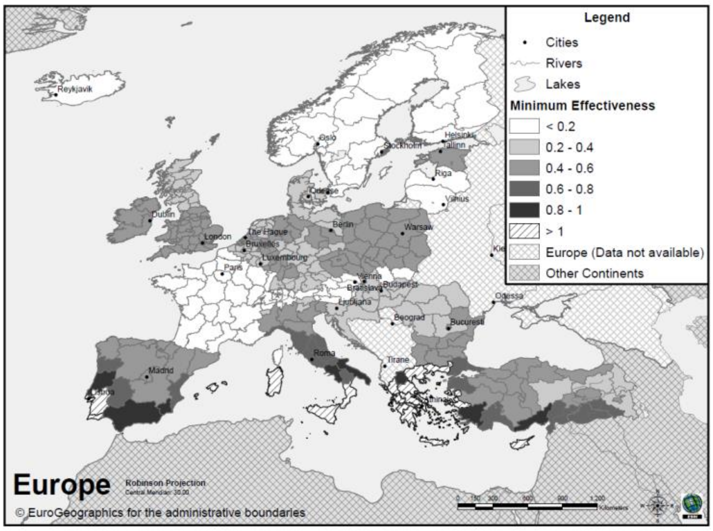

Figure 5.7 Minimal HRU effectiveness for break-even operation with Δ SFP 1750 J/m^3
calculated with CO_2 *conversion factor*

When only the emission of CO_2 is considered (Figure 5.6), one is immediately struck by the huge difference seen in France. This is readily explained by the large fraction of nuclear power present in this country, as was explained above. This feature is so dominant that it overshadows all climatic influence and requires a seasonal HRU effectiveness no higher than 0.2 to allow for break-even operation of heat recovery ventilation, putting France in the top of the EU for CO_2 emission reduction potential using heat recovery ventilation along with the Scandinavian countries. This same effect is, to greater or lesser extent, seen in every country that uses nuclear power in its electric energy mix or relies heavily on renewables, depending on the weight of these fuels in the mix. Non-nuclear countries mostly see only a small change in heat recovery ventilation feasibility, although renewable energy sources for electricity production evidently have a positive impact. Since few countries around the Mediterranean basin have large nuclear power capacities, heat recovery ventilation operation in this region is not strongly promoted by the carbon dioxide metric.

Although the available data for energy prices (Figure 5.8) was not as elaborate as the other databases, gas prices are high compared to electricity prices in North-West Europe, allowing for economic benefits from running heat recovery ventilation from a seasonal HRU effectiveness as low as 50%. In the UK, Romania, Spain, Croatia and Turkey, however, lower gas prices favour natural and mechanical exhaust ventilation. In the last three of these, this effect is of course further emphasized by the low number of HDD.

Quite different from the results seen so far, with primary energy, carbon dioxide and price conversion factors for electric energy, is the situation when the exergy conversion factor is used. In contrast to the other conversion factors, this factor rightfully rates thermal energy at room temperature as a very low grade, virtually useless energy quality. When this is considered, heat recovery ventilation is not feasible anywhere in Europe (Figure 5.10).

The equivalent effectiveness of both air-to-air heat recovery units and exhaust air heat pumps for domestic hot water production, taking into account an increase in fan energy of 2500 J/m^3, primary energy and a thermal effectiveness of 80% for the HRU, an air change rate of 0.5 ACH, a heated volume of 350 m^3, a domestic hot water demand of 79 l/day or 1934 kWh per year, seasonal performance factor of 2.96 and 85% efficiency for a gas fired domestic hot water boiler [438], is shown in Figure 5.12 and Figure 5.13, respectively.

Air-to-air heat recovery is only efficient with equivalent effectiveness of 40% and more in the northern part of Europe. Exhaust air heat pumps for domestic hot water production are less effective in this region. In contrast to HRU's, their performance is higher in the Mediterranean basin, although it is still very low (equivalent effectiveness < 0.4).

5.4 Discussion

In the previous sections, the trade-off has been analysed assuming fixed values for the different system characteristics. This section will focus on the sensitivity of the results with regard to realistic variations in these assumptions.

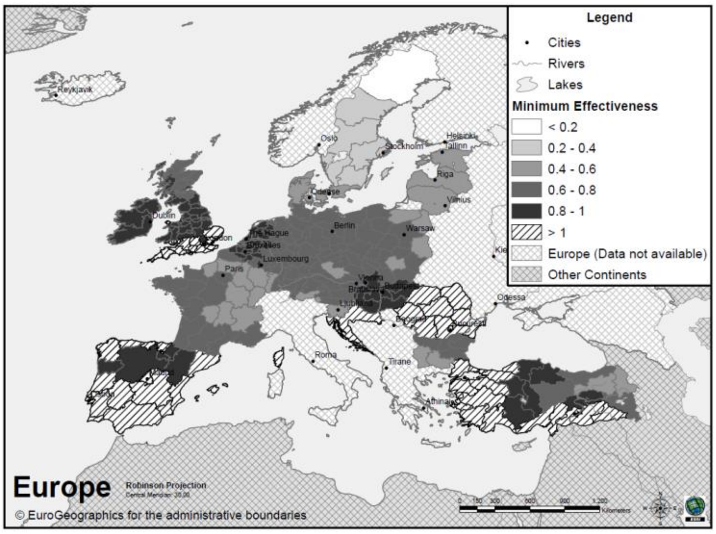

Figure 5.8 Minimal HRU effectiveness for break-even operation with Δ SFP 2500 J/m³ calculated with price conversion factor

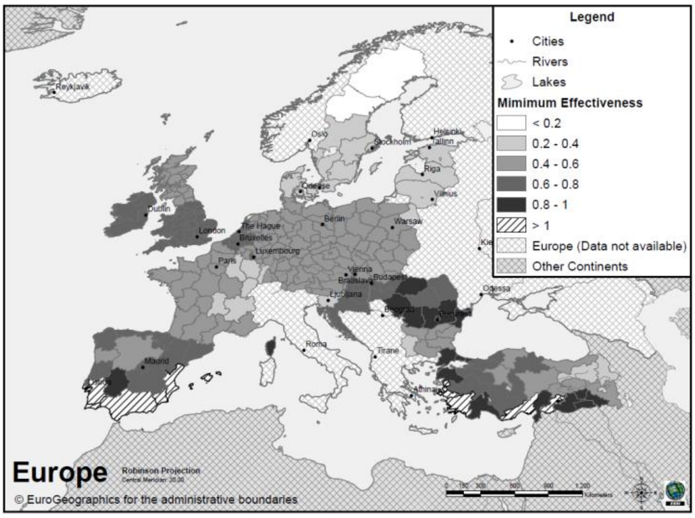

Figure 5.9 Minimal HRU effectiveness for break-even operation with Δ SFP 1750 J/m³ calculated with price conversion factor

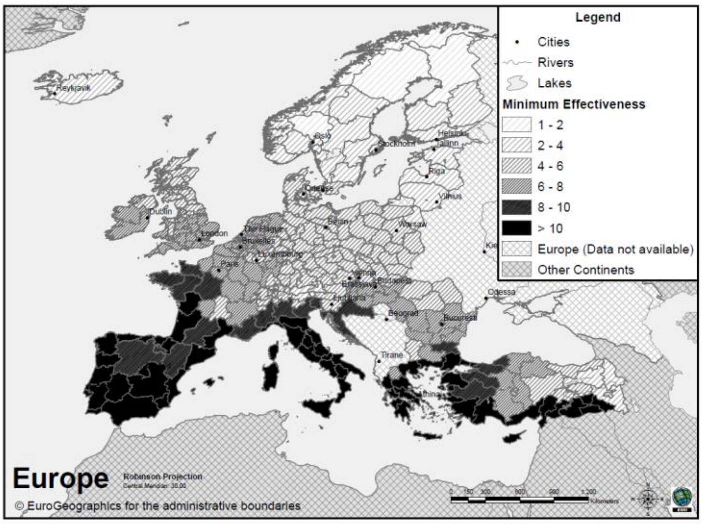

Figure 5.10 Minimal HRU effectiveness for break-even operation with Δ SFP 2500 J/m^3 calculated with exergy conversion factor

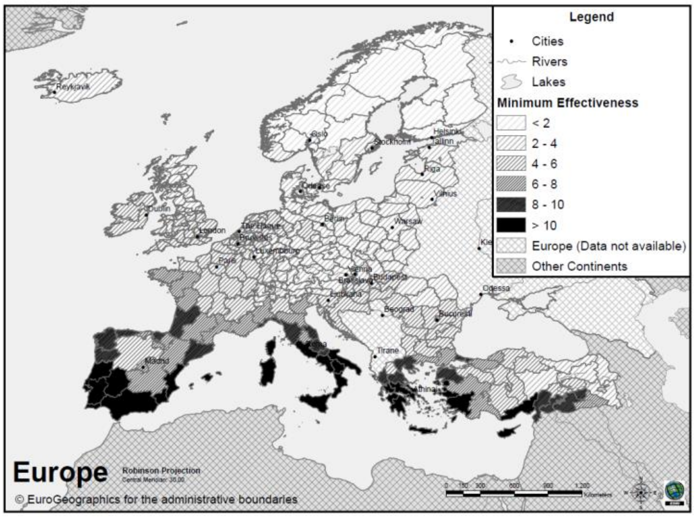

Figure 5.11 Minimal HRU effectiveness for break-even operation with Δ SFP 1750 J/m^3 calculated with exergy conversion factor

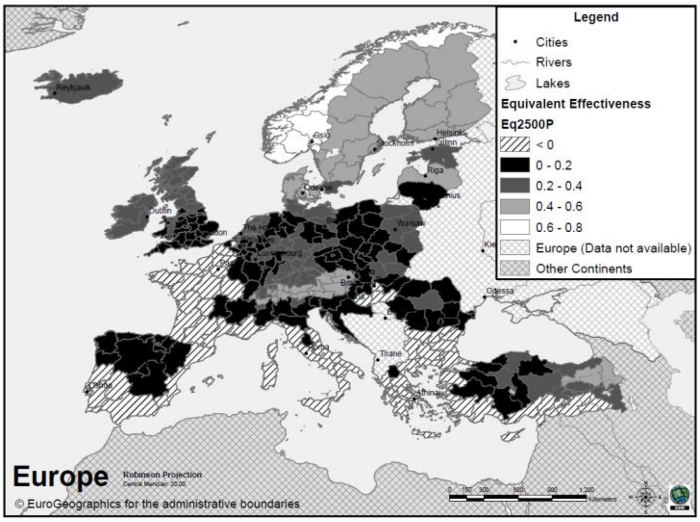

Figure 5.12 Equivalent effectiveness of HRU in Europe with ΔSFP 2500 J/m^3, ε *= 0.8 and* η_{DHW} *= 0.85*

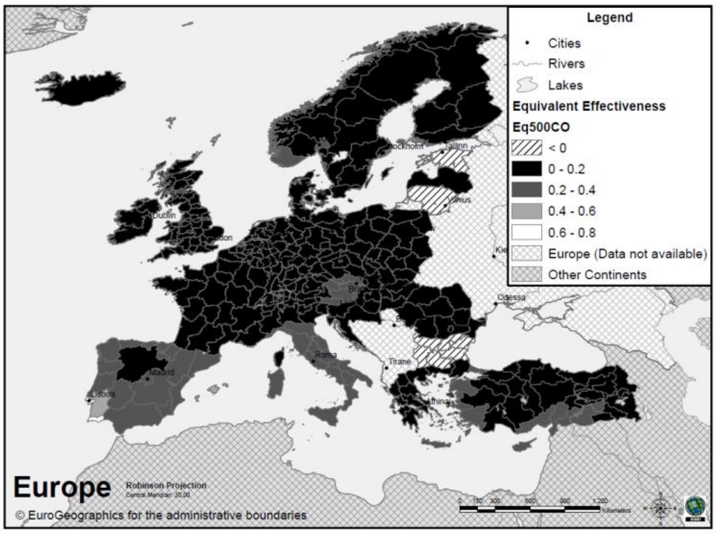

Figure 5.13 Equivalent effectiveness of EAHP in Europe with 0.5 ACH, SPF = 2.96 and η_{DHW} *= 0.85*

5.4.1 Methods and Basic Parameters

The presented method employs constant values for HRU effectiveness, pressure drop, fan power and conversion factors. All of these are subject to variations. HRU effectiveness is affected by temperature and humidity conditions as well as by the flow rate through the unit, while pressure drop and fan power are equally affected by dust accumulation in the ducting system and HRU, wear of the ventilator and of course by the flow rate. Additionally, frost and subsequently required defrosting cycles can have a considerable impact on the seasonal effectiveness.

The conversion factors fluctuate over time due to variations in peak load or partial load conditions of the electricity grid resulting in the subsequent change of the energy mix, due to energy price speculation etc. The seasonal performance factor of a system should be calculated by dividing the heating season integrated heat production by the heating season integrated electrical load. The use of average temperatures to calculate the Carnot efficiency therefore introduces an error. Based on hourly Meteonorm [439] weather data for the Uccle, Barcelona and Tampere climate, this error proved to overestimate the conversion factor by 13, 23 and 29% respectively. Nevertheless, correcting for this error does not drastically change the results. Concerning the primary energy and carbon dioxide emission coefficients, only the losses stated in the IEA data were taken into account. These do not include distribution losses to the domestic customer. Taking these into account will increase the conversion coefficients by approximately 10% [440]. Alternatively, the central heating systems are also faced with distribution and other system losses amounting to 10-15%. The increasing integration of the internal market and the choice for renewables for electricity production mandated by the EU will cause the primary energy factor within Europe to converge and decrease, favouring heat recovery ventilation. The effect of a completely converged market at the current state of electricity production (uniform primary energy conversion factor at the EU average) on the minimal HRU effectiveness is shown in Figure 5.3.

5.4.2 Fan Operation

In the present analysis, the fans are, based on the default values from the EN 13779 standard [230], assumed to operate continuously at their design flow rate and to have a total SFP of 2500 J/m^3 for a heat recovery system with supply and exhaust fans or of 750 J/m^3 for a mechanical exhaust system. Although these values correspond roughly to values measured in situ for residential ventilation systems [441], the examples shown in Figure 5.14 show that they are subject to large variability and are often lower than these default values.

Additionally, occupants tend to run their systems at lower flow rates than the design flow rate. For example, Rosseel [424] reported 52% of families disposing of a three-position frequency control switch on their ventilation systems run it at the lowest flow rate (position 1). Van Dijken [425] reported even higher incidence rates of running the system primarily in the lowest flow rate, of 80% and more. In this survey however, the medium flow rate (position 2) corresponded to the design flow rate, so the results are quite similar. These specific operating conditions as well as the implementation of demand control ventilation equipped with frequency controlled fans or low pressure drop bypass dampers reduce the average pressure drop and associated fan power considerably. As an example, the effect of three-position switch operation on specific fan power from in situ measurements [441] from residential ventilation systems are shown in Figure 5.15. When systems are run at the lowest instead of the design flow rate, the SFP is on average reduced by 55%. Based on this, a clear potential in ventilation flow reduction during summer can easily be demonstrated. Note that reduced fan efficiency at low flow rates can cause the specific fan power to increase in some cases, demonstrated by the high upper quartile flag of the boxplot in position 1.

A further note to be made in the case of mechanical exhaust ventilation directly relates to the sizing of trickle ventilators: conventional trickle sizing is thus that the total intended ventilation flow rate – the total outdoor airflow rate minus the infiltration – is far larger than the mechanically exhausted rate. The additional fraction is caused by adventitious ventilation through the trickle ventilators and should be taken into account when calculating the SFP of the system, thus reducing the latter. Based on numerical simulations with a monte carlo approach [347], the ratio of the heat loss through the mechanically exhausted flow rate to that of the total intended ventilation flow rate varies from 0.3 to 1 (Figure 5.16) for different approaches to mechanical exhaust ventilation systems, with and without demand control. As is to be expected, in the demand controlled systems where the mechanical flow rate is reduced in function of the demand (Cpres reducing exhaust flow to the minimum rate except in case of presence, Crh reducing exhaust flow rate if relative humidity is low), this ratio, with a median of 0.5, is smaller than in the mechanical exhaust system (C), while it is higher in the system where motorized trickle ventilators are used to control the air flow without changing the fan speed, with a median of 0.95. For the demand controlled systems of the first type (Cpres, Crh), taking this effect into account will reduce the SFP by 50%, creating more favourable operating conditions for this kind of system compared to heat recovery ventilation where this kind of effect is not possible.

Due to differences in the quality of implementation, specific operating conditions, control and trickle ventilator sizing, the additional electricity consumption of a heat recovery system, ΔSFP, could be considerably different from the original assumptions. The large effect of different ΔSFP on the minimal effectiveness of the heat recovery unit that allows break-even operation of heat recovery ventilation, for the capitals of the three countries chosen as examples in Table 5.1, is shown in Table 5.2, using the primary energy conversion factor.

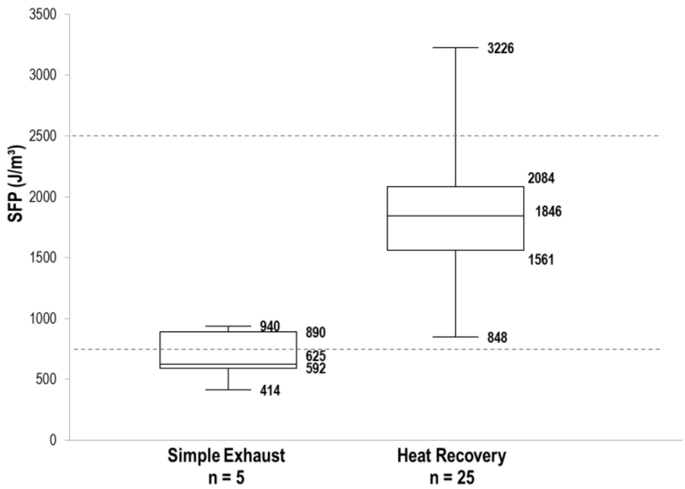

Figure 5.14 Measured SFP in mechanical exhaust and heat recovery residential ventilation systems, with default values in dashed lines

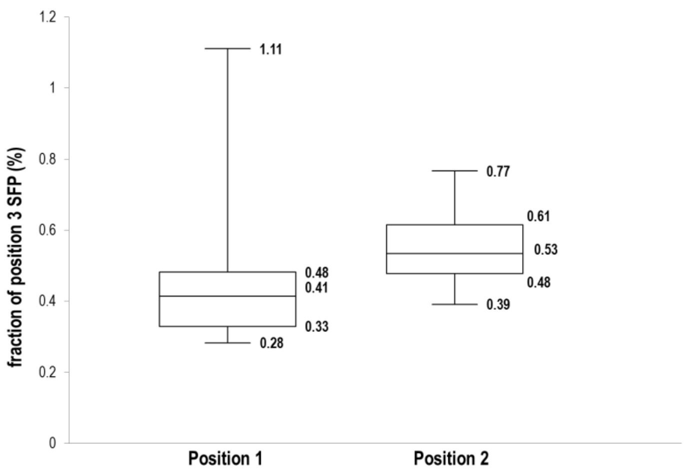

Figure 5.15 Measured SFP in function of control switch position in residential ventilation systems, relative to position 3 SFP (n = 22) [441]

It is clear that attention to pressure drop during design and execution is of vital importance for net positive operation of a heat recovery system. Reducing specific fan power by 50 to 75% in reference to the default values given in the standard, which is ambitious, but, as is shown in Figure 5.14, possible, renders heat recovery ventilation profitable in virtually the whole of Europe.

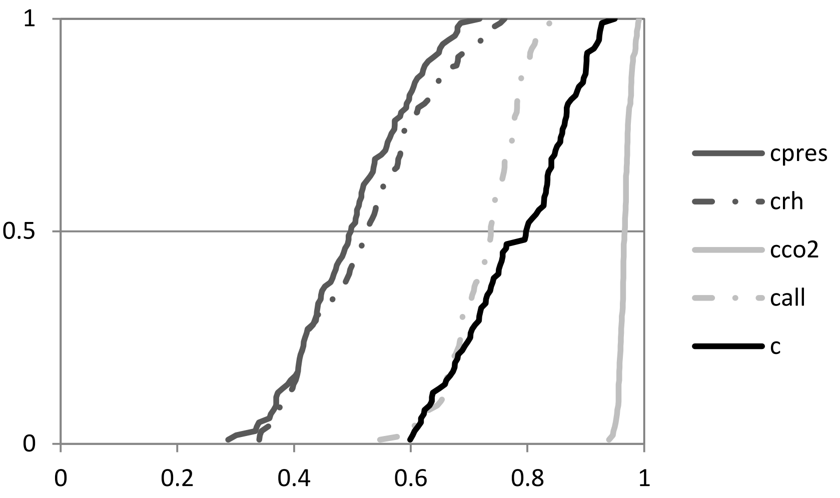

Figure 5.16 Cumulative distribution ratio of heat loss of mechanically exhausted flow rate to that of total intended ventilation flow rate for 1 simple (C) and 4 demand controlled (Crh, Cco_2, Cpres and Call) mechanical exhaust ventilation systems from Monte Carlo based numerical simulations (Chapter 4)

The performance of heat recovery ventilation is thus shown to be highly dependent on the efficiency of the fan. The equivalent effectiveness calculated above takes into account fans with performance levels at the mid-range of the recommended default values by the European standard (1250 J/m^3 per fan). The measurements presented show that the performance in practice can range over a broad spectrum. Improving fan performance with one class (750 J/m^3 per fan), to the top of the CEN default values, extends their profitability to the South, with Germany, Poland, Belgium and The Netherlands as an interesting region for their application (Figure 5.12).

Table 5.2 Minimal effectiveness of the heat recovery unit that allows break even operation of heat recovery ventilation, for different SFP-levels, for the capitals of the 3 countries chosen as examples in Table 5.1 using the primary energy conversion factor

ΔSFP (J/m^3) (% of original)	625 (25%)	1250 (50%)	1875 (75%)	2500 (100%)	3125 (125%)
Paris	0.28	0.56	0.84	1.12	1.40
Oslo	0.05	0.10	0.15	0.20	0.25
Madrid	0.25	0.50	0.75	0.99	1.24

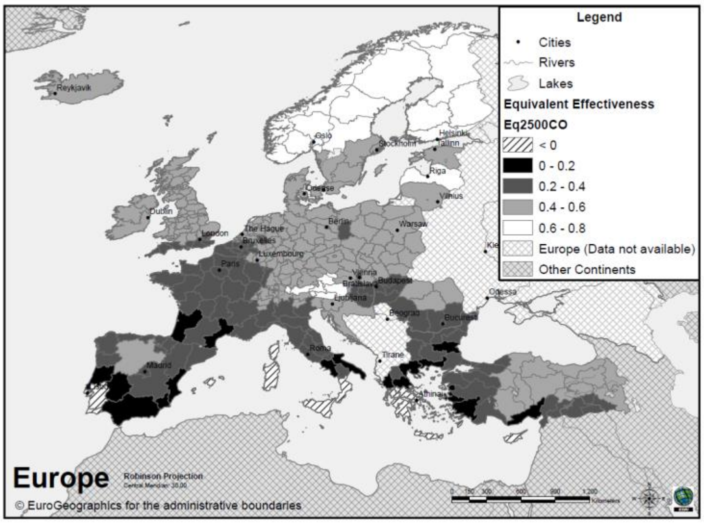

Figure 5.17 Equivalent effectiveness of HRU in Europe with ΔSFP 1500 J/m^3, ε = 0.8 and η = 0.85

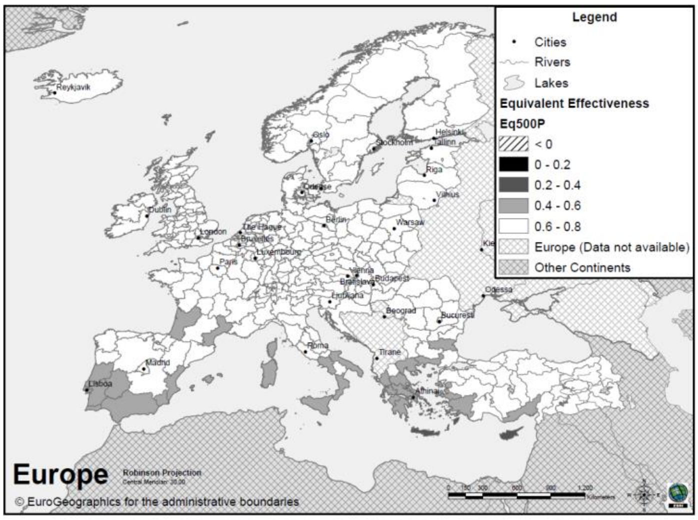

Figure 5.18 Equivalent effectiveness of HRU in Europe with ΔSFP 500 J/m^3, ε = 0.8 and η = 0.85

When a difference in fan power of only 500 J/m^3 is considered (Figure 5.18), which would correspond to the addition of a low pressure drop HRU in a mechanical supply and exhaust system, the application of air to air heat exchanger technology is vastly beneficial anywhere in Europe. Mind that this scenario is only valid when mechanical supply and exhaust is necessary anyway and with high end HRU's that combine a high thermal effectiveness ($\varepsilon = 0.8$) and low pressure drop.

5.4.3 Climate

The results shown above were obtained using the Eurostat Heating Degree Day data, as explained in the methods section. Although heating degree days correlate reasonably well with the heat loss of a heated building [442], the fact that a unique reference temperature is used does not take into account the thermal quality of the building envelope, nor its airtightness and the effect of both on the utilization of internal and solar gains.

Furthermore, the presented results only take into account heating demand. Although the potential for heat(/cold) recovery is rather small due to the small temperature difference typical for cooling regimes in Europe, neglecting it underestimates the potential of heat recovery ventilation in the most Southern regions.

The effect of using a different base temperature for the calculation of heating degree days and the inclusion of cooling degree days on the minimal effectiveness of the heat recovery unit that allows break-even operation of heat recovery ventilation, for the capitals of the three countries chosen as examples in Table 5.1, is shown in Table 5.3, using the primary energy conversion factor. The heating degree days for different base temperatures and the cooling degree days were calculated from the raw data of the Eurostat heating degree day calculation, daily minimal and maximal temperature data for the same 2000-2009 period [443]. The 12.5 and 10 °C base temperatures were chosen to represent high (low energy house) and extreme (passive house) thermal quality of the building envelope respectively. The cooling degree days were calculated with a 22°C base temperature.

Table 5.3 Minimal effectiveness of the heat recovery unit that allows break even operation of heat recovery ventilation, for the capitals of the 3 countries chosen as examples in Table 5.1, with standard (15°C), high (12.5 °C) and extreme (15°C) thermal quality of the building envelope, with and without cooling degree days (CDD) taken into account, using the primary energy conversion factor

Base temperature (°C)	15		12.5		10	
CDD included?	no	yes	no	yes	no	yes
Paris	1.12	1.10	1.60	1.55	2.46	2.33
Oslo	0.20	0.20	0.24	0.24	0.31	0.31
Madrid	0.99	0.84	1.46	1.15	2.41	1.66

Due to the typically low number of cooling degree days, they tend to have a limited impact on the minimal HRU effectiveness while the latter is within realistic values. Note that high thermal envelope quality has a large influence on the feasibility of heat recovery ventilation in moderate climate regions like France, in contrast with popular high efficiency building concepts.

A similar effect on the minimal HRU effectiveness can be seen with the implementation of demand controlled ventilation. Demand controlled ventilation will reduce the heat loss associated with the maintenance of a certain level of indoor air quality, e.g. the systems shown in Figure 5.16. reduce the heat loss by 30 – 60 % [347]. The change in minimal HRU effectiveness that this entails will depend on the decrease in average SFP that is associated with the demand control.

5.5 Conclusions

The assessment of the profitability considering operating energy of heat recovery ventilation as opposed to mechanical exhaust or natural ventilation in Europe is strongly dependent on the type of conversion coefficient between electrical energy and fuel combustion for heating that is used. With primary energy, carbon dioxide emission, price and exergy, Four broadly accepted frameworks for the comparison of these types of energy were used to calculate conversion factors for the different countries of the EU. The obtained factors showed to be sensitive to the specific energy mix and the climatic conditions of a region. This is clearly demonstrated by the cases of France and Norway, where the highest and lowest conversion factors have a ratio of 150 and 500 respectively. This huge range is attributable to nuclear respectively renewable power production on the one hand, drastically reducing the carbon dioxide factor, and the relatively small average indoor/outdoor temperature difference, resulting in high conversion factors in the exergy analysis. The results presented here demonstrate that, unless low specific fan power is achieved, for the moderate climate region of middle Europe, natural ventilation, mechanical exhaust ventilation and heat recovery ventilation have no clear advantage over each other as far as operating energy and associated ecologic (CO_2) and economic (Household consumer price) effects are concerned. The choice between the different systems should be made based on building specific characteristics, investment and maintenance cost. In the Mediterranean basin, heat recovery ventilation can only be operated profitably in low pressure drop and low fan power systems, while it is advantageous under virtually all tested conditions in the Scandinavian region. In contrast to low fan power, high thermal building performance tends to create unfavourable conditions for heat recovery ventilation. Overall, heat recovery ventilation can be made profitable all over Europe with regard to primary energy, carbon dioxide emissions and household consumer operating energy cost by achieving realistic best practice low specific fan power.

6

Residential IAQ: Sleep

This final chapter of the dissertation revisits the design objectives put forward in the introduction by highlighting the dominance of the exposure in the bedrooms in the assessment of residential indoor air quality as well as the poor validity of available performance indicators and assessment methods in this environment. The presented results from experimental campaigns studying the physiological response of sleeping subjects to changes in indoor air quality and the impact of the sleep micro-environment on exposure clearly demonstrate a need for further research in this area to achieve a comprehensive assessment methodology for residential ventilation systems. The results of the last section have been published in Building and Environment [262].

6.1 Bedroom IAQ

On average, a person spends about one third of his or her life asleep. In the US, the average daily time spent asleep for persons over 15 years old reported in time use studies is 8.7 hours [358]. Similar results are found in other time use surveys, e.g. in Belgium 9 hours and 3 minutes on average for persons over 12 years old [357]. Expanding on the Belgian study, taking into account that, on average, the same population spends 16 hours and 55 minutes per day at home and assuming that all sleep is done at home, this amounts to about 54 % of the total time spent in dwellings. This increases to 58% for people with a full-time job, making the bedroom the most likely place for exposure to indoor air pollutants. This likelihood is emphasised by the fact that bedrooms are often poorly ventilated [444].

While exposure to pollutants emitted by building material and furnishing can and should effectively be mitigated by source control measures and the selection of adequate materials [81, 182], exposure in the bedroom to activity based pollution can, however, still represent a large fraction of the total exposure to these pollutants due to the long and continuous presence in the bedroom and to redistribution of pollutants emitted in other spaces throughout the dwelling [eg. 129]. Local exhaust and, again, source control provide the best strategy for the mitigation of activity based pollutants.

Pollutants emitted by the occupants themselves, such as moisture from exhalation and perspiration, exhaled carbon dioxide as a by-product of the metabolism, particulates from skin shedding and a large number of chemical compounds both directly produced by or resulting from secondary chemical reactions with the body [9]are usually grouped under the denominator ‘human bio-effluents’. Increased concentrations of these pollutants have been shown to negatively correlate with perceived indoor air quality [67].

The strength of the emission of these pollutants is generally related to the level of activity of the person, due to the positive association of activity level and metabolic work [218, 398]. Notwithstanding that considerable inter-person variability in emission rates exists [259], carbon dioxide is widely recognised as a good tracer for human bio-effluents as a whole due to its chemical stability, relatively low number of competitive sources and established sensor technology.

Most of the available ventilation standards today are based on the perceived air quality theorem presented by Fanger [67]. The test data on which this theorem is based documents the perception of the test subjects upon entering a polluted room. In these tests, a rather straight forward relation between the pollution load (expressed in olf) and the appreciation of the air quality by the test panel was found. Based on these findings and the evidence that carbon dioxide (CO_2) is a good indicator for metabolic pollution loads in an environment with no open combustion, a correlation between desired indoor air quality and CO_2 was established, eg. in European ventilation standards [230].

In a second stage, the influence of the perceived air quality on office work performance [62, 63], absenteeism or learning [58, 59, 61, 445] was investigated, establishing a correlation that can be used to trade-off the benefits of higher ventilation rates and the energy and investment cost related to it [66]. Finally, an large consolidation effort demonstrated that the incidence of negative health effects correlates inversely to the ventilation flow rate up to flow rates of 25 l/s per person [36, 69]. Although the human body has been expected to be indifferent to exposure to carbon dioxide at concentration levels normally found indoors (< 1%), recent studies suggest that carbon dioxide in itself can also cause physiological reactions, reducing the decision making ability of test subjects [446, 447]. All of these experiments and monitoring projects, however, relate to an active environment.

As was mentioned above, some exceptions aside, we spend about 50 to 60 % of the time we spend at home asleep [357, 358]. During this time, we are in a semi-conscious state and are not engaged in tasks of which the performance can be measured by output. Therefore, the performance indicators referred to above are only valid in a limited part of the net time-use as far as residential ventilation is concerned. Nevertheless, the exposure conditions in the bedroom are fundamentally different in the bedroom than in the other rooms, mainly due to the fact that in a large majority of the time spent in the bedroom, the occupants are asleep. Therefore, traditional IAQ ratings based on perceived air quality are not applicable.

In contrast to other activities, there is relatively little variance in the amount of sleep per day and, what is more important, the largest part of the daily sleep time is consumed in a single event. The metabolic rate and therefore the source strength of bio-effluents is low and constant during sleep [259], although in the master bedroom, the occupancy is fairly dense compared to that of other spaces in the house. Due to the long exposure time, exposure in the bedroom is close to steady state and differs significantly from that in other spaces of the dwelling, where fairly short activities in different spaces succeed one another and concentrations drop to the background level with each change in space. Additionally, since the occupants are asleep during the largest part of their exposure in the bedroom, they are not able to act upon reaching undesirably poor indoor air quality, e.g. by opening windows. The long term proximity of occupants to a range of sources located in or near the bed creates specific exposure conditions that require higher ventilation rates, while perceived air quality based indoor air quality levels may be less applicable and lower health based requirements [51] more appropriate.

One of the most used strategies to reduce heating demand in low energy buildings is reducing the leakage level of the building envelope. Dedicated ventilation systems are then installed to compensate for the reduced air change rate in an energy efficient way. Although there is some literature on bedroom indoor air quality [202, 249, 444], most research focusses on other spaces in the dwelling such as living room, bathroom and kitchen [121, 128, 199, 200] or on the thermal environment in the bedroom [448, 449]. Since the bedroom is the most intensively used room of a dwelling, considering the time spent in the room, with long continuous exposure time and little opportunity for the occupant to adapt the indoor environment during exposure, residential ventilation and indoor air quality research should focus more on its characteristics and how they differ from those in other spaces. This chapter first reviews the results of simulations presented in the previous chapters looking specifically at exposure in the bedrooms. It then presents the results from carbon dioxide monitoring efforts in the bedroom and living room of 47 standard and 34 low energy dwellings in Belgium and compares the typical exposure profiles for both spaces, verifying the observations made based on the simulation results. In the last two sections of the chapter, the validity of perception based criteria for bedrooms is tested in an intervention study on students and lab experiments with a breathing thermal manikin.

6.2 Bedroom IAQ in Simulations

In this section, the results of the simulations of the first 3 demand control strategies presented in the first section of chapter 4 are analysed with regard to their distribution within the dwelling, focussing on the differences between the bedrooms and the rest of the dwelling.

Excess CO_2 concentration will be used as the assessment parameter for the indoor air quality. As was discussed above, the usability of this parameter for a dormant subject is debatable, but due to lack of a better alternative, excess CO_2 at least provides an indication of the exposure of the subjects to bio-effluents. It can therefore be considered an indicator for the risk of contamination associated with the breathing. Since the results are based on stochastic modelling and are therefore representative for a large population of cases, the parameter is called 'expected' exposure.

The aim is to compare the expected exposure in the bedrooms with the overall expected exposure and that in the other spaces of the dwelling. Therefore, two additional assessment parameters are introduced, namely the fraction of the total exposure within a certain IAQ-class that takes place in the bedrooms and the ratio of the relative exposure to a certain IAQ-class (defined as the occurrence of exposure within the class as a fraction of the total time within the room type) in the bedrooms to that in either the entire dwelling and the other spaces in the dwelling. The IAQ-classes referred to are the IDA classes specified in the EN 13779 [230].

Figure 6.1 - Figure 6.4 show the cumulative distribution of the expected exposure of the occupants to CO_2 in all of the 4 simulated systems for both the total time spent in the dwelling and the time spent in the bedrooms based on the simulations. You can clearly see that the exposure in the bedrooms is higher than the overall expected exposure, except for the case where the trickle ventilators are manipulated. Remember that the outdoor concentration is set to 350 ppm.

The individual exposure profiles show that the typical exposure to excess CO_2 concentrations in the bedroom is high over a long period in the second half of the night. The results for the other two assessment parameters are shown in Table 6.3 and Table 6.4, while Table 6.1 gives an overview occurrence of the exposure to an air quality within the respective IAQ-classes during the time spent in the dwelling. As was already apparent in Figure 6.4, the exposure to IAQ generally considered to be unacceptable (IDA 4) is very limited for the system where the trickle ventilators are controlled. This is mainly caused by the fact that by manipulating the trickle ventilators, the buoyancy driven component in the flow dynamics in the building is reduced and the underpressure created by the mechanical exhaust is increased, considerably increasing the proportional influence of the forced flow and reducing the sensitivity to environmental parameters. Likewise, Tables 6.2 to 6.4 demonstrate that this also heavily influences the relative exposure to these high concentrations in the bedrooms. This is explained by the fact that closing the trickle ventilators in the rest of the dwelling at night (due to low CO_2 concentrations in these spaces) will force more air to enter through the bedrooms.

In contrast to these rather positive results in the case with trickle ventilator control, the results for the other cases show that the exposure to bad perceived air quality (IDA 4) is dramatically higher in the bedrooms than in the other parts of the dwelling. Nevertheless, the differences in relative exposure to good perceived air quality (IDA 1 and 2), taking up about 80% of the total exposure time, are less pronounced.

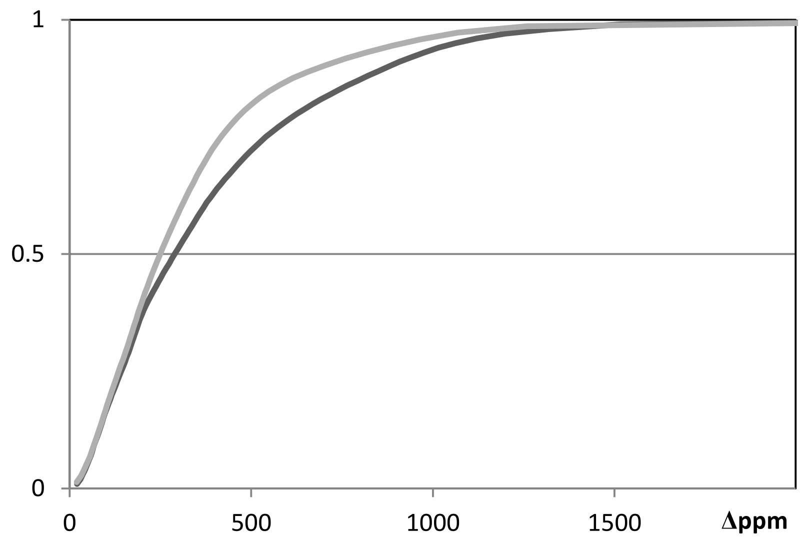

Figure 6.1 Cumulative distribution of the exposure to CO_2 (ppm) in the total dwelling (grey) and in the bedroom (black) in the case with continuous flow mechanical exhaust ventilation

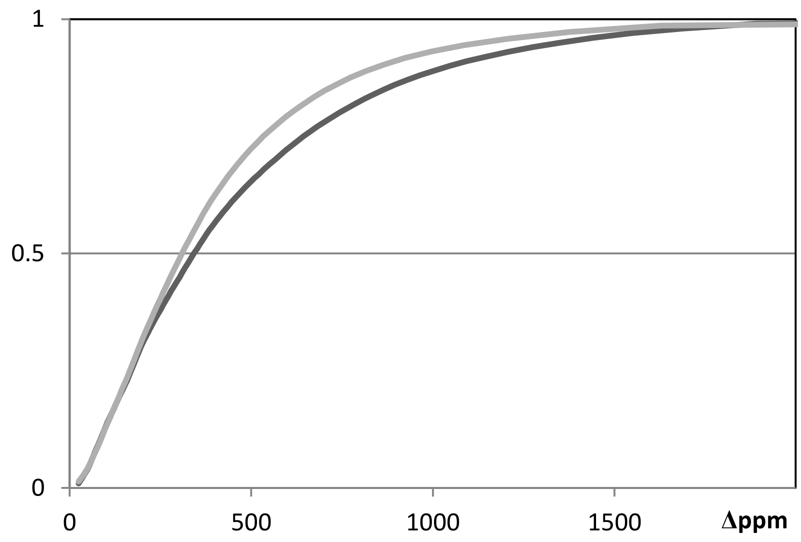

Figure 6.2 Cumulative distribution of the exposure to CO_2 (ppm) in the total dwelling (grey) and in the bedroom (black) in the case with humidity controlled mechanical exhaust ventilation

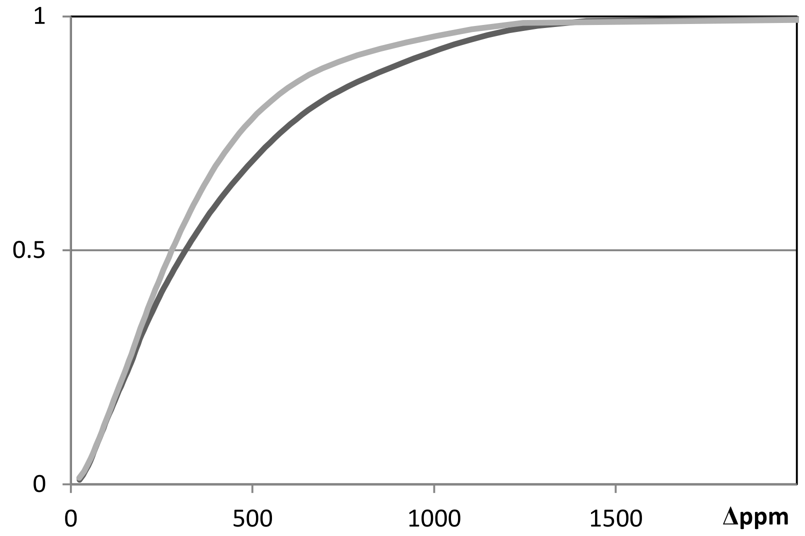

Figure 6.3 Cumulative distribution of the exposure to CO_2 (ppm) in the total dwelling (grey) and in the bedroom (black) in the case with presence controlled mechanical exhaust ventilation

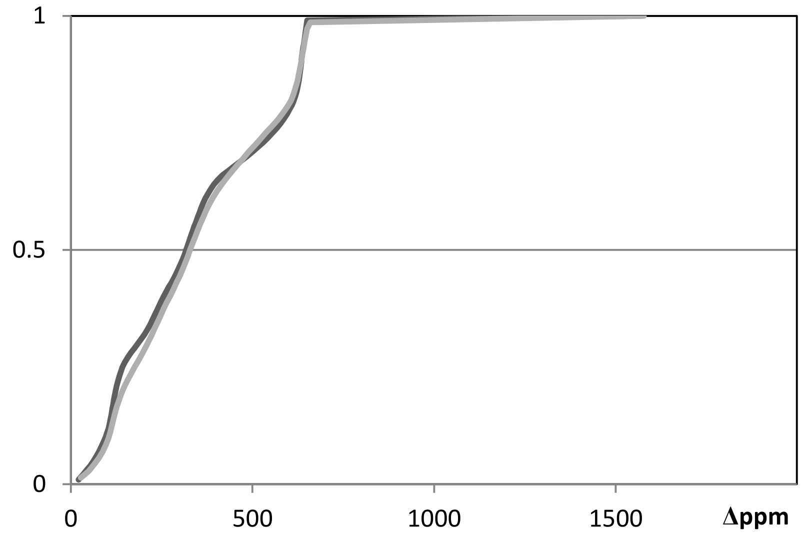

Figure 6.4 Cumulative distribution of the exposure to CO_2 (ppm) in the total dwelling (grey) and in the bedroom (black) in the case with supply trickle controlled mechanical exhaust ventilation

Table 6.1 Occurrence of the exposure to an air quality within the respective IAQ-classes for 334 occupants under the 4 ventilation systems

IAQ-class	Continuous	RH	Presence	Trickle
IDA 1 (0-400 ppm)	0.73	0.62	0.68	0.62
IDA 2 (400-600 ppm)	0.14	0.17	0.17	0.19
IDA 3 (600-1000 ppm)	0.09	0.14	0.11	0.19
IDA 4 (> 1000 ppm)	0.04	0.07	0.04	0.00

Table 6.2 Fraction of the total exposure to an air quality within the respective IAQ-classes taking place within the bedrooms for 334 occupants under the 4 ventilation systems

IAQ-class	Continuous	RH	Presence	Trickle
IDA 1 (0-400 ppm)	0.47	0.49	0.47	0.54
IDA 2 (400-600 ppm)	0.60	0.5	0.54	0.42
IDA 3 (600-1000 ppm)	0.87	0.65	0.78	0.56
IDA 4 (> 1000 ppm)	0.95	0.87	0.94	0.07

Table 6.3 Ratio of the relative exposure to an air quality within the respective IAQ-classes within the bedrooms to that within the entire dwelling for 334 occupants under the 4 ventilation systems

IAQ-class	Continuous	RH	Presence	Trickle
IDA 1 (0-400 ppm)	0.87	0.91	0.88	1.04
IDA 2 (400-600 ppm)	1.1	0.93	1.01	0.8
IDA 3 (600-1000 ppm)	1.6	1.2	1.45	1.08
IDA 4 (> 1000 ppm)	1.75	1.61	1.74	0.13

Table 6.4 Ratio of the relative exposure to an air quality within the respective IAQ-classes within the bedrooms to that within the rest of the dwelling for 334 occupants under the 4 ventilation systems

IAQ-class	Continuous	RH	Presence	Trickle
IDA 1 (0-400 ppm)	0.75	0.82	0.77	1.09
IDA 2 (400-600 ppm)	1.25	0.85	1.02	0.65
IDA 3 (600-1000 ppm)	5.52	1.58	3.03	1.18
IDA 4 (> 1000 ppm)	16.13	5.59	12.54	0.06

6.3 Case Study

To verify the simulation based observations made in the previous section, the concentrations measured in living rooms and bedrooms during monitoring campaigns in Flemish residences executed by our research group over the last 5 years are aggregated and analysed in this section. It first discusses the main characteristics of the cases included in the study and includes a brief overview of the measurement techniques used. After that, the typical exposure profiles found in living room and bedrooms are presented.

6.3.1 Measurements

In each of the dwellings, the carbon dioxide concentration in the living room and the master bedroom was monitored using a non-dispersive infrared logger with an accuracy of ± 30 ppm ± 3 % for a median duration of 9 consecutive days. The leakage level of the envelope was measured in accordance with the pressurization method [288]. Typical occupancy and window opening behaviour of the occupants was collected with a questionnaire after completion of the monitoring, in order not to affect their behaviour during the monitoring period.

6.3.2 Cases

The sample of cases is composed of single family houses of varying age and energy performance. Eight cases are social housing units dating from the 60'ies, are not insulated, very leaky and have no dedicated ventilation system (group 1). They are representative for the majority of the housing stock in Belgium. The measured annual heating demand for this group is about 150 kWh/m^2. A second group of 35 cases consists of houses of about 5 years old with standard contemporary insulation levels, a heating demand of about 75 kWh/m^2 and mechanical exhaust ventilation systems (group 2). These dwellings are considerably less leaky. Most of the recently built dwellings in Belgium are very similar to the specifications of the dwellings in this group. The remaining 34 cases are low energy houses (group 3), sixteen of which are passive houses (group 4). These all have mechanical supply and exhaust systems with heat recovery units and are designed to heating demands of 30 and 15 kWh/m^2 respectively. The low energy houses have leakage levels slightly lower but comparable to those of the second group, while the passive houses are extremely airtight. The characteristics of these 4 groups are summarized in Table 6.5 and Figure 6.5 shows the distribution of envelope leakage in each subgroup of the set.

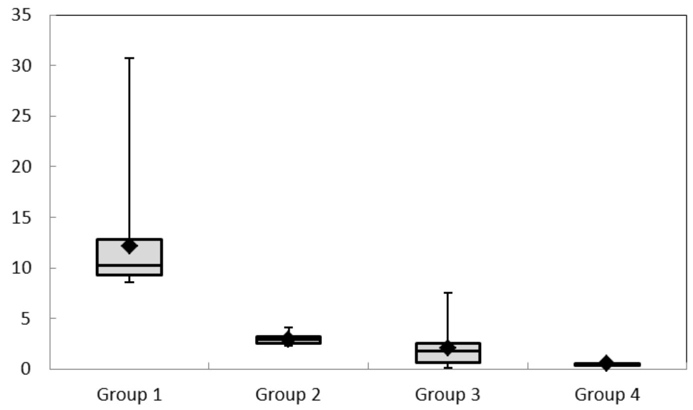

Figure 6.5 Measured leakage level of the building envelope (ACH_{50}) of the different groups in the sample

6.3.3 Occupancy and Window Airing

Since the dwellings in the first group were built by social housing programs, they are occupied by a generally older and economically less active population that, on average, spends considerably more time at home during the day than the occupants of the remaining 3 groups. The latter are mainly young families with full time jobs. Figure 6.6 and Figure 6.7 show the typical occupancy schedules for bedrooms (black) and living rooms (grey) for the first and the second group, along with the probability that at least one occupant is at home (grey line)The occupants were asked to report their average occupancy profile on an hourly basis.

The data shown is for the whole neighborhoods in which the cases of group 1 and 2 are situated [303]. Note that, although there is a large difference in occupancy in the living room during the day, the occupancy of the bedroom is similar in both groups.

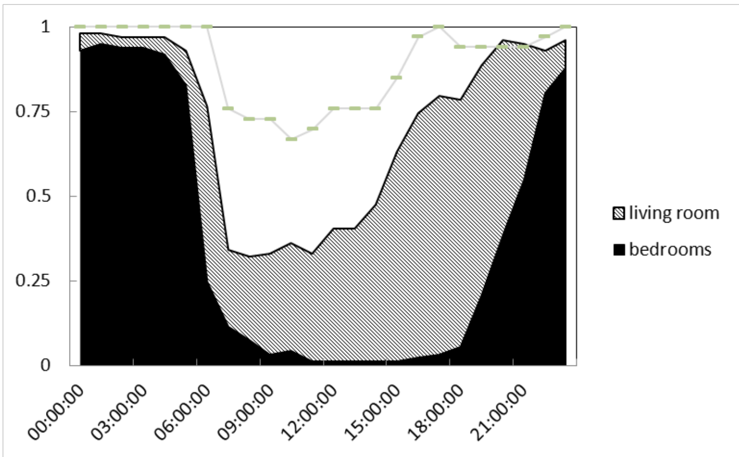

Figure 6.6 Cumulative probability of presence in bedrooms and living room for occupants in the neighbourhoods of the cases of group 1, along with the probability that at least 1 occupant is at home over the course of the day.

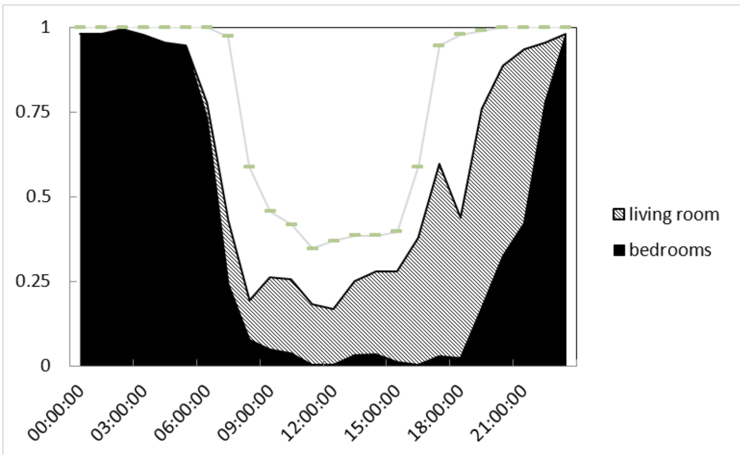

Figure 6.7 Cumulative probability of presence in bedrooms and living room for occupants in the neighbourhoods of the cases of group 2, along with the probability that at least 1 occupant is at home over the course of the day.

Table 6.5 Overview of cases included in the study

Group	Number of cases	Ventilation type	Average leakage level
1	8	none	12 ACH_{50}
2	39	exhaust	3 ACH_{50}
3	19	heat recovery	2 ACH_{50}
4	16	heat recovery	0.5 ACH_{50}

By comparing the responses to the questionnaire about window opening behaviour for the neighbourhoods of both groups 1 and 2, shown in Figure 6.9 and Figure 6.10, to the occupancy profiles shown above, it is obvious that the occupants tend to ventilate the spaces by window airing only during absence. Bedroom windows are scarcely opened during the night. Their impact on exposure to human bio-effluents in the bedroom is therefore negligible.

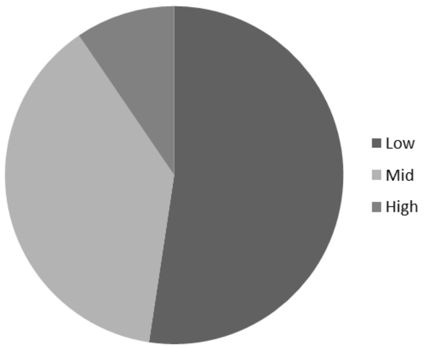

Figure 6.8 reported selected flow rate by mechanical supply and exhaust system owners for systems with a 3-position switch in a Belgian survey [424]

Next to making use of window airing, occupants select an appropriate flow rate for the ventilation system to control the indoor air quality within the dwelling . A survey on flow rate selection by mechanical supply and exhaust owners in Belgium, demonstrates that the majority permanently selects the lowest flow rate (roughly corresponding to 1/3 of the design flow rate) by default [424], as is shown in Figure 6.8**.** Similar results were reported in the occupancy questionnaires for the cases in groups 2-4. The measurements made on the fan in the case studies presented in chapter 4 revealed that little to no modulation of the default position occured over the measurement period. Additionally, a flow rate selection switch is usually only located in the kitchen of the dwelling, thereby further reducing the probability that a higher flow rate will be selected during the night.

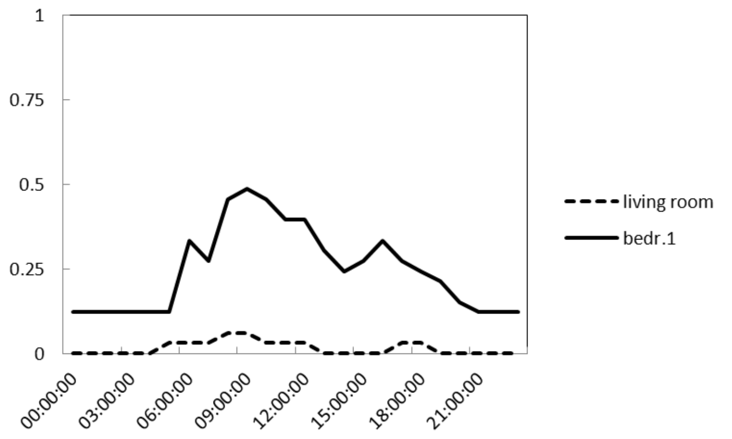

Figure 6.9 Average probability of window opening in bedrooms and living room in the neighbourhoods of the cases of group 1

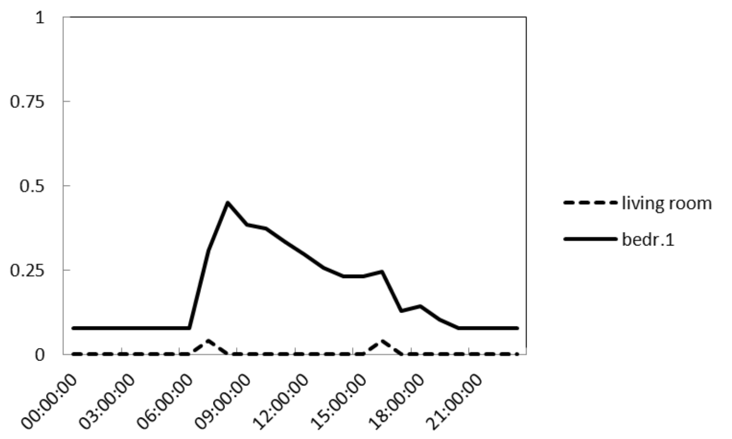

Figure 6.10 Average probability of window opening in bedrooms and living room in the neighbourhoods of the cases of group 2

6.3.4 Exposure Profiles

Figure 6.11 and Figure 6.12 present the average profiles of the carbon dioxide concentration measured in the living rooms and bedrooms for all cases. The reported concentrations are concentrations in excess of the outdoor concentration. The data was first aggregated to a single average exposure profile per room per case, the profiles shown in the figures below are the aggregates of these single profiles. Due to the fact that in some cases multiple bedrooms were monitored, the living room profile is compiled with 81 single profiles, while for the bedroom profile 116 profiles were available. There is a clear difference both in timing and height of the concentrations. The timing is of course directly linked to the occupancy profiles. Note that there is a good match with the reported occupancy profiles. The height of the concentrations depends on the combination of ventilation rate and occupancy density.

In the European standard EN 13779 [230], 400, 600 and 1000 ppm above the outdoor concentration are reported as limit values for high (IDA 1), medium (IDA 2) and moderate (IDA 3) indoor air quality respectively. Based on a minute ventilation of 6 l/min [259] and 35 mmHg end-tidal carbon dioxide concentration, the ventilation rate of 4 l/s/person recommended by the HealthVent project in conditions dominated by exposure to bio-effluents [51], 1250 ppm above the outdoor concentration is considered a threshold to avoid adverse health effects. These reference values are included in Figure 6.11 and Figure 6.12 with grey solid lines in the background.

In the living room, the average and median concentrations during the period of intensive occupancy are within the limits of the median indoor air quality class (IDA 2) proposed by the European standard for the living room, while for the bedrooms, they are well within the moderate air quality class (IDA 3). Towards the end of the night, the average concentration is even slightly within the poor air quality class (IDA 4). The 95-percentile values for the bedrooms are well above the health-effects threshold of 1250 ppm. This confirms the observations made in the previous section, suggesting that the probability of exposure to high carbon dioxide concentrations is much higher in the bedroom than it is in the living room [400].

Table 6.6 exposure probabilities for different indoor air quality levels

Threshold	P(living)	P(bed)	Rel. P_{occ}	Rel. P_{pres}
1250 (HealthVent)	0.02	0.09	13.2	6.8
1000 (IDA 3)	0.04	0.14	10.0	5.2
600 (IDA 2)	0.13	0.30	6.0	3.1
400 (IDA 1)	0.28	0.42	3.7	1.9

Table 6.6 lists the probability of exposure to concentrations above the thresholds proposed above, as well as the relative probability of these exposures both normalized to the total occupancy time (Rel. P_{occ}) and to the occupancy time of the specific room (Rel. P_{pres}). These durations were based on the occupancy profiles reported above, weighted by the number of cases in group 1 and groups 2-4 respectively.

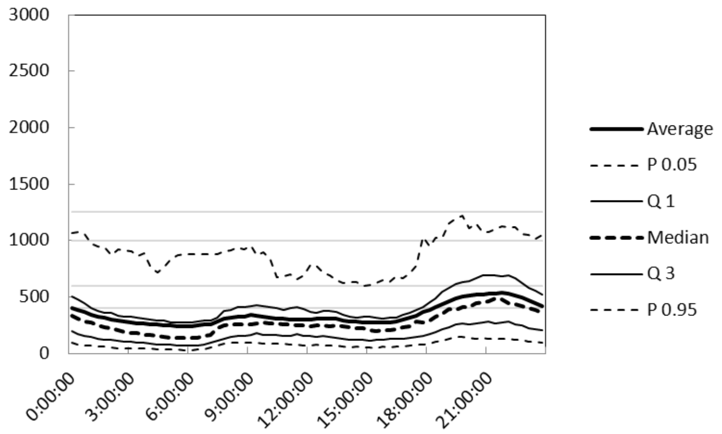

Figure 6.11 Average carbon dioxide concentration above outdoor concentration (ppm) profiles measured in the living rooms of all cases as well as their quartiles and 5-95 % percentiles

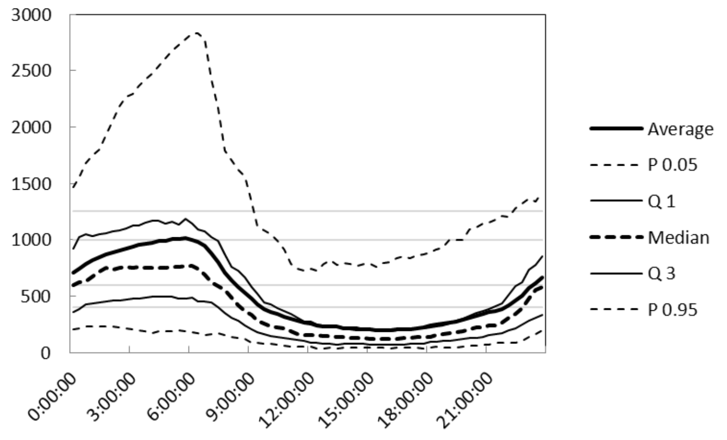

Figure 6.12 Average carbon dioxide concentration above outdoor concentration (ppm) profiles measured in the bedrooms of all cases as well as their quartiles and 5-95 % percentiles

Figure 6.13 and Figure 6.14 present the average profiles of the carbon dioxide concentration measured in the living rooms and bedrooms per group, for all cases of that group. In both spaces, the indoor air quality observed in the dwellings in group 1 is in the same range as that in the other groups, although no dedicated ventilation system is installed. The leakage through the building envelope ensures a sufficient flow rate of fresh air to achieve medium indoor air quality in the living room and moderate indoor air quality in the bedroom.

The performance of mechanical exhaust ventilation seems to be more sensitive to changes in occupancy levels (higher amplitude of the concentration) compared to mechanical supply and exhaust. This is readily explained by the fact that, although the flow rate in mechanical exhaust ventilation systems is assured by a fan, it is exhausted in typically polluted spaces such as kitchens, bathrooms and toilets. The distribution of the flow rate to the occupied spaces such as living room and bedroom will therefore depend on the overall flow regime within the dwelling. The living room is usually open to the kitchen, assuring its ventilation at an acceptable level. The bedrooms, however, are most likely situated on the second floor, with the bedroom doors closed during occupancy and the exhaust situated in the bathroom, again with the door closed during occupancy. The flow rate in the bedrooms of cases with mechanical exhaust ventilation systems is therefore mainly governed by unreliable driving forces such as wind and buoyancy acting on the trickle ventilators. Additionally, the location of the bedrooms on the second floor, which usually coincides with the neutral pressure plane, reduces the flow rate due to leakage. The combination of both effects renders rather undesirable indoor air quality levels in the bedrooms of mechanical exhaust ventilation cases. These effects have also been observed in simulation studies [400].

In the cases with mechanical supply and exhaust, the fresh air is distributed directly to the occupied spaces, resulting in a more stable concentration over time and across the spaces of the dwelling. The fact that the measured concentrations are still relatively high is caused by the tendency of the occupants to operate their ventilation system at low flow rates. As was mentioned earlier, a survey on flow rate selection by mechanical supply and exhaust owners in Belgium, demonstrates that the majority permanently selects the lowest flow rate (roughly corresponding to 1/3 of the design flow rate) [424] and similar results were reported in the occupancy questionnaires for the cases in groups 2-4. With this low flow rate, the impact of leakage is still relatively important, explaining the higher concentrations observed in the extremely airtight passive houses.

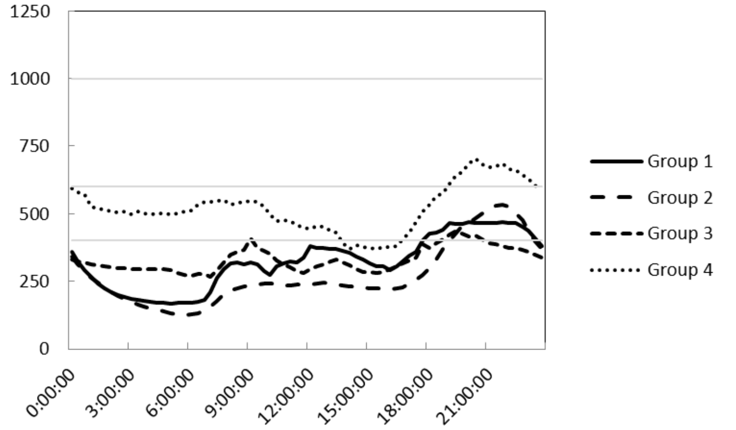

Figure 6.13 Average carbon dioxide concentration above outdoor concentration (ppm) profiles measured in the living rooms of all cases in the 4 groups

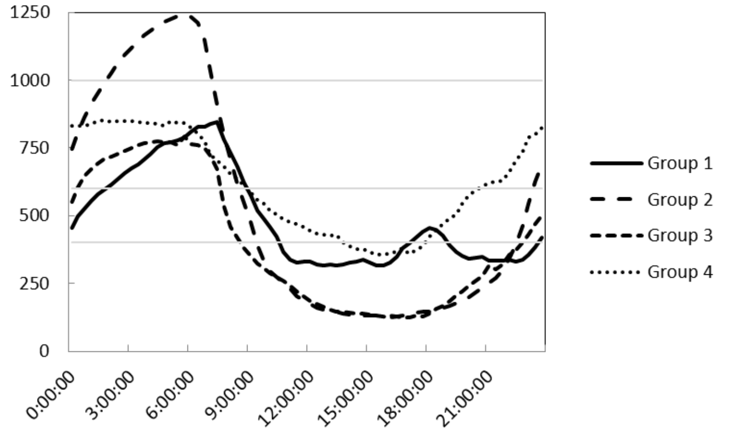

Figure 6.14 Average carbon dioxide concentration above outdoor concentration (ppm) profiles measured in the bedrooms of all cases in the 4 groups

6.3.5 Summary of Case Study

This section presented the results from carbon dioxide monitoring in the bedroom and living room of 47 standard and 34 low energy dwellings in Belgium and compares the typical exposure profiles for both spaces, confirming that the risk of exposure to high carbon dioxide concentrations in the bedroom is about 6 times higher than that in the living room. Additionally, it showed that indoor air quality levels comparable to that in leaky buildings are reached in airtight construction combined with mechanical supply and exhaust systems. Specific concerns with the latter, however, include the adequate ventilation of bedrooms, especially when situated on the second floor, in dwellings with mechanical exhaust ventilation systems and the selection of low ventilation rates by occupants in extremely airtight construction.

6.4 Response to Low Ventilation Rates

Based on the carbon dioxide concentrations measured in the bedrooms in the case study presented in the previous section, bedroom indoor air quality is at least cause for concern. Although some first evidence of a direct impact on decision making exists, carbon dioxide is generally accepted to be nothing but a marker for perceived indoor air quality. Since occupants spend most of the time in the bedroom asleep, the validity of a perception based exposure threshold is at best shaky. This section therefore presents the results from an intervention study that aimed to investigate the impact of low ventilation rates on sleep quality, both by objective wrist-actigraphy measurements and through the analysis of questionnaires.

6.4.1 Methods

As was mentioned in the introduction, a field study was set up to analyse the effect of ventilation rate on the sleep of the test subjects. This is a pilot study, a first attempt to document the effect of room ventilation on sleep. Therefore, although this contains lots of confounding factors, the effects are studied on the basis of high level parameters such as sleep efficiency and overall ventilation rate in the room rather than on more specific parameters such as air speed or respiratory rate. The field study was set up in the student dorms because all of the rooms in the dorms are identical and therefore the test conditions are very similar for all of the subjects, while the subjects are still tested in their normal sleeping environment. This was preferred over lab conditions since the move to the lab would introduce adaptation effects.

The dorm rooms have a habitable floor area of 12 m^2 and are designed for single occupancy. The dorms are not fitted with a ventilation system. All fresh air supply is due to infiltration and the opening of windows. While the geometry, occupancy and main furniture are equal in all rooms, personal items of the students such as books, bedding and decoration can still cause large differences in chemical emission rates in the different rooms. Additionally, students have arranged the furniture in the room differently, according to their personal preference, some sleeping directly next to the window, while others positioned their bed away from the façade, next to the door. The complete study was repeated one year later in a new dorm fitted with mechanical exhaust ventilation where each room disposed of its own private bathroom. The total floor area of these rooms is 20 m^2.

The test subjects were monitored in two different periods, a first group in the fall of 2010, a second group in the spring of 2011. From the original first group of 10 students, 2 were eventually eliminated due to their failure to comply with the basic test conditions, namely sleeping alone and abstaining from alcohol during the test period. Likewise, in the second group, 1 subject had to be excluded from the original 18 subjects. In total 25 subjects over the 2 groups were monitored, of which 6 male and 19 female, all of which were in good general health and between 19 and 25 years of age. The second study included 23 subjects between 18 and 25 years of age.

The tests were executed over a period of 1 month. The test period was divided in 2 sub periods of 2 weeks, in which the test subjects were asked to leave the window open at night during one of the sub periods and to close it during the other. This way, 2 separate conditions were measured, one with a high ventilation rate (window open) and one with a low ventilation rate (window closed). The ventilation rate in each individual room and (sub) period is subject to considerable variation due to differences in ambient conditions during the period, differences in orientation of the room and differences in local leakage. In the second study, the mechanical exhaust ventilation was activated in the first period and deactivated in the second period. The trickle ventilators were also closed manually in this period. Half of the students were subjected to a placebo experiment where no actual change in ventilation rate was applied.

Over the course of the whole test period, carbon dioxide levels (CO_2), temperature and relative humidity were measured in each of the subjects' room in 10 minute intervals. This data is used to characterise the sleeping environment of the test subjects. The CO_2 concentration is used as a proxy for the ventilation rate. The same instruments were used to monitor the sleeping environment of the same subject during the 2 sub periods of the test in order to minimize uncertainty in the data. The instruments were placed in the direct vicinity of the head of the test subject, usually on a nightstand or a bookcase above the head board. Care was taken that subjects did not breath directly on the instrument.

The sleep pattern of the subjects is monitored using the actigraphy technique. This technique was selected because of its easy applicability in situ, and its low impact on the sleep of the subject compared to poly-somnography, that is the most commonly used and most accurate measurement technique for sleep patterns, while still rendering acceptably accurate results [450]. Additionally, the technique is relatively cheap, allowing to test a larger group of subjects simultaneously and over a long period.

The data, collected with the actigraph devices in 60 second epochs, was analyse with the Sadeth algorithm [451] and is expressed in numerical values by the number of awakenings, the average time awake, sleep onset and sleep efficiency. Sleep efficiency is the ratio of the time asleep to the total time spent in bed, expressed as a percentage. Sleep efficiencies between 80 and 95 % are considered normal. The devices were worn on the non-dominant wrist and, as with the instruments for the monitoring of the sleeping environment, the same device was used on the subject for the 2 sub periods of the test.

Next to the physical measurements, the subjects were also asked to complete a number of questionnaires in order to collect information about their attitude towards sleep and about their subjective appreciation of the quality of their sleep during the test period.

For this purpose, a number of questionnaires presented by Billiard [452] were translate to Dutch. The 'Morning Questionnaire' was completed every morning upon awaking to characterise the appreciation of the sleep of that night as well as to identify specific events and disturbances for that particular night.

At the beginning of the test period, the subjects completed the 'Sleep Impairment Index'-questionnaire [453], the 'Horne and Östberg' questionnaire [454] and the 'Dysfunctional Beliefs and Attitudes about Sleep' questionnaire [455] in order to assess their attitudes towards appreciation of sleep. The data from these particular questionnaires is mainly used to filter out any subjects that suffer from severe sleep disorders or dysfunctional sleep.

The 'Pittsburg Sleep Quality Index' questionnaire [456] was completed before the start of the measurement, in between the 2 sub periods and at the end of the test period to assess differences in the appreciation of sleep quality and sleep patterns between the 2 test conditions and the 'Leeds Sleep Evaluation Questionnaire' [457], designed to assess the effect of medication on sleep over a longer period, was completed at the end of the experiment. In the latter questionnaire, the subjects are asked to assess the difference in their perception of their sleep quality over the 2 sub periods of the test directly, while the differences in answers to the former are an indirect measure.

The temperature difference and draft induced by the higher ventilation rate was minimized by selecting the autumn and spring season for the monitoring campaigns. The weather conditions during these months are such that the indoor / outdoor temperature difference is minimal. Opening the window also influences the sound level in the room. We were not able to monitor the sound level, but we can generally assume it to be higher with the opened window. This is therefore a confounding factor in the collected data.

6.4.2 Results

For each of the following results, test subjects that had incomplete data for the results reported were eliminated. The number of subjects concerned is reported with each result. The subjective results from the questionnaires will be treated first, followed by those from the objective measurements. Since the distributions of the answers can't be assumed to be normally distributed, non-parametric significance tests are used to analyse the data.

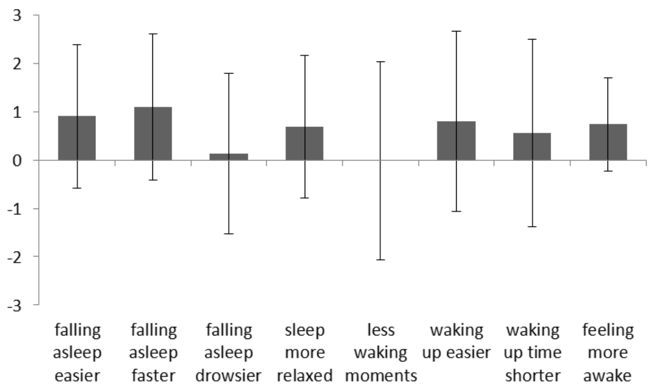

Figure 6.15 Average and standard deviation of responses of 20 subjects to Leeds questionnaire window open vs. closed

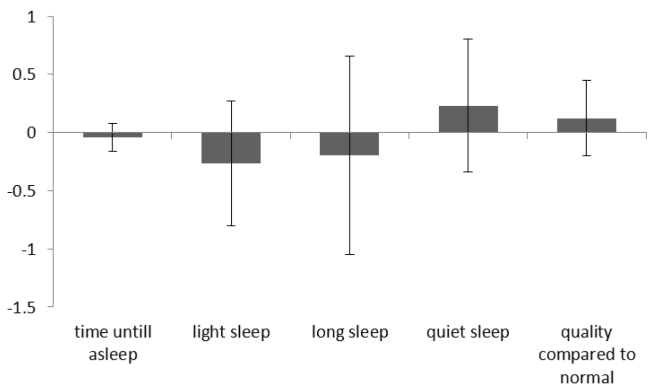

Figure 6.16 Average and standard deviation of difference in sub period averaged response of 21 subjects to the morning questionnaire, window opened vs. closed

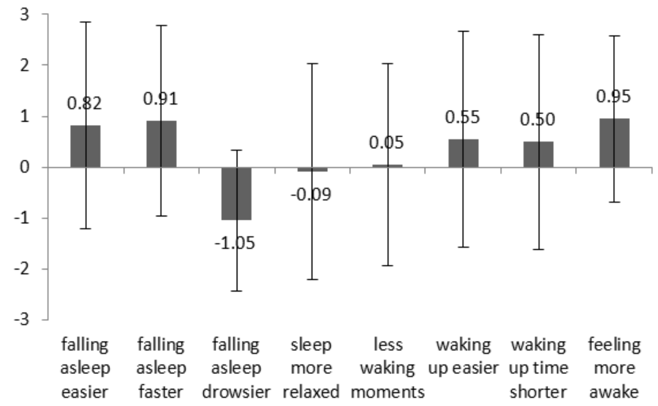

Figure 6.17 Average and standard deviation of responses of 11 subjects to Leeds questionnaire mechanical exhaust ventilation vs. no ventilation

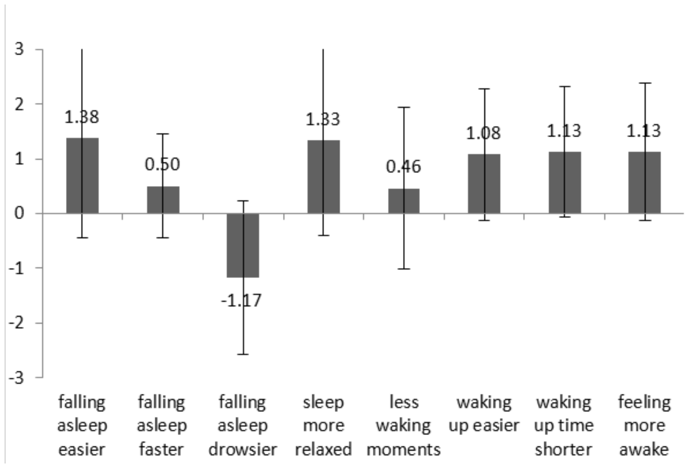

Figure 6.18 Average and standard deviation of responses of 12 subjects to Leeds questionnaire mechanical exhaust ventilation vs. placebo

The Leeds sleep evaluation questionnaire, as was mentioned in the methods section, asks the subjects to directly rate their perceived sleep quality in the two sub periods against one another with a mark on a continuous line. The scale ranges from -4.5 to 4.5. The average answers to the different questions in the questionnaire and their standard deviation are shown in Figure 6.15 (n subjects = 20). It is remarkable that the sleep quality is rated higher for the sub period with opened window on all of the questions in the questionnaire, although the test subjects do not seem to perceive a change in the number of waking moments. Using the Wilcoxon signed rank test, this and the level of drowsiness before falling asleep were the only questions found to not be perceived significantly different between the sub periods ($p > 0.1$). The most significant differences were reported in ease to fall asleep, speed of falling asleep and feeling generally well rested and awake the day after ($p < 0.01$), while the sleep was felt to be more relaxed and waking up was perceived to be easier ($p < 0.5$) and more quickly ($p < 0.1$).

In Figure 6.16., the average over 22 subjects of the difference in the average answer for the separate sub periods for each subject and the standard deviation of the same difference are shown for the questions with scaled answers in the morning questionnaire that showed significant differences for both test periods ($p < 0.1$) using the Wilcoxon signed rank test (n subjects = 22, n nights window closed = 215, n nights window opened = 232). The sleep in the sub period with opened window was perceived to be deeper ($p < 0.02$) and of higher quality ($p < 0.05$). At the same time the subjects reported shorter intervals to sleep onset, shorter total sleep and more rest full, quiet sleep ($p < 0.1$).

The temperature in the rooms was on average 18 °C and the relative humidity 63 %. No significant differences were found between both test periods. While the subjects reported to be better rested and deeper asleep in the sub period with the window opened, the actigraph measurements suggest that the sleep efficiency during this period was lower. In Figure 6.19, the average sleep efficiency in both sub periods and their standard deviations are shown for each of the subjects (n subjects = 22, n nights window closed = 211, n nights window opened = 201). Analysing the paired averages for both test periods for each subject with the Wilcoxon signed rank test, the sleep efficiency in the period with the opened window was significantly lower than that in the sub period with closed window ($p < 0.1$). Although the differences are generally small, applying the Wilcoxon rank sum test on the nightly data of a single subject showed that the sleep efficiency was significantly lower in the sub period with opened window for 4 subjects ($p < 0.1$).

The results from the repeated study show very similar results (Figure 6.17), but the test subjects of the placebo group reported virtually identical effects (Figure 6.18). Plotting the sleep efficiency versus the peak carbon dioxide concentration over the outdoor concentration for all nights and all subjects in both studies, like in Figure 6.20 (n = 634), also shows no clear correlation between these two values.

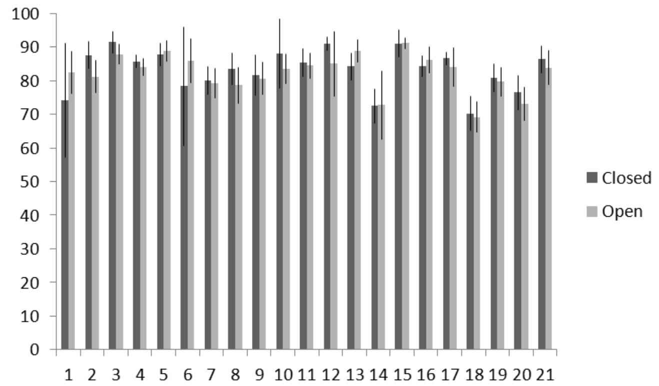

Figure 6.19 Average and standard deviation of sleep efficiency, window opened vs. closed (n subjects = 21, n nights window closed = 211, n nights window opened = 201)

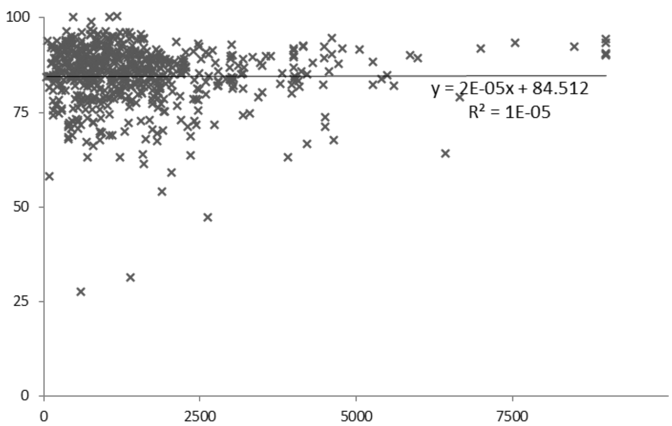

Figure 6.20 Sleep efficiency vs. CO_2 concentration (ppm, n = 634)

6.5 Exposure While Asleep

From the previous sections, one could conclude that, although indoor air quality conditions in the bedroom seem less than optimal considering traditional assessment parameters based on perceived air quality and bio-effluents, there seem to be little to no obvious acute repercussions on sleep quality associated with these exposure level. A further conclusion would then be that flow rates in bedrooms are currently oversized and substantial energy savings could be envisioned by lowering design ventilation rates for bedrooms. This can, however, only be the case under the condition that bio-effluents are the sole type of pollutant to be considered.

There is very little information about human exposure in the sleeping environment, even though the conditions in this environment are at least worrisome. The exposure during sleep is characterized by long exposure time, both absolute and relative, prevalence of specific pollutants and uncustomary proximity of sources to the breathing zone. This section presents experimental results that show the impact of the proximity of possible emission sources such as mattress, pillow and toy, on exposure of the sleeping subjects to these emissions. Based on full scale experiments in an environmental chamber using a breathing thermal manikin, the intake fractions for gaseous pollutants are measured as well as the occurrence of rebreathing. The study reports intake fractions for several sleep positions as well as different bedding arrangements. The results show that human metabolism and corresponding heat release by the human body are dominant factors in the dilution of pollutants emitted in close proximity of the nose, reducing exposure by 40% compared to a case without metabolic heat output. This effect is more important than the sleep position. An additional important finding is that sleeping with the head under the covers increases intake by a factor 24 and results in a rebreathing rate of over 60%.

6.5.1 Context

Exposure to harmful air pollutants in dwellings can largely be classified in 2 categories [1, 2]. The first category contains all activity related exposures, such as cooking [118, 119, 121, 128, 134], cleaning [10, 137, 138, 142], indoor smoking [28, 112, 117, 176]. The second category is composed of exposures to building material and furniture emitted pollutants [6, 11, 72, 182, 387]. On average, a person spends about one third of his life asleep. This amounts to about 60% of the total time spent in dwellings [357, 358], making the bedroom the most likely place for exposure to pollutants from building, furniture and other indoor materials, especially since bedrooms are likely to be poorly ventilated [400, 444]. While human triggered pollutant emission in the bedroom is limited mostly to the bed and bedding materials, exposure in the bedroom to activity based pollution can, still represent a large fraction of the total exposure to these pollutants due to the long and continuous presence in the bedroom and to redistribution of emissions in other spaces throughout the dwelling [eg. 129].

In bedrooms, typical room specific elements such as mattresses may be sources of phthalates, isocyanates and formaldehyde and are contributing to the total material emissions [90, 91]. Additionally, mattresses, pillows and bed linens are often heavily treated with flame-retardants [eg. 93], detergent components and other substances that are known to have an impact on human health [92, 458, 459]. The mattress is also known to be a great biotope for dust mites [25, 95, 96]. Their faeces are a source of fine and ultra-fine particles.

Next to the general toxicological consequences of these exposures, a secondary concern with these specific sources is their proximity to the sleeping person. Most dose-response models are based on the well-mixed room assumption. The perfect mixing concentration is used to predict exposure [eg. 460]. Likewise, carbon dioxide controlled ventilation systems are based on the concentration measured by a sensor that is, at best, positioned to represent the well-mixed pollutant concentration. While asleep, the nose and mouth are in direct proximity of emission sources such as mattresses and bedding materials for a long period of time. It is fair to assume that under these circumstances, the real concentration of the emitted pollutants in the inhaled air will, due to the proximity of source and sleeping person, be higher than the perfect mixing concentration. Therefore, traditional exposure assessments based on the well-mixed room assumption are not applicable.

Exposure to one particular gas in the sleeping environment is worth further attention. The main product of our metabolism is carbon dioxide. Traditionally, carbon dioxide (CO_2), an odorless, colorless and virtually inert gas, has been considered to be nothing more than an indicator for the concentration of other, harmfull metabolic by-products in an environment with no open combustion, which in turn correlate with perceived air quality [67]. Based on these findings, a correlation between desired indoor air quality and CO_2 was established, eg. in European ventilation standards [230]. Nevertheless, the exposure conditions in the bedroom are fundamentally different in the bedroom than in the other rooms, mainly due to the proximity issue and to the fact that in a large majority of the time spent in the bedroom, the occupants are asleep. During this time, a person is in a semi-conscious state. Therefore, traditional IAQ ratings based on perceived air quality are not applicable. Recent studies, however, suggest that carbon dioxide in itself can also cause physiological reactions, reducing the decision making ability of test subjects [461].

Additionally, the control of normal breathing during sleep is still under debate. Sleep medicine is a relatively young field and the mechanisms or even the purpose of sleep are not yet fully understood. Breathing disorders during sleep, of which apnea and sudden infant death syndrome are the most severe and the most commonly known, are subject to extensive research, leaving little room for fundamental research in normal breathing during sleep. Breathing during the first stages of non-REM sleep seems to be inherently unstable and regulated by slow processes, while in full non-REM sleep, the overall breathing rate is lowered and breathing reaches a steady state again. Carbon dioxide concentration seems to play an important role in the control processes in both stages [462].

$C0_2$ is exhaled at high concentration (4-5 %) and, although it is counterphase with the inhalation from a time perspective, this kind of source within the breathing zone itself raises some concern. Due to the specific flow conditions created around the head in the sleeping environment, exhaled air could be trapped in the breathing zone. Increased rebreathing of exhaled air will dramatically increase the inhaled CO_2 concentration and could lower mental fitness or destabilize the normal breathing control process. The latter is one of the suggested causes of sudden infant death syndrome [463, 464].

The goal of the experiments presented in this chapter is quantifying the effect of the proximity of pollution sources to the breathing zone in the sleeping environment on the exposure of the sleeping subject to the emitted pollutants. More specifically, this effect is studied with regard to two categories of sources, namely the bed and the human metabolism. Quantifying the amount of re-inhalation of exhaled metabolic CO_2 ('Rebreathing') will allow to investigate the benefits of different sleep positions and should provide valuable information to further the understanding of role of CO_2 in the control of breathing during sleep, while exposure risk analysis that back chemical usage restrictions for mattresses, bedding or pillows can only be relevant if they take the effect considered into account. The next section explains the way the experiments were set up, what configurations were tested, how the results were processed and the methods used to assess uncertainty in the results. The third section focusses on the presentation and discussion of the individual results, while the last section brings them together in more general conclusions.

6.5.2 Methods

In this chapter, the impact of source proximity and rebreathing for emissions in gas phase is assessed with environmental chamber experiments involving the use of a breathing thermal manikin. Allowing both greater flexibility in and better control of the test conditions, this approach was preferred over measurements on living subjects. Furthermore, the experiments can be used as validation cases for subsequent numerical studies that focus on specifics related to exposure in the sleeping environment. In this section, the experimental setup will be explained first, followed by an overview of the experimental matrix, a discussion on results processing, uncertainty analysis and quality control.

Experimental Setup

A 70 m^3 stainless steel environmental chamber, in the Air Pollution and Energy Flow Testing Facility at the University of Texas at Austin, was used for all experiments. A HVAC unit is situated on top of the chamber and the whole chamber sits within a climate controlled laboratory. The temperature in the lab is 24 ± 1 °C, the internal volume of the lab is 500 m^3 and is ventilated at 8 ACH. The chamber is ventilated at a rate of 0.5 ACH, corresponding to the upper boundary of the IDA 2 class for an occupancy of 1 person, as proposed in the EN 13779 standard [230]. There is no recirculation and the fresh air is taken from the lab. The return air is exhausted to the exterior to prevent contamination of the fresh air. The air is supplied to the room with a low velocity diffuser sitting on the ground on the north side of the environmental chamber, exhaust is situated in the west wall, near the sealing, at 2.45 m height.

A breathing thermal manikin [258, 465, 466] is positioned on a twin size mattress on a bedframe, with the headboard against the centre of the east wall of the chamber. Nose breathing is chosen for the manikins breathing mechanism, which is the most common way of breathing for non obstructed airways. The minute ventilation was set to 6 l/min and respiratory rate to 12 breaths a minute, in correspondence with observations made in sleeping subjects [462] and the metabolic heat load of the manikin is set to 70W of sensible heat loss, corresponding to a person in rest.

To mimic pollutants emitted from the different sources, 0.1 % SF_6 in nitrogen was used as a tracer in the experiments. Depending on specific emission technologies that simulate different sources, it represented gaseous pollutants emitted from various sources. Polluted exhaled air and emissions from 3 sources were studied, namely: the mattress or bedding that covers the mattress, a pillow or bedding that covers the pillow and as a common children's toy a teddy bear.

All lines in the tracer gas injection mechanism and breathing mechanism were made from nylon tubing. A multi-point distribution system from nylon tubing was built in each of the sources to achieve uniform source strength over the whole surface of the selected sources. Each of the distribution systems was built on a fractal geometry such that the distance from the central supply line to each of the point sources was the same and is composed of an equal number of T sections. The mattress was a hollow spring mattress with felt padding and woven poly-propene lining. The sides and bottom were sealed with poly-ethylene foil and 8 point sources were situated in the spring compartment. 4 point sources were positioned in the centre plane of the poly-ester fibre filling of the pillow. The teddy bear was, the same way as the pillow, equipped with 2 point sources. For the intake fraction experiments, the SF_6 was injected in the central supply line using a Omega FMA5400 0-10 ml/min mass flow controller.

For the rebreathing experiments, the SF_6 was injected into the exhaled air line. The two lines were joined together at the hip of the manikin. In this case, the volume of this line (0.012 l) is small compared to the dead space in an actual human (0.167 l). This is done to reduce possible cross contamination of the lines.

The SF_6 concentration in the inhaled air is measured using a Lagus Autorac GC-ECD analyser with a calibrated range of 0.1-10 ppb and a precision of ± 3%. In order to be able to measure the concentration of the pulsating (0.2 Hz) inhalation flow, the inhaled air is gathered in a 10 l sample bag and the analyser draws samples from this bag. In the experiments assessing relative intake fraction, the exhaled air is taken from the bag. In the rebreathing experiments, the bag is emptied at a continuous flow rate and the exhaled air is returned to the environmental chamber to avoid contamination of the fresh air intake. This setup is only suited to characterize the relative intake fraction and rebreathing rate in steady state conditions. The air flow pattern around the body can be assumed to stabilize quickly and sleep is typically a long term condition. Therefore, the steady state assumption is deemed to be representative. Particulate pollutants are governed by resuspension mechanisms that are inherently transient and were therefore not studied. The SF_6 concentration is simultaneously monitored at 5 additional positions in the environmental chamber, of which 1 in the return duct, 3 in the bulk volume of the chamber and 1 fixed to the mattress surface about 0.15 m from the top left corner. The first of the sample points in the bulk room is positioned at 1.6 m from the north wall and 0.8 m from the west wall, at a height of 0.6 m. The second is positioned at 1 meter from the north wall and 1 meter from the east wall, at a height of 1.1 m and the last one is positioned at 1 m from the south wall and 1 meter from the east wall, at a height of 1.6 m. All positions are shown in Figure 6.22.

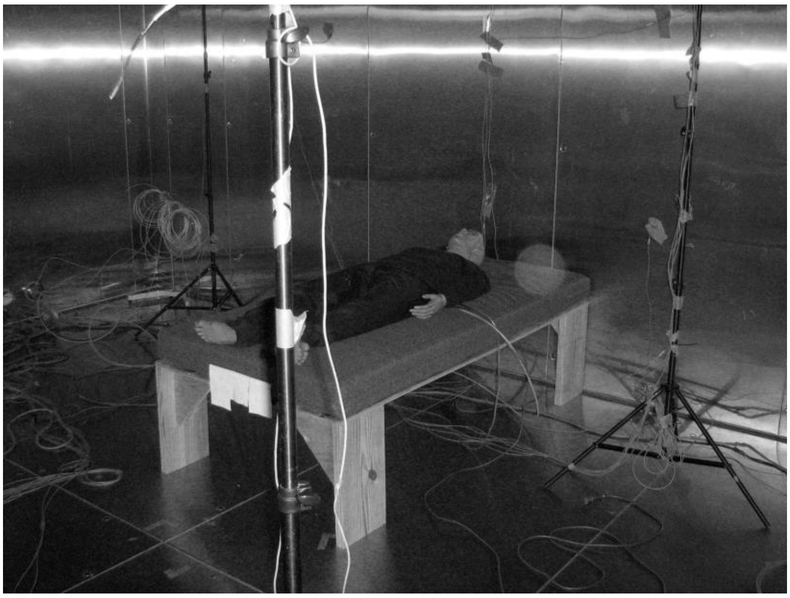

Figure 6.21 Picture of the inside of the climate chamber during base case experiment showing the manikin in supine position

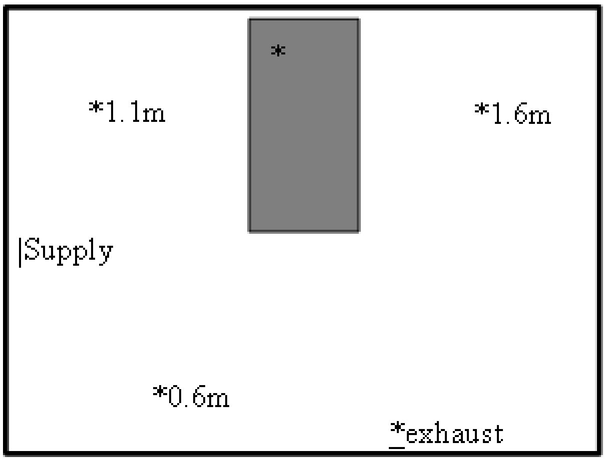

Figure 6.22 Experimental setup: (left) floor plan of the environmental chamber showing the position of the sample tubes (*), the bed, supply and exhaust, (right) interior of the chamber with the manikin in the supine position on the bed

Experimental Matrix

The experimental configurations considering the different studied positions with the experimental specifics are listed in Table 6.7, along with the number of repetitions that have been done for each setup. The manikin is positioned in a way that corresponds to 3 different common sleeping positions. Figure 6.23 shows a schematic of the 3 positions, the first position is the supine position as is shown in Figure 6.21. The manikin is positioned on his back on the centre line of the bed, with the head about 10 cm away from the east wall, the arms positioned on the side of the torso. This position with the mattress source is chosen as the base case. In a second position, the manikin is again positioned on the centre line of the bed, the head 10 cm from the top, but lying on his right side, facing north. In this lateral position, the arms are held alongside the torso, with a 90° forward angle at the elbow. In a third position, the manikin is again on the centreline of the bed and is turned on the chest. In this prone position, the arm is positioned along the torso, while the other is positioned upward with the left hand next to the head of the manikin. This allows the left shoulder to be slightly propped up and the face of the manikin to be turned to the left for about 10°. With the pillow source, only supine and lateral positions are tested. The teddy bear source is tested in supine position with the teddy bear sitting in the arm pit and in prone position with the bear propped up against the nose.

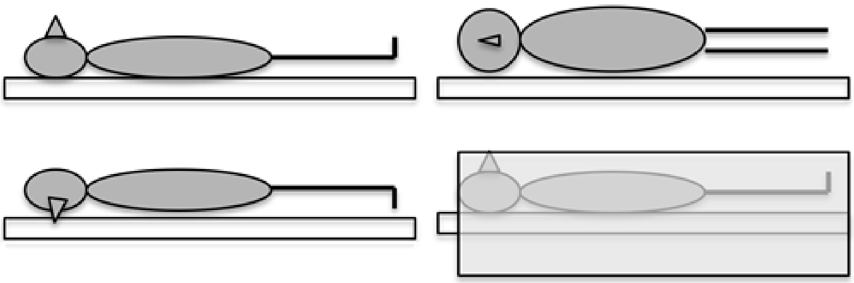

Figure 6.23 Schematic of supine (top left), supine (top right) and prone (bottom left) sleeping position, as well as the supine position with the blanket covering the head of the manikin (bottom right)

In the base case, no bedding was used. The influence of this is investigated by adding a pillow (without SF_6 source) and a blanket. Two arrangements for the blanket are tested: covering the manikin from the torso down and covering the whole manikin, including the head. The latter arrangement is also shown in Figure 6.23.

Table 6.7 Test Configurations

	source	position		repeat
1	Mattress	Supine	Base Case	8
2	Mattress	Supine	High Mixing	1
3	Mattress	Supine	Metabolism 0 W	3
4	Mattress	Supine	Low SF_6 source strength	2
5	Mattress	Supine	Pillow	1
6	Mattress	Supine	Pillow and blanket	2
7	Mattress	Supine	Blanket covering head	3
8	Mattress	Lateral		3
9	Mattress	Lateral	Arm propped under head	3
10	Mattress	Lateral	Arm under head, covered with blanket	1
11	Mattress	Prone		1
12	Pillow	Supine		3
13	Pillow	Lateral		2
14	Pillow	Lateral	Head at edge of pillow	1
15	Teddy	Supine		3
16	Teddy	Supine	High Mixing	1
17	Teddy	Lateral		3
18	Teddy	Lateral	High Mixing	1
19	Rebreathing	Supine		1
19	Rebreathing	Supine	Low SF_6	1
20	Rebreathing	Supine	Blanket covering head	2
21	Rebreathing	Lateral		1
22	Rebreathing	Prone		1

Primary Comparative Metrics

Since SF_6 is used as a tracer and could represent any gas phase emission from the source, only the ratio between the measured intake fraction and the intake fraction calculated based on the well-mixed concentration is relevant. The intake fractions [261] are, as in the previous chapters, calculated as the ratio of mass intake to the respiratory system and mass emitted from the source:

$$IF = \frac{\int Q_b(t) \cdot C(t)dt}{\int E(t)dt}, \qquad (6.1)$$

where

IF	=	intake fraction (g/g)
Q_b	=	breathing rate (l/s)
C	=	concentration (l/l)
E	=	emission rate (l/s).

The relative intake fraction can then be found by:

$$\beta = \frac{IF_{real}}{IF_{wm}} = \frac{\dfrac{\int Q_b(t) \cdot C_b(t)dt}{\int E(t)dt}}{\dfrac{\int Q_b(t) \cdot C_{wm}(t)dt}{\int E(t)dt}} = \frac{\int C_b(t)dt}{\int C_{wm}(t)dt} = \frac{\bar{c}_b}{\bar{c}_{wm}}, \qquad (6.2)$$

where

C_b	=	breathing zone concentration (l/l)
C_{wm}	=	well-mixed concentration (l/l).

In the test chamber, the well-mixed concentration of SF_6 is equal to the concentration measured in the exhaust.

Quality control and uncertainty analysis

Four preliminary tests were run to test the test setup. To test for leaks in the sampling system, SF_6 was first injected directly in to the supply ventilation duct of the environmental chamber. When steady state is reached, this will create uniform concentrations in the whole room. After 10 hours, the breathing mechanism was started and after 1 more hour, the analyser was started. Results are shown in Figure 6.24. A uniform concentration is observed throughout the chamber, validating the assumption of well-mixed conditions. The concentration in the sample bag was consistent with those in the camber, confirming minimal losses or artefacts in the sampling system. The sample bag was sampled every 6 minutes, each of the other locations every 30 minutes. All subsequent test respected the same sequence. Each test was run for 3 hours or more.

All uncertainties reported in the chapter are 95% confidence intervals based on a student-t distribution. The test uncertainty was reduced by averaging both concentrations (C_b and C_{wm}) over the test period. Absolute systematic error was mitigated by calibrating the GC against a standard before every test, while relative systematic error was cancelled by division. These confidence intervals, however, only include errors due to the instruments and not those due to the repeatability of the experiments. The repeatability of the test procedure, due to exact position of the manikin, uncertainty in boundary condition etc. was assessed by testing the base case 8 times, rebuilding the test setup, flushing the test chamber with outdoor air and restabilising it over night between each test. A number of test configurations were repeated 2-3 times to quantify the repeatability in other conditions.

To explore the effects of flow conditions in the test chamber, mixing fans were installed in the test room and the metabolic rate of the manikin was reduced to 0 W. The sensitivity of the results to the source strength was also tested by reducing the SF_6 supply. In the cases where the test was repeated three times, the mean relative intake fraction and its confidence interval are reported assuming the mean to be student-t distributed with n-1 degrees of freedom (dark grey bars in the graphs in the results section). Except in the repeatability test with the base case, due to the low number of tests, these confidence intervals are much larger than the spread between the measured values (3 tests, 2 degrees of freedom, $CR_{0.95}$ = 4.3 times the standard deviation). Therefore, the measured values are also reported as such (light grey bars). If only 2 tests were completed, both values are reported.

6.5.3 Results

In the following section, the measured intake fractions are reported. Starting from the base case, the effects of sleep position and bedding arrangements are discussed, continuing with the pillow and teddy bear source and rebreathing.

Base Case Tests

The measured concentrations for the first base case (supine position) test are shown in Figure 6.25. The measured relative intake fraction was 1.25 ± 0.04. The inhaled SF_6 concentration was greater than the well-mixed concentration but much lower than the concentration measured on the mattress surface. Since the supply flow rate was low (0.5 ACH) and all walls were effectively adiabatic, the thermal plume caused by the buoyancy effect of the heated manikin appears to play an important role in the dilution of the pollutant in the breathing zone. The results of the repeated base case test indicated good repeatability, with a relative standard deviation of just under 3%. The mean relative intake fraction was 1.24 ± 0.03. Since the error due to repeatability issues is larger than the measurement error, the measurement equipment is considered sufficiently precise.

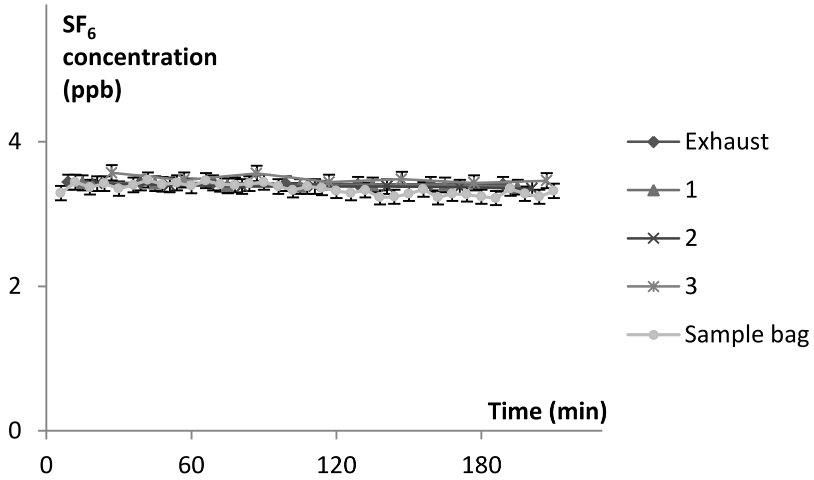

Figure 6.24 SF_6 concentration in the sample bag, return duct and 3 locations in the environmental chamber under uniform SF_6 concentration conditions, showing good agreement of the measured concentrations for all sample points

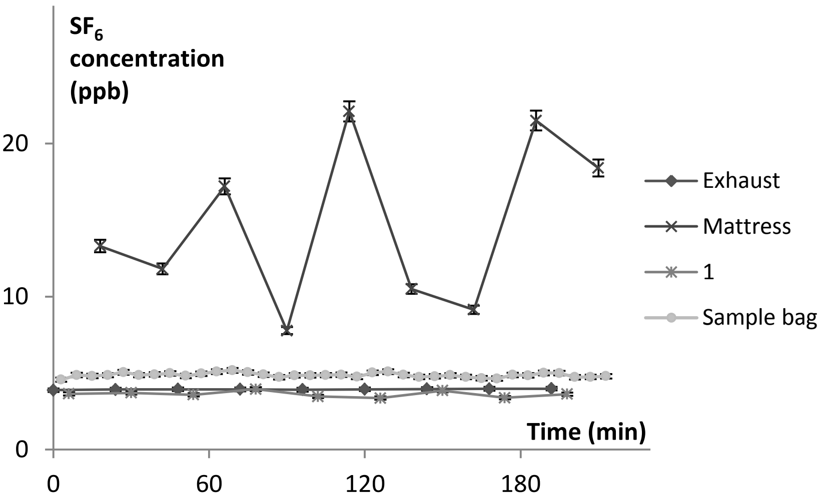

Figure 6.25 SF_6 concentration in the sample bag, return duct, 1 location in the climate chamber and on the mattress surface under base case conditions, showing a mildly elevated concentration in the sample bag versus high concentrations on the source surfaces

As was described in the methods section, the sensitivity of the relative intake fraction to the flow conditions in the chamber was tested and the results of these tests are shown in Figure 6.26.

Higher mixing had virtually no impact on the relative intake fraction. This can be explained by the nature of the mattress as a source. In the base case, the thermal plume was responsible for the efflux of fresh air to the breathing zone for dilution of the emitted pollutant. The high mixing disturbs this flow pattern, but nevertheless, the air entering the breathing zone has to travel about the same distance over the emission source, to the breathing zone, which is completely encapsulated within the emission area. The indifference to the reduction of the emission rate confirms the validity of the test setup to test the relative intake fraction. The crucial role of the thermal plume in the base case is clearly demonstrated by the significant ($p = 0.016$) increase (78%) of the relative intake fraction in the experiment without metabolic heat flux, with a mean value of 2.2 ± 0.8. The large confidence interval is due to the low number of repetitions of the test. Nevertheless, in the absence of the thermal plume, the flow field was much less stable, which is reflected by the increase in relative uncertainty on the result of a single test compared to the base case, from 2 % to 7 %.

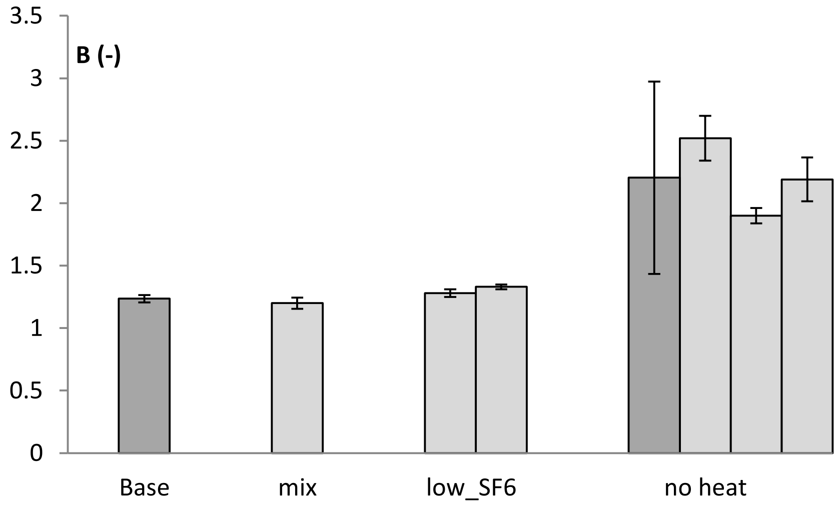

Figure 6.26 Results of the sensitivity analysis. Presented are: mean relative intake fractions (dark bars) measured in the case with metabolic heat load (base) and without metabolic heat load (no heat) as well as individually measured relative intake fractions in each repeated experiment (light bars). The diagram also shows the measured relative intake fraction for the case with higher air mixing (mix) and the case with lower SF_6 source strength (low_SF6)

Effect of Sleep Position

In the following step, the impact of the sleeping position was assessed. The results for the base case (supine position), the lateral and prone position are shown in Figure 6.27. The results show that the mean relative intake fraction in the lateral position was slightly higher ($p = 0.011$) than in the supine position (1.39 ± 0.13 vs. 1.24 ± 0.03), and even higher in the prone position (1.73 ± 0.09). Since only one result was available for the latter, significance of this increase could not be tested. Considering the confidence intervals on both results, the increase is substantial.

In both cases, lateral position and prone position, the breathing zone is closer to the source. Additionally, the free afflux of fresh air due to thermal buoyancy is blocked of by the mattress surface. When sleeping on the side, it is common for an individual to place the lower arm under the head. This configuration has a considerably impact on the relative intake fraction (Figure 6.27). The additional transport of fresh air caused by the thermal plume of the arm in the breathing zone reduces the mean concentration even below the well-mixed concentration. Both the test with the lateral configuration and with the lateral configuration with the arm propped under the head were repeated 3 times to eliminate random effects. Even with the large confidence intervals due to the low number of repetitions, the mean relative intake fractions, 1.39 ± 0.13 and 0.88 ± 0.11 respectively, are significantly different ($p = 10^{-4}$) between both configurations.

Effect of Bedding Configuration

The impact of bedding on the relative intake fraction is rather limited, as long as it only covers body parts other than the face. For example, the use of a pillow under the head in supine position, for example, does not have a sizable impact (relative intake fraction 1.21 ± 0.06). Covering the entire body from feet up to the torso with a blanket had only a very moderate impact (relative intake fraction 1.13 ± 0.03 and 1.05 ± 0.04 from two tests, $p = 0.076$). The latter can be explained by the fact that the blanket introduces an extra diffusion and flow resistance between the source and the breathing zone and that the thermal plume will only develop above the blanket, thereby bringing in slightly fresher air. On the other hand, the plume will also be attenuated due to the lower surface temperature of the blanket.

When the entire manikin, including the head, was covered with the blanket, there is a substantial change in exposure (Figure 6.28). The relative exposure is increased more than 25-fold, from 1.24 ± 0.03 in the base case to 32.7 ± 1.4 in the case with the blanket covering the face ($p = 5 \cdot 10^{\wedge -5}$). The same effect (relative intake fraction 21 ± 0.2) was found with the manikin the lateral position with the arm propped under the head and completely covered by the bedding. The extra diffusion resistance of the blanket and the deviation of the thermal plume, which will develop above the blanket and only move the air above the blanket, create a short circuit in the flow pattern in the breathing zone, further emphasizing the role of the thermal plume. It is clear that this situation is not a good sleeping environment and is best avoided if possible.

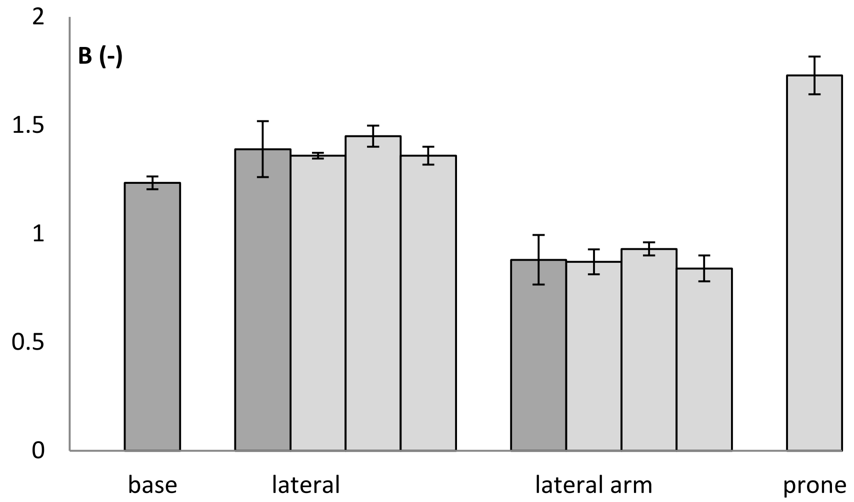

Figure 6.27 Results for different sleep positions. Presented are: mean relative intake fractions (dark bars) measured in the supine sleeping position (base), lateral sleeping position (lateral) with and without arm propped under the head. The diagram also shows individually measured relative intake fractions in each repeated experiment (light bars) along with the measured relative intake fraction for the prone sleeping position case (prone)

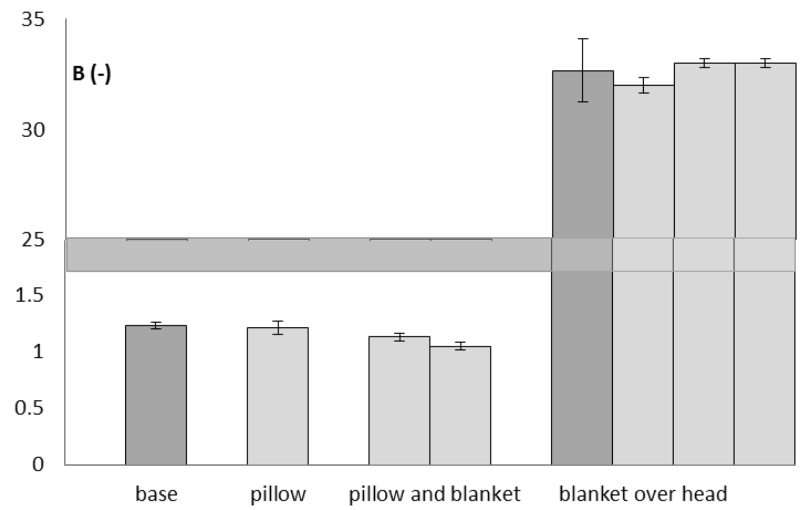

Figure 6.28 Results for different bedding arrangements. Presented are: mean relative intake fractions (dark bars) measured in the supine case (base) and in the case where the face of the manikin is covered by the blanket (blanket over head). The diagram also shows individually measured relative intake fractions in each repeated experiment (light bars) along with the measured relative intake fractions for the case with a pillow (pillow) and the case with a pillow and a blanket covering only the torso and legs (pillow and blanket)

Experiments with Sources from Pillow and Children's Toy

The relative intake fraction for the supine and lateral positions with the pillow as a source are shown in Figure 6.29. The tests with this source proved to be highly repeatable, with a mean relative intake fraction, based on three tests, of 2.16 ± 0.14 for the supine position. Note that the relative intake fraction is considerably higher than with the mattress source due to emission directly into and closely around the breathing zone from the pillow. In the lateral position, the relative intake fraction increased to 3.39 ± 0.08 and 4.02 ± 0.09 based on two tests. The effect of the smaller emission surface, makes the results more sensitive to relatively small changes in position of the manikin. As an example, the manikin's face was moved to the edge of the pillow in the lateral position (also a common sleeping position), which reduced the relative intake fraction considerably to 1.31 ± 0.02.

With a children's toy (a teddy bear) as a source, two sleep positions were tested that further illustrate the sensitivity of the relative intake fraction to source and breathing zone position for smaller emission sources. For the first configuration, the manikin was placed in the supine position, with the teddy bear placed in the arm pit of the manikin. In the second configuration the manikin was put in the prone position with the teddy bear propped against the nose. The results in Figure 6.30. show the impact of these alternative positions is large ($p = 4{\cdot}10^{\wedge-5}$), with mean relative intake fractions of 0.67 ± 0.03 and 13.4 ± 0.2 respectively. The small emission surface, relative to the mattress source, also changes the impact of greater mixing of the air inside the test chamber. Where no effect was found by activating the mixing fans with the mattress as a source, the relative intake fraction for the teddy bear source increased to 1.14 ± 0.07 in the first configuration and decreased to 11.1 ± 0.7 in the second configuration when fans were activated. In the first configuration, the increased mixing prevented the emitted SF_6 from being driven away from the breathing zone by the thermal plume. In contrast, in the second configuration, with the manikin in the prone position, the increased mixing forced more fresh air into the breathing zone.

Rebreathing Experiments

For the rebreathing experiments, the SF_6/nitrogen flow rate was 1 ml/min for the lateral and prone positions. A test was also completed with a flowrate of 0.25 ml/min for the supine position to test the sensitivity of the results to the this parameter.

The thermal plume created by the manikin was sufficient to create close to well mixed conditions in the chamber. This is shown in Figure 6.31. which compares the concentrations measured at the different positions in the chamber to the concentration in the exhaust. This relative uniformity in the chamber, illustrated in Figure 6.31. allowed us to use the SF_6 concentration in the exhaust as being equal to the well mixed background concentration for the calculation of the rebreathing rate. The latter, the fraction of exhaled air re-entering the nose, can therefore be defined as:

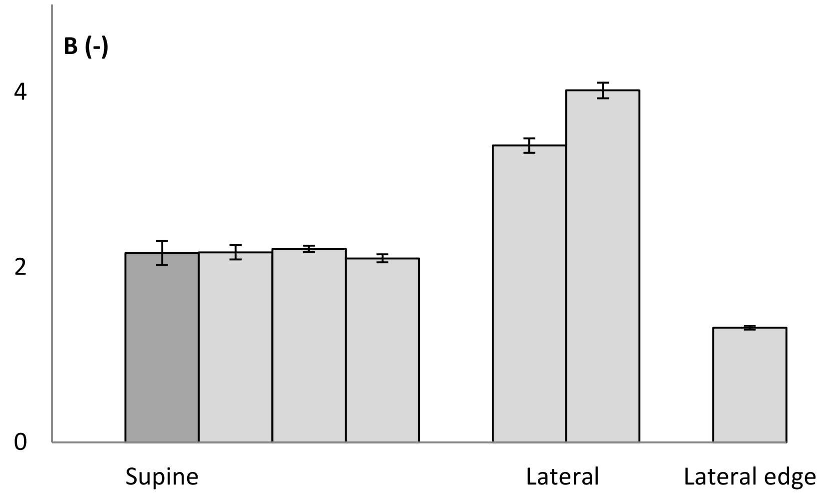

Figure 6.29 Results for different sleep positions with the pillow source. Presented are: mean relative intake fraction (dark bar) measured in supine case with pillow source as well as individually measured relative intake fractions in each repeated experiment (light bars) along with the measured relative intake fraction for the lateral case with pillow source and for the lateral case with the face of the manikin at the edge of the pillow source facing out

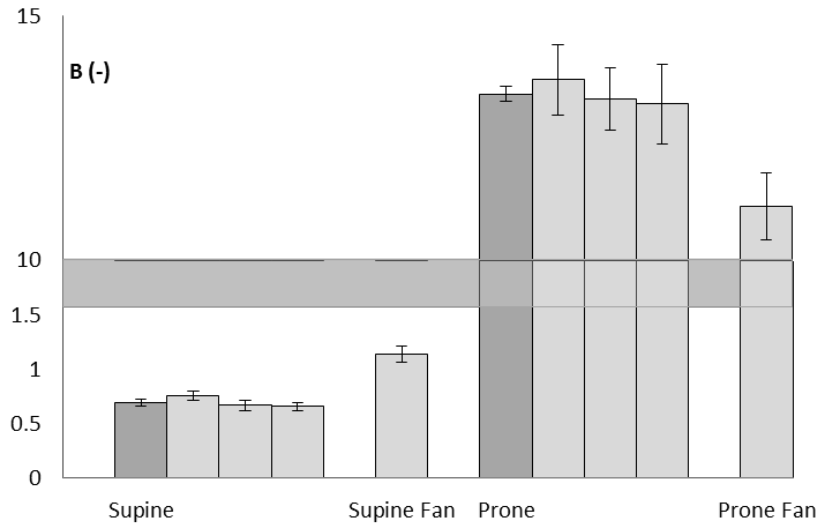

Figure 6.30 Mean (dark) relative intake fraction measured in supine and prone case with teddy bear source as well as individually measured relative intake fractions in each repeated experiment (light) along with the measured relative intake fraction for these same cases with increased mixing

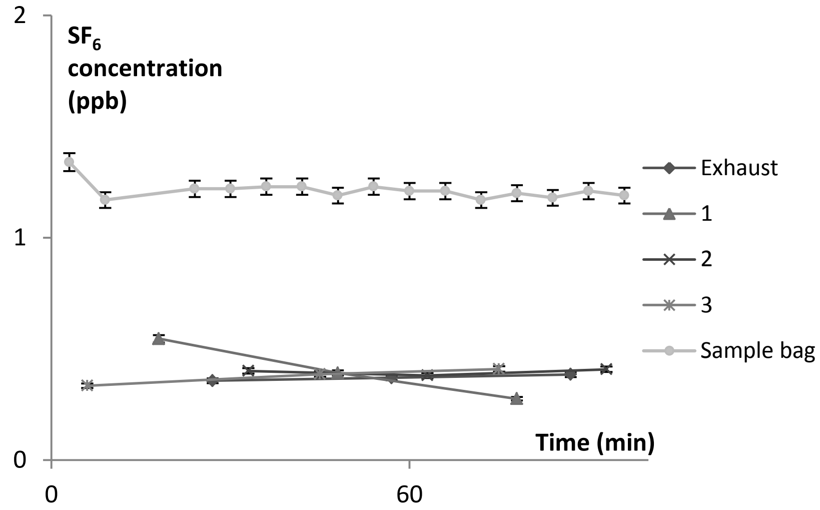

Figure 6.31 SF_6 concentration in the sample bag, return duct and 3 locations in the test chamber for the rebreathing case with the manikin in the supine position

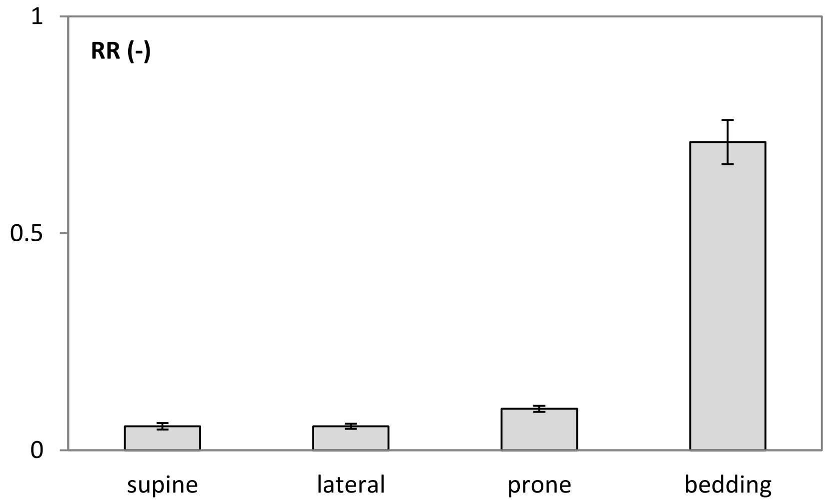

Figure 6.32 Calculated rebreathing rates based on the tests for the supine, lateral, prone positions and for the supine position where the manikin is covered with bedding

$$RR = \frac{C_b - C_{wm}}{C_e - C_{wm}}, \tag{6.3}$$

where

RR	=	rebreathing rate (-)
C_b	=	breathing zone concentration (l/l)
C_e	=	exhaled concentration (l/l)
C_{wm}	=	well mixed concentration, measured at exhaust (l/l)

The calculated rebreathing rate was found to be 5.5 ± 0.7 %, 5.5 ± 0.6 % and 9.5 ± 0.7 % for the supine, lateral and prone positions respectively (Figure 6.32). For the supine position where the face of the manikin was covered by the bedding, the rebreathing fraction was 65.7 ± 5.5 %. No substantial difference was found for the repeated supine test with the higher SF_6 source (5.1 ± 0.4 %). Significance could not be tested since only single tests were completed for these configurations.

6.5.4 Summary of Exposure Experiments

The sleeping environment as a specific context for exposure has received little attention in the published literature. Exposure to pollutants while sleeping may be important due to long exposure time, both absolute and relative, prevalence of specific pollutants and close proximity of sources to the breathing zone. In this chapter, the impact of the proximity of three typical sources, namely mattress, pillow and children's toy on the intake fraction for gaseous pollutants as well as the occurrence of rebreathing was investigated using a breathing thermal manikin. The results demonstrate that human metabolism is a dominant factor in the dilution of pollutants emitted in close proximity to the nose, reducing exposure by 40% compared to a case without metabolic heat output.

The intake fractions for several sleep positions as well as different bedding arrangements are reported for the mattress source. With reference to the supine sleeping position, a 12% and 40% increase in intake fraction was measured in the lateral and prone sleeping position respectively. No increase in rebreathing was found in the lateral position compared to the supine position. In both cases a 5.5% rebreathing rate was found, while in the prone position, rebreathing increased by more than 2/3 to 9.5%. The bedding arrangement was found to have little impact on the intake fraction. Sleeping with the head under the covers, however, increases intake by a factor 24 and results in a rebreathing rate of over 60%, high enough to create hypoxia.

With respect to emissions from smaller sources, the results clearly show that impact of proximity to the breathing zone increases with increasing spatial concentration of the source. The intake fraction for the toy, for example, increased 20-fold when it was positioned at the nose instead of under the armpit.

We can conclude that the particular micro-environment created by sleeping position, bedding arrangement and specific sources is not well represented by well mixed assumptions. Exposure to indoor air pollutants in the sleeping environment is a context that requires further investigation.

6.6 Conclusions

This chapter revisited the design objectives for residential ventilation from the perspective of exposure in the bedrooms. In the first section of the chapter, carbon dioxide measurements in the living room and bedroom of 81 cases demonstrated that the exposure profile in the bedroom, in contrast to that in the other spaces of the dwelling, is characterized by long and continuous exposure at virtually steady state conditions. The measurements also confirmed the observations made in simulations that bedrooms situated on the second floor of the dwelling in mechanical exhaust ventilation systems are often affected by low ventilation rates and poor indoor air quality due to their relative isolation from the exhaust vent holes and their position close to the neutral pressure plane in the dwelling. Both in measurements and simulations, the probability of exposure to carbon dioxide concentrations generally accepted to be an indication of undesirable indoor air quality is up to 10 times higher in the bedrooms than it is in the living room.

Although the results from the first section are cause for some alarm, the validity of perceived air quality based exposure limits for bedrooms may be questioned since the occupants are asleep most of the time spent in the bedroom. The second part of the chapter presented the results from an intervention study that attempted to chart the effects of low ventilation rates on sleeping subjects, using carbon dioxide measurements and sleep-actigraphy. The results indicate that little to no acute effects on sleep can be found. Although subjects reported changes in depth and restfulness of their sleep, similar differences were found in a placebo experiment.

From the previous sections, one could conclude that, although indoor air quality conditions in the bedroom seem less than optimal considering traditional assessment parameters based on perceived air quality and bio-effluents, there seem to be little to no obvious acute repercussions on sleep quality associated with these exposure levels. A further conclusion would then be that flow rates in bedrooms are currently oversized and substantial energy savings could be envisioned by lowering design ventilation rates for bedrooms. This can, however, only be the case under the condition that bio-effluents are the sole type of pollutant to be considered. Therefore, the third section of the chapter analysed the impact of typical sleep micro-environments on the exposure to near field sources of gaseous pollutants and found that these are generally higher than the estimates based on well mixed conditions used in exposure risk assessments for consumer products. This, combined with the fact that the concentrations measured in the first section exceeded the level considered a threshold for health effects in just under 25 % of the occupancy time in the bedrooms, warrants great caution in lowering design flow rates.

Combined, the results from this chapter clearly demonstrate a need for further research in this area to achieve a comprehensive assessment methodology for residential ventilation systems.

7

Conclusions and Perspectives

7.1 Conclusions

Although ventilation is a hot topic, both in popular press and in academia, residential ventilation systems have traditionally received less attention due to their relatively low capacity and complexity. Nevertheless, the large energy saving potential of the residential building sector has made it one of the key targets of climate change mitigation strategies in the European Union. With the rather successful penetration of insulation measures and weatherization, ventilation heat loss now dominates the total heating demand in newly built dwellings, opening a renewed debate on ventilation rates and sizing. Meanwhile, demand control and heat recovery technologies claim to provide substantially better performance than traditional systems. Both have acquired an important but highly competitive share in the market.

This dissertation addressed the performance trade-off between heat loss and indoor air quality inherent to ventilation and focusses on the effectiveness of design strategies, demand control and heat recovery for residential systems with respect to this trade-off. After sketching a context and defining a set of design objectives for residential ventilation in the first chapter, it explored the performance level achieved by the ‘state of the art’, represented by the design strategies included in a series of contemporary standards. In this analysis it focussed on perceived air quality as a performance indicator since, provided the risk posed by other emissions is well managed, human bio-effluents are considered the dominant source in a dwelling.

Subsequently, the effectiveness of the different ventilation strategies was assessed in the third chapter by relating their performance to the Pareto optimal performance. Considering average exposure to carbon dioxide, optimized mechanical supply and exhaust performs only slightly better compared to mechanical exhaust ventilation, while the latter in turn achieves slightly better performance than natural ventilation. These dedicated ventilation systems achieve 30-40% lower exposure at equal ventilation heat loss than uniformly distributed leakage as a means of ventilation.

Using these results as a reference, the energy saving potential of demand controlled ventilation was investigated next. Again starting from the 'state of the art' with an assessment of the performance of systems available on the Belgian market, the analysis expanded to the impact of the local context created by ventilation standards or climate and to Pareto optimal performance. The latter was up to 50% better than that of continuous flow systems.

This was then contrasted to the performance of the main competitive technology, heat recovery ventilation, under different boundary conditions. In the Mediterranean basin, heat recovery ventilation can only be operated profitably in low pressure drop and low fan power systems, while it is advantageous under virtually all tested conditions in the Scandinavian region. In contrast to low fan power, high thermal building performance tends to create unfavourable conditions for heat recovery ventilation.

The last part of the dissertation revisited the design objectives by highlighting the dominance of the exposure in the bedrooms in the assessment of residential indoor air quality as well as the poor validity of available performance indicators and assessment methods in this environment. Sleeping subjects showed virtually no physiological response to changes in indoor air quality and an experimental study on the impact of the sleep micro-environment on exposure clearly demonstrated that exposures in this environment are generally higher than the estimates based on well mixed conditions used in exposure risk assessments for consumer products.

7.2 Perspectives

The presented results from experimental campaigns studying the physiological response of sleeping subjects to changes in indoor air quality and the impact of the sleep micro-environment on exposure clearly demonstrate a need for further research in this area to achieve a comprehensive assessment methodology for residential ventilation systems. Although no single indicator for indoor air quality is apparently available in bedroom environments, it is clear that designing ventilation systems for them is faced with similar trade-off issues as in the other spaces. The general methodology followed in this dissertation can therefore be expanded to take newly established criteria into account.

Other research perspectives include expanding the methodology to take manual interaction of the occupants with continuous flow ventilation systems into account in the assessment of demand controlled systems, including the known adaptation mechanisms in the criteria for perceived indoor air quality and investigating the impact of high energy performance dwellings on the performance of residential ventilation systems.

I see a clear market potential for two different types of systems. The first is a low tech, low pressure drop hybrid ventilation system for mild climates or buildings with moderate ventilation rates and heating demand, while the second is a complex, low pressure drop heat recovery system with decentralised demand control for cold climates or buildings with high ventilation rates in milder climates.

In practice, although the proposed methodology for the assessment of demand controlled ventilation has successfully been adopted by the Belgian ventilation industries, so far the interest has mostly come from mechanical exhaust ventilation manufacturers. These systems have rapidly gained technological maturity and are now generally applied. The first attempts to include demand control in commercially available mechanical and natural ventilation system have been made, but generally, these systems have fallen a bit behind in adopting this technology. It is my hope that the large potential for demand control in both systems demonstrated in this dissertation will be a stimulus for innovation.

Finally, the work presented in this dissertation has both served as a platform to test the limits of the standing residential ventilation standard and raised support for a highly anticipated recast. The methodology presented can be applied to select and test a set of simple design rules, as well as serve to derive performance based design objectives to be included in the new standard.

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