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Ciclo XXI

ENERGY SAVING WITH PERSONALIZED VENTILATION AND COOLING FAN

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PREFACE

The thesis is based on results presented in six papers, three of which are international peer-reviewed conference papers, two have been published in a scientific peer-reviewed journal and one has been submitted (November 2008) to a scientific peer-reviewed journal.

The research work started in January 2006 and the project has been mainly sponsored by the University of Padova (Italy). Secondary funding agents have been the Tsinghua University (P.R. China), the Aldo Gini Foundation (Italy), the Otto Mønsteds Fond (Denmark) and the International Centre for Indoor Environment and Energy at Technical University of Denmark.

The PhD project has been developed in three universities. Location and visiting period are summarized in the table below. Totally 18 months has been spent abroad.

Period (from – to)	Location	Activity
Jan. 2006 - Feb. 2006	University of Padova, Italy	Literature review about PV
Feb. 2006 - Jan. 2007	Tsinghua University, P.R. China	Chinese language course CFD simulation Literature review about PV
Jan. 2007 - Oct. 2007	University of Padova, Italy	Occupation density Energy simulation
Oct. 2007 - May 2008	ICIEE, DTU, Denmark	Energy simulation Laboratory experiments
May 2008 - Dec. 2008	University of Padova, Italy	Analysis of experimental data and writing of the thesis

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I wish to thank Michele Bretini, Franco Fiorani, Lisbeth L. Schack, Snjezana Skocajic, and Xu Yan for the administrative support.

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Padova, December 2008
Stefano Schiavon

"It's probably equally difficult to solve an unimportant problem as it is to solve an important problem, so you might as well pick an important problem" *Daniel Koshland*

SUMMARY

Indoor environmental quality substantially influences health, comfort and productivity. The cost related to a poor indoor environment is high. Numerous field studies have documented substantial rates of dissatisfaction with the indoor environment in many buildings, therefore an increment of the actual indoor environmental quality is necessary. Global warming of the climate system is now unequivocal and it has had a discernible influence on many physical and biological systems, therefore, it is needed to reduce the greenhouse gases emission. On this challenge, an important role is played by the building sector. Technological solutions able to improve the indoor environment and to reduce the energy consumption simultaneously should be developed.

In warm environments elevated air movement is a widely used strategy for cooling of occupants. Increasing the air movement let the opportunity to set the maximum permissible room temperatures to higher values. According to many authors this solution leads to substantial energy savings. In the present international indoor climate standards a relationship is present between the air speed and the allowed increment in operative temperature. The air movement increase can be produced by several devices as cooling fans (ceiling, floor standing, tower and table fans) or Personalized Ventilation (PV) systems.

The cooling fans ability to cool the human body is limited because they operate under isothermal conditions. Cooling fans may save energy but they do not improve the indoor environmental quality. Appearance, power consumption and price are the main parameters considered when purchasing cooling fans while their cooling capacity and efficiency of energy use are unknown. Comparison of the performance of cooling fans regarding cooling capacity and energy consumption is important for their application in practice.

The personalized ventilation is an individually controlled micro-environmental system that provides clean air close to occupants. Numerous studies show that PV in comparison with traditional mechanical ventilation system may improve health, inhaled air quality, thermal comfort, and self-estimated productivity and it may decrease the risk of airborne transmission of infectious agents. Little is known about its energy consumption. Personalized ventilation systems have better performance than cooling fans with regard to thermal comfort since they may operate under non-isothermal conditions, i.e. the supplied air can be cooled below the room air temperature in addition to increased velocity. The PV system affects the pollutant concentration and the thermal conditions mostly in the microenvironment at the workstation. Therefore, occupant's exposure to pollutant and his/her thermal comfort depend on the ratio of time occupant stays at the workstation over total time he/she stays in the room.

The main objectives of the present work were to study, by means of computer simulation, the energy saving when providing occupants with thermal comfort with increased air movement at elevated room temperature, the energy consumption of a personalized ventilation system and energy saving strategies which can be used to control a PV system, and to develop and to test in laboratory, an index for evaluating the cooling fan efficiency. An additional objective of the study was to develop and to test an index for assessing the air quality improvements in rooms with non-homogeneous contaminant distributions (e.g. with personalized ventilation) taking into account the occupant location pattern.

The potential saving of cooling energy by elevated air speed, which can offset the impact of increased room air temperature on occupants' thermal comfort, was quantified by means of simulations using EnergyPlus software. Fifty-four cases covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem, Athens), three indoor environment categories and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated. Cooling energy savings in the range of 17-48% and a reduction of the maximum cooling power in the range 10-28% have been obtained. The results reveal that the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature. Under the assumptions of this work, energy saving may not be achieved with the methods for air speed increase, such as ceiling, standing, tower and desk fans widely used today when the power consumption of the fan is higher than 20 W. From the results of the simulation it can be deduced that knowing the cooling capacity and the energy consumption of the fan is important.

A new index has been developed, named "cooling fan efficiency index", defined as the ratio between the cooling effect generated by a fan and its power consumption. The cooling effect is calculated as the difference of manikin-based equivalent temperature measured with and without the fan in operation. The cooling fan efficiency can be a useful index for comparison performance of fans, for costumers, fan designers/manufacturers, policymakers, HVAC designers and building managers. The index was determined experimentally for a ceiling fan, a desk fan, a standing fan and a tower fan in a real office at three room air temperatures and at different fan velocity levels. The results revealed that the index is sensitive enough to identify differences in the performance of the cooling devices. The cooling fan efficiency index of the four fans differed substantially. The whole-body cooling effect and the local cooling effect for body segments caused by the fans also differed and were strongly non-uniform. The desk fan had a significantly (p-value<0.01) higher efficiency than the other three fans tested. A standard method for measuring the cooling fan efficiency index should be developed.

The cooling fans generate a non-uniform velocity field around occupants which cannot be described with a single value. This makes the recommendation in the standards for elevated velocity in warm environments difficult to use in practice. The present thermal comfort standards need to be revised to better address the issue of thermal comfort in warm environments.

The energy consumption of a PV system installed in a high quality standard Scandinavian building located in a cold climate have been studied by means of simulations with IDA-ICE software. An optimization algorithm was used to determine the optimal supply air temperature. The effectiveness of the following energy saving strategies have been studied: reducing the outdoor airflow rate due to the higher ventilation effectiveness of PV, expanding the room temperature comfort limits by taking advantage of PV's ability to create a controlled thermal microenvironment and supplying the personalized air only when the occupant is present at the desk. The results showed that the control strategy of the supplied personalized air temperature has a marked influence on energy consumption. The energy consumption with personalized ventilation may increase substantially (between 60% and 270%) compared to mixing ventilation alone if energy-saving strategies are not applied. Among the studied energy-saving strategies the most effective way of saving energy is to increase the maximum permissible room temperature (saving up to 60% compared to the mixing ventilation may be achieved) but it can be applied only in offices where occupants spend most of their time at the desk. Reducing the airflow rate does not always imply a reduction of energy consumption because the outdoor air may have a free cooling effect. Supplying the personalized air only when the occupant is at the desk is not an effective energy-saving strategy. The best supply air temperature control strategy is to provide air constantly at 20°C, i.e. the minimum permissible supply temperature.

A further index has been developed in this work, named "occupant normalized concentration", which makes it possible to assess more realistically occupant's exposure in a room characterized by a non-uniform pollution distribution. The index can be used to evaluate the average pollutant exposure as function of the pollutant distribution in a space and of the occupant activity, and it can be used to compare and quantify the variation in terms of inhaled pollution by occupant in a room with PV in conjunction with a total-volume ventilation system. The results of the application of the index to data collected during full-scale room measurements showed that it can be used at the design stage for assessment the benefit of PV when applied in practice for office buildings with different occupation patterns. It is demonstrated that displacement ventilation alone was able to provide the occupant with better inhaled air quality than displacement ventilation in conjunction with PV when the occupant stay less than 50% of the office time at the desk. These analyses are performed under steady state conditions, i.e. without disturbance of the displacement pattern due to occupants' walking.

SOMMARIO

La qualità dell'ambiente interno influenza significativamente salute, comfort e produttività e il costo economico legato ad un ambiente di bassa qualità è elevato. Numerosi studi hanno documentato che la percentuale di persone non soddisfatta dall'ambiente interno che occupa è alta, e risulta quindi necessario sviluppare e utilizzare tecnologie in grado di aumentare la qualità dell'ambiente interno. Poiché il riscaldamento globale è inequivocabile e influenza negativamente molti sistemi fisici e biologici occorre ridurre l'emissione di gas serra. In questa sfida, il settore dell'edilizia gioca un ruolo chiave. È importante sviluppare soluzioni tecnologiche che possano, allo stesso tempo, ridurre i consumi energetici e incrementare la qualità ambientale.

In ambienti con temperature relativamente elevate (maggiori di 24-26°C) è possibile ottenere una condizione di comfort termico aumentando la velocità dell'aria nell'ambiente. L'incremento della velocità dell'aria permette di aumentare la temperatura massima accettabile e, secondo molti autori, questa soluzione permette di risparmiare energia. Negli attuali standard internazionali per il comfort termico esiste una relazione tra la velocità dell'aria e l'incremento del limite massimo di temperatura. Il movimento dell'aria può essere generato da molte apparecchi quali, ad esempio, i ventilatori per il raffrescamento (*cooling fans:* ventilatori da tavolo, a pavimento, a soffitto o a torre) e i sistemi di ventilazione personalizzata.

La capacità di raffrescamento del corpo umano generata dai ventilatori è limitata poiché questi operano in condizioni isoterme. Inoltre, i ventilatori possono far risparmiare energia ma non migliorano la qualità dell'ambiente interno. Quando vengono acquistati, i principali parametri presi in considerazione dal cliente sono l'estetica, il costo e, più raramente, la potenza assorbita. La capacità di raffrescamento e l'efficienza con la quale utilizzano l'energia sono parametri non noti anche se potrebbero essere utili per la scelta e l'utilizzo del ventilatore.

La ventilazione personalizzata è un sistema per il controllo individuale del micro-ambiente e immette aria esterna in prossimità della persona. Un elevato numero di studi ha dimostrato che la ventilazione personalizzata, rispetto a un sistema tradizionale di ventilazione, migliora la salute, la qualità dell'aria inalata, il comfort termico, le prestazioni, e può ridurre il rischio di diffusione delle malattie trasmesse per via aerea. Le conoscenze sui consumi energetici della ventilazione personalizzata sono limitate. La ventilazione personalizzata ha prestazioni migliori dei ventilatori per quanto riguarda la capacità di raffrescare le persone poiché può operare in condizioni non isoterme, cioè l'aria viene immessa a una temperatura inferiore a quella della stanza. La ventilazione personalizzata influenza il comfort termico e la qualità dell'aria principalmente in prossimità della postazione di lavoro dove è installata, e quindi, l'esposizione della persona agli inquinanti e il suo comfort termico dipendono fortemente dal rapporto tra la quantità di tempo che l'occupante passa alla scrivania rispetto al tempo totale speso nell'ambiente ventilato.

I principali obiettivi della ricerca sono stati: a) studiare, attraverso delle simulazioni computazionali, il risparmio energetico dovuto all'incremento del movimento dell'aria, il consumo energetico della ventilazione personalizzata e le strategie di risparmio energetico che possono essere applicate per il suo controllo; b) sviluppare e verificare in laboratorio un indice per la valutazione dell'efficienza dei ventilatori utilizzati per il raffrescamento.
Un altro obiettivo della ricerca è stato lo sviluppo, e l'applicazione, di un indice per la valutazione dei miglioramenti della qualità dell'aria in ambienti con una distribuzione degli inquinanti non omogenea.

Il potenziale di risparmio di energia per il raffrescamento dovuto all'aumento delle temperature massime consentite in una stanza è stato studiato e quantificato utilizzando delle simulazioni energetiche (EnergyPlus). Sono stati simulati cinquantaquattro casi che comprendono sei città (Helsinki, Berlino, Bordeaux, Roma, Gerusalemme e Atene), tre categorie di qualità ambientale e tre livelli di velocità dell'aria (<0,2, 0,5 e 0,8 m/s). Si è ottenuta una riduzione dell'energia per il raffrescamento, variabile tra il 17 e il 48%, e della potenza frigorifera variabile tra il 10 e il 28%. I risultati hanno mostrato che la potenza del ventilatore utilizzato per il raffrescamento è un fattore critico per l'ottenimento di un risparmio energetico. Secondo le assunzioni di questa ricerca, non è possibile ottenere un risparmio energetico se vengono utilizzati gli attuali ventilatori come quelli a soffitto, a pavimento, a torre o a tavolo se la potenza del ventilatore è superiore a 20 W. Dai risultati della simulazione si può dedurre che è utile conoscere la capacità di raffrescamento del ventilatore e il suo consumo energetico.

È stato pertanto sviluppato un nuovo indice chiamato "*Cooling Fan Efficiency*", *CFE*, (efficienza del ventilatore per il raffrescamento), definito come rapporto tra l'effetto di raffrescamento generato dal movimento dell'aria e la potenza elettrica assorbita dal ventilatore. L'effetto di raffrescamento viene misurato come differenza di temperatura equivalente di un manichino termico che simula, in modo codificato, il sistema di termoregolazione del corpo umano. *CFE* è un indice utile per la comparazione delle prestazioni dei ventilatori per i clienti, per i progettisti dei ventilatori, per i progettisti dei sistemi termotecnici, per i produttori, per i responsabili della gestione degli edifici e per il legislatore. L'indice è stato determinato sperimentalmente per un ventilatore a soffitto, uno a pavimento, uno a torre e uno da tavolo in un ufficio per tre livelli di temperatura e per vari livelli di velocità. I risultati hanno dimostrato che l'indice è sufficientemente sensibile per identificare le differenze di prestazioni dei diversi ventilatori. L'indice varia significativamente per i quattro ventilatori studiati. Gli effetti di raffrescamento sull'intero corpo del manichino, e quelli sulle singole parti, variano notevolmente e sono fortemente non uniformi. Il ventilatore da tavolo ha un'efficienza significativamente superiore, statisticamente dimostrata (*P-value*<0,01), a quella degli altri tre ventilatori. Sarà necessario sviluppare una procedura standard di misurazione dell'indice.

I ventilatori per il raffrescamento generano un campo di velocità non uniforme che non può essere descritto da un unico valore di velocità. Ciò rende la relazione tra velocità dell'aria e incremento della temperatura massima ammissibile, presente negli standard internazionali, non sufficientemente precisa per essere utilizzata in pratica. Gli attuali standard per il comfort termico dovrebbero essere rivisti per meglio affrontare la possibilità legata al risparmio energetico dovuto all'incremento delle temperature massime ammissibili.

I consumi energetici dovuti alla ventilazione personalizzata installata in dodici postazioni di lavoro in un edificio scandinavo per uffici di elevata qualità sono stati studiati attraverso delle simulazioni energetiche (IDA-ICE). Un algoritmo di ottimizzazione è stato utilizzato per determinare il profilo di temperature di immissione che minimizza il consumo energetico. Sono state anche studiate le seguenti strategie per la riduzione dei consumi energetici: riduzione della portata di rinnovo grazie alla maggiore efficienza di ventilazione tipica della ventilazione personalizzata; espansione dei limiti massimi di temperatura grazie alla capacità del sistema di controllare il micro-ambiente attorno alla persona; e immissione della portata di rinnovo solo quando la persona è seduta alla scrivania. I risultati hanno dimostrato che la strategia di controllo della temperatura di immissione della ventilazione personalizzata ha un'influenza notevole sui consumi energetici. Rispetto alla ventilazione a miscelazione, il consumo energetico dovuto alla ventilazione personalizzata può aumentare in modo considerevole (tra il 60 e il 270% circa) se non vengono applicate delle strategie per il risparmio energetico. Tra le strategie per il risparmio energetico quella più efficace è quella dell'espansione dei limiti massimi della temperatura. Con questa soluzione è possibile risparmiare fino al 60% rispetto alla ventilazione a miscelazione: questa strategia può però essere utilizzata solo se gli occupanti spendono la maggior parte del tempo alla scrivania. La riduzione della portata di rinnovo non implica sempre una riduzione dei consumi energetici poiché l'aria esterna può avere anche un effetto di raffrescamento gratuito. Immettere l'aria solo quando la persona è seduto alla scrivania si è rivelata una strategia non efficace. Il miglior controllo della temperatura dell'aria di immissione consiste nel fornire costantemente l'aria a 20°C, che è il valore minimo permesso poiché l'aria viene diffusa in ambiente molto vicino alla persona.

In questo lavoro è stato anche sviluppato un indice, chiamato concentrazione normalizzata rispetto all'occupante (*occupant normalized concentration*), che permette di valutare in modo più realistico l'esposizione dell'occupante agli inquinanti in ambienti caratterizzati da una distribuzione non uniforme dei contaminanti, come accade negli ambienti dove la ventilazione personalizzata viene utilizzata. L'indice può valutare l'esposizione media dell'occupante agli inquinanti in funzione della loro distribuzione spaziale e della sua ubicazione temporale, e può essere utilizzato per comparare vari sistemi di ventilazione in termini di variazione di inquinanti inalati. L'indice è stato applicato a misure ottenute in laboratorio. I risultati hanno dimostrato che l'indice può essere impiegato nella fase di progettazione come utile indicatore dell'efficacia della ventilazione personalizzata, in funzione dei possibili comportamenti degli occupanti. Inoltre, per la prima volta, è stato mostrato che per gli inquinanti associati a sorgenti di calore, la ventilazione a dislocamento può fornire una qualità dell'aria migliore della ventilazione personalizzata quando l'occupante rimane alla scrivania per meno del 50% del tempo che trascorre in ufficio.

TABLE OF CONTENTS

LIST OF PUBLICATIONS

The thesis is based on the work contained in the following papers.

A. Schiavon, S., and Melikov, A. 2008. Energy saving and improved comfort by increasing air movement. *Energy and Buildings* Volume 40, Issue 10, Pages 1954-1960 doi:10.1016/j.enbuild.2008.05.001

B. Schiavon, S. and Melikov, A. 2008. Energy saving strategies with personalized ventilation in cold climates. *Energy and Buildings* doi: 10.1016/j.enbuild.2008.11.018

C. Schiavon, S. and Melikov, A. 2008. Introduction to the Cooling Fan Efficiency Index. (Submitted to *HVAC&R Research*)

D. Schiavon, S., and Melikov, A. 2008. Energy saving and improved comfort by increasing air movement. *Proceedings of International Conference Indoor Air 2008*. Copenhagen, Denmark.

E. Schiavon, S. and Melikov, A. 2008. Energy analysis of a personalized ventilation system in a cold climate: influence of the supplied air temperature. *Proceedings of 29th International Conference AIVC 2008.* Kyoto, Japan. (Best Poster Award)

F. Schiavon, S., Melikov, A.K., Cermak, C., De Carli, M., and Li X. 2007. An Index for Evaluation of Air Quality Improvement in Rooms with Personalized Ventilation Based on Occupied Density and Normalized Concentration. *Proceedings of International Conference Roomvent 2007*. Helsinki, Finland.

OTHER PUBLICATIONS

During my PhD project I worked on other papers that are not included in this thesis but they are listed below.

Papers in Conference Proceedings

- Schiavon, S. and Melikov, A. Evaluation of the Cooling Fan Efficiency index for a desk fan and a computer fan. *Proceedings of International Conference Roomvent 2009*. Busan, South Korea. (abstract accepted, paper submitted)
- Schiavon, S., and Zecchin, R. 2008. Indoor Climate and Productivity in Offices. *Proceeding of the International Conference Expocomfort 2008.* Milan 12-13 March.
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- Schiavon, S. 2006. Design methods for displacement ventilation: Critical review. *Proceeding of Chinese National HVAC&R Conference*, Hefei, Anhui, China 31/10-4/11/2006. (In Chinese)

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- Schiavon, S., and Zecchin, R. 2009. Clima Interno e Produttività negli Uffici. (*Indoor Climate and Productivity in Office Building*) CDA. January 2009. (in press)
- Naydenov, K., Schiavon, S., and Zecchin, R. 2008. Quanto dobbiamo ventilare gli edifici residenziali? (*How much we should ventilate the residential building?*) (submitted to CDA)
- Bekő, G., and Schiavon, S. 2008. I filtri possono inquinare l'aria? – Parte 2. (*May the filters pollute the indoor air?*). *L'Installatore Italiano*, n. 8 settembre 2008, pp 38-41.
- Bekő, G., and Schiavon,. S. 2008. I filtri possono inquinare l'aria? – Parte 1. (*May the filters pollute the indoor air?*). *L'Installatore Italiano*, n. 7 luglio 2008, pp 31-35.
- Schiavon, S. and Zecchin, R. 2007. Cambiamenti climatici: mitigazione dei cambiamenti climatici. (*Climate Change: Mitigation of Climate Change*). *CDA*, n11 Dicember 2007, pp 42-46.
- Schiavon, S. and Zecchin, R. 2007. Cambiamenti climatici: impatti, adattamento e vulnerabilità. (*Climate Change: Impacts, Adaptation and Vulnerability*). *CDA*, n. 10 November 2007, pp 30-33.
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Translated books

- Nielsen PV, Allard F, Awbi HB, Davidson L, Schälin A. Computational Fluid Dynamics.*REHVA Guidebook no 10*. 2007 (Translation for AICARR)
- Wargocki P., Seppänen O., Andersson J., Boerstra A., Clements-Croome D., Fitzner K., Olaf Hanssen S. Indoor Climate and Productivity in Offices. How to integrate productivity in life-cycle cost analysis of building services. *REHVA Guidebook no 6*. 2006 (Translation for AICARR)

NOMENCLATURE AND TERMS

Terms, symbols, and units when it was possible were based on CEN EN 12792-2003 "Ventilation for buildings – Terminology, symbols and units".

Abbreviations

ACE	Air Change Effectiveness index
AHU	Air Handling Unit
ATD	Air Terminal Device
CDD_{18}	Cooling Degree Days with a base temperature of 18°C
CF	Ceiling Fan
CFD	Computational Fluid Dynamic
CFE	Cooling Fan Efficiency index, [°C/W]
COP	Coefficient Of Performance [-]
DF	Desk Fan
DV	Displacement Ventilation
HVAC	Heating Ventilating and Air Conditioning
IAQ	Indoor Air Quality
ICS	Individually Controlled System
MV	Mixing Ventilation
PECS	Personally Environmental Control System
PV	Personalized Ventilation
SBS	Sick Building Syndrome
ST	Standing Fan
TACS	Task Ambient Conditioning Systems
TC	Thermal Comfort
TF	Tower Fan
UFAD	Under-Floor Air Distribution

Symbols

Latin letters

$E^{v=i}{}_{N,C}$	Energy need for cooling ($E_{N,C}$) obtained when the air velocity is i ≤0.2 or i =0.5 or i =0.8 m/s [kWh/(m^{2}y)]
AHU Cooling	Energy that is extract by the AHU from the outdoor airflow rate in one year [kWh/(m^{2}y)]
AHU Heating	Energy that is supplied by AHU to the outdoor airflow rate in one year [kWh/(m^{2}y)]
C	Occupant normalized concentration [-]
c	Contaminant concentration [mg/m^3, ppm]
$\dot{c}$	Normalized contaminant concentration [-]
c_E	Contaminant concentration in the exhaust air [mg/m^3, ppm]
c_i	Contaminant concentration in the breathing zone [mg/m^3, ppm]
C_{PVnp}	Occupant normalized concentration for the non-protected occupant when the total-volume system is working in conjunction with a personalized ventilation system
C_{PVp}	Occupant normalized concentration for the protected occupant when the total-volume system is working in conjunction with a personalized ventilation system

c_S	Contaminant concentration in the supply air [mg/m^3, ppm]
$\dot{c}_S$	Normalized contaminant concentration in the air inhaled by the occupant standing in the background area of the room [-]
C_{TV}	Occupant normalized concentration for the total volume ventilation alone
$\dot{c}_W$	Normalized contaminant concentration in the air inhaled by the occupant at the workstation [-]
$E_{el,Cool}$	Electrical energy consumed by the chiller [kWh/(m^{2}y)]
$E_{el,Fan}$	Electrical energy consumed by the fan [kWh/(m^{2}y)]
$E_{el,Net}$	Net electrical energy saved [kWh/(m^{2}y)]
ELA	Equivalent Leakage Area [m^2]
$E_{N,C}$	Energy need for cooling [kWh/(m^{2}y)]
$h_{cal,i}$	Dry heat transfer coefficient of *i*-th segment of the manikin, determined during calibration, [W/(°Cm^2)]
h_i	Annual number of hours that the fan is operating for increasing the air velocity. It is calculated for an air velocity of 0.5 m/s ($h_{0.5}$) and 0.8 m/s ($h_{0.8}$)
h_{tot}	The total occupant working hours [h]
OD	Occupied Density
ODB	Background Occupied Density index
ODW	Workstation Occupied Density index
P_f	Fan air power, [W]
$Q_{t,i}$	Sensible heat loss of *i*-th segment, [W/m^2]
q_v	Personalized airflow rate per person [l/s pers.]
Room Cooling	Energy that is extracted by the fan coil units from the room in one year [kWh/(m^{2}y)]
Room Heating	Energy that is supplied by the fan coil units to the room in one year [kWh/(m^{2}y)]
SF_6	Sulfur hexafluoride
teq	Whole-body manikin based equivalent temperature [°C]
$t_{eq,i}$	Segmental equivalent temperature [°C]
$t_{sk,i}$	Skin temperature of *i*-th segment of the manikin [°C]
U-value	Overall heat transfer coefficient [W/m^2 K]

Greek letters

Δt_{eq}	Whole-body cooling effect [°C]
$\Delta t_{eq,i}$	Segmental cooling effect [°C]
ε_V	Ventilation effectiveness [-]
η	Energy losses from emission, distribution and storage for cooling. It is the ratio between the energy need for cooling and the thermal energy that the chiller has to produce [-]
θ or t	Temperature [°C]
θ_{IDA}	Indoor air temperature [°C]
θ_{mr}	Mean radiant temperature [°C]
θ_{ODA}	Outdoor air temperature [°C]
θ_{op}	Operative temperature [°C]
θ_{SUP}	Supply air temperature [°C]
θ_{UP}	Upper room operative temperature limit [°C]
λ	Thermal conductivity [W/(Km)]
τ	Time [h]
τ_S	Time the occupant spends standing in the office-room but no at to the workstation [h]
τ_{TOT}	Total time the occupant stays in the ventilated room [h]
τ_W	Time the occupant spends at the workstation [h]

1 INTRODUCTION

1.1 Indoor environment: the need of a paradigm shift

The indoor environment is an environment within a building or an enclosed space. Indoor Environmental Quality (IEQ) substantially influences health, comfort and productivity. A deteriorated indoor environment increases sick building syndrome symptoms, respiratory illnesses, sick leave and the risk of infection transmission and reduces comfort and productivity.
Wargocki et al. (2002), based on a literature review, concluded that there is a strong association between the level of ventilation and comfort (perceived air quality), health (sick building syndrome (SBS) symptoms, inflammation, infections, asthma, allergy, short-term sick leave) and productivity.

The cost related to a poor indoor environment is high. Improvement of the indoor environment is economically efficient when health and productivity are taken into account (Wargocki and Djukanovic 2005, Fisk et al. 2003, Smolander at al. 2003). Seppänen (1999) showed that the cost of deteriorated indoor environments is higher than building heating costs. Macro–economic estimates indicate that large economic benefits are possible from improved IEQ (Fisk 2000, Mendell et al. 2002).

Numerous field studies (Bluyssen et al., 1996; Mendell, 1993) have documented substantial rates of dissatisfaction with the indoor environment in many buildings. At the same time the studies show that meeting today's standards does not prevent widespread complaints of poor air quality and frequent building-related symptoms.

For these reasons Fanger (2005) affirmed that it is needed a paradigm shift in the field of Heating Ventilation and Air-Conditioning (HVAC). The aim is to provide an indoor environment where even the most sensitive persons find the environment acceptable, and the majority would found it pleasant. According to Fanger (2005) to decrease to a negligible number the dissatisfied by the indoor air quality (IAQ), an improvement would require an increase of IAQ by one or two orders of magnitude. To reach this aim he proposed to use the following methods:

- source control
- air cleaning
- cool and dry air
- personalized ventilation.

Large individual differences exist between occupants in regard to physiological and psychological response, clothing insulation, activity, air temperature and air movement preference, etc., therefore a unique set of thermal parameters (i.e. air temperature, mean radiant temperature, relative humidity, air velocity) is not able to fulfill the requirements of everybody. Thermal conditions acceptable for most occupants in rooms may be achieved by providing each occupant with the possibility to generate and control his/her own preferred microenvironment.

1.2 Climate change: the need of saving energy

It is not rational to provide occupants with an excellent indoor environment by consuming a huge amount of energy. According to the 2007 report of Intergovernmental Panel on Climate Change, warming of the climate system is unequivocal, as it is now evident from observations of increases in global average air and ocean temperatures, widespread melting of snow and ice, and rising global average sea level. Global atmospheric concentrations of carbon dioxide, methane and nitrous oxide have increased markedly as a result of human activities since 1750 and now far exceed pre-industrial values (IPCC WG1, 2007a). A global assessment of data since 1970 has shown that it is likely (66 to 90% probability) that anthropogenic warming has had a discernible influence on many physical and biological systems (IPCC WG2, 2007b). Moreover, climate change will have marked impacts on the indoor environment. Examples of these impacts range from greater use of air conditioning; increased risk of mold from flooding; increased exposure to ozone indoors; increased pressure to reduce ventilation rates; increased risk from vector-borne diseases; increased risk of pesticide exposure (Girman et al. 2008). Therefore, it is needed to reduce the greenhouse gases emission. On this challenge, an important role is played by the building sector. In fact, according to IPCC the residential and commercial sectors have the highest global potential to reduce emissions among all sectors studied in the report. Moreover, energy efficiency options for new and existing buildings could considerably reduce CO_2 emissions with net economic benefit. Energy efficient buildings, while limiting the growth of CO_2 emissions, can also improve indoor and outdoor air quality, improve social welfare and enhance energy security (IPCC WG3, 2007c).

1.3 Cooling people with elevated air movement

In a warm environment elevated air movement is a widely used strategy for cooling of occupants. In the present international indoor climate standards (ASHRAE 55-2004, ISO 7730-2005 and CEN EN 15251-2007) the operative temperature comfort limits are based on an air speed limit of 0.20 m/s. However, according to the standards, elevated air speed can offset the indoor temperature rise and provide occupants with thermal comfort. An air speed increase is necessary in order to maintain the heat exchange between the human body and the environment, this being a prerequisite for thermal comfort. The relationship between the air speed and the allowed increment in operative temperature, as included in the present standards (ASHRAE 55-2004 and ISO 7730-2005) is shown in Figure 1. The recommended speed increase depends not only on the air temperature but also on the difference between mean radiant temperature (θ_{mr}) and air temperature (θ_{IDA}) (see Figure 1). When the mean radiant temperature is lower than the air temperature, the elevated air speed is less effective for increasing the heat loss from the body. Conversely, elevated air velocity is more effective for increasing the heat loss when the mean radiant temperature is higher than the air temperature. The relationship is based on a theoretical calculation; however, it has been verified in human subject experiments (Toftum et al. 2003). Individual differences exist between people with regard to the preferred air speed (Toftum et al. 2003 and Melikov et al. 1994 a and b). Therefore, the standards require personal control over the speed. Thus it may not be appropriate to offset a temperature increase by increasing the air speed within a centrally-controlled air system (Olesen and Parsons 2002).

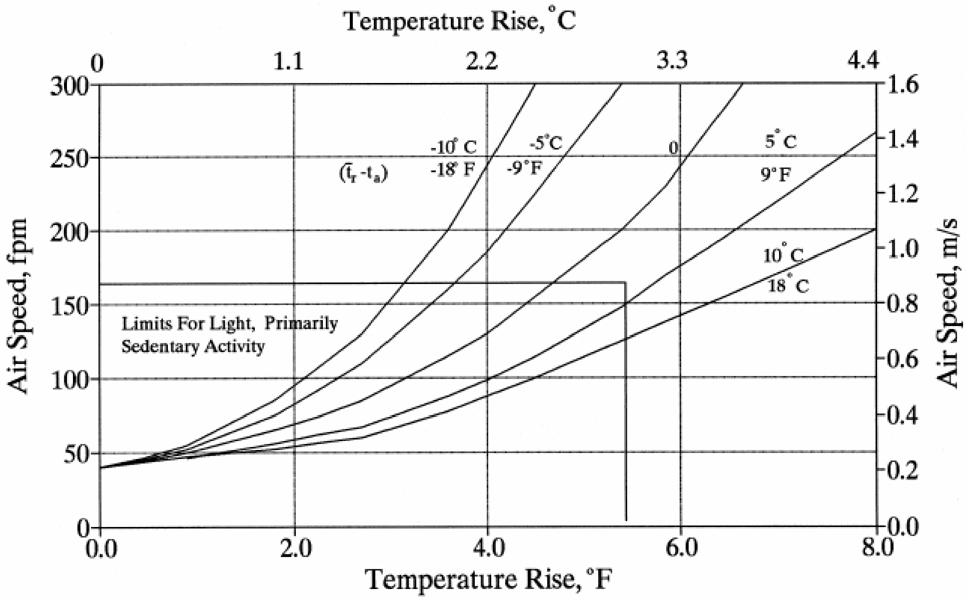

Figure 1. Air speed required to offset increased temperature. (Figure 5.2.3 from ASHRAE 55-2004).

The conditions defined in Figure 1 may be applied only to a lightly clothed person with a clothing insulation between 0.5 clo and 0.7 clo (0.08 to 0.1 m^2K/W) who is engaged in near sedentary physical activity with metabolic rates between 1.0 met and 1.3 met (58.15 to 75.6 W/m^2). The effect of elevated speed on the heat loss from the human body increases at high activity and lighter clothing (ASHRAE 55-2004). Moreover, the increase in operative temperature cannot be higher than 3.0°C above the values for the comfort zone and the elevated air speed must not be higher than 0.8 m/s.

The possibility of increasing the upper operative temperature limit at elevated velocity may reduce the energy consumption without negatively affecting occupants' thermal comfort (Sekhar 1995; Olesen and Brager 2004; Aynsley 2005, Atthajariyakul and Lertsatittanakorn 2008).

The air movement increase can be produced by several devices such as cooling fans (ceiling, floor standing, tower and table fans), furniture-installed personalized ventilation, body-attached ventilation devices and, under certain conditions, operable windows. The underfloor air distribution system, which is one of the total volume ventilation principles used in practice, also allows for increase or decrease of the velocity close to workplaces. In this work the focus was on the personalized ventilation system and on cooling fans. These systems are described in the following paragraphs.

The cooling capacity of cooling fans is limited because they operate under isothermal conditions, i.e. the cooling of the body is a result of increased velocity only. The use of cooling fans in practice is easy and does not require special installations. The personalized ventilation systems (Melikov 2004) and the task-ambient conditioning systems (Arens et al. 1991) perform better with regard to thermal comfort since they may operate under non-isothermal conditions, i.e. the supplied air can be cooled below the room air temperature in addition to elevated velocity. Appearance, power consumption and price are the main parameters considered when purchasing cooling fans, while their cooling capacity and efficiency of energy use are unknown. Other factors such as ergonomics, control options, etc., are also important. Comparison of the performance of cooling fans from the point of view of cooling capacity and energy consumption is important for their application in practice.

1.4 Cooling fans

Cooling fan is a general term used for all the devices that increase the air movement in order to cool an object. Typical examples are the computer fans. In this work this term is used just for fan used for cooling humans. There are a big variety of these kinds of fans. They change in shape, appearance, price, power consumption, nominal airflow, controllability, applicability, etc. On the market the most common cooling fans for humans are the ceiling fan, the standing (also named pedestal) fan, the tower fan and the desk fan. These fans are well known, and their size and speeds have been standardized by the I.E.C. (International Electrotechnical Commission). These fans may cool just one person (e.g. desk fan) or a limited number of persons (e.g. standing fan or ceiling fan). They may cool people only in some specific part of the body (desk fan) or generate more homogeneous cooling effect (ceiling fan). The air velocity field generated by the cooling fan may vary significantly. Usually it is measured at three diameters far from the impeller (Daly 1992) for the standing and table fan and 1.5 m below the impeller for the ceiling fan. For economical reasons, cooling fans have just step-wise velocity control. Many cooling fans are equipped with oscillation mechanisms to swing the jet stream through an arc of 60° to 120° because gusting air is perceived more pleasant that constant air.

Appearance, power consumption and price are the main parameters considered when cooling fans are purchased while their cooling capacity and efficiency of energy use are unknown. The main advantages of these types of cooling devise are the low price and easy installation procedure. A more detailed description of these fans can be found in Bleier (1998).

1.5 Personalized ventilation

Personally Environmental Control System (PECS) is an individually controlled micro-environmental system that improves thermal comfort and/or air quality to suit the needs of the person (i.e. customize the personal micro-environment). Another name for PECS is Individually Controlled System (ICS). PECS can be divided in two subgroups depending on the main objective of the system. When the principal aim is to increase the inhaled air quality then a PECS is commonly named a Personalized Ventilation (PV) system. Task Ambient Conditioning (TAC) is the name of the system when the main aim is to increase the thermal comfort. Other systems that seek to achieve the same objectives of PECS as collars, masks, breathing tubes, special clothing, occupant-controlled rooms, diffusers, radiation panels etc. may also be considered PECS. This thesis will focus on personalized ventilation system.

PV in comparison with Total Volume (TV) systems as mixing ventilation, displacement ventilation and under floor air distribution system has the following advantages:

- potentials to improve the inhaled air quality.
- Potentials to increase thermal comfort because each occupant is delegated the authority to optimise and control temperature, flow rate (local air velocity) and direction of the locally supplied personalized air according to his/her own preference.
- Potential to increase worker productivity.
- Potentials to reduce the risk of airborne transmission of infectious agents.

An extensive review of the literature about personalized ventilation has been performed by Melikov (2004). In the following paragraphs are reported only the results published after Melikov's review. At the end of the section 1.5.3 there is a schematic summary of this review.

1.5.1 Thermal comfort

Kaczmarczyk et al. (2004) compared the human response of 30 subjects to personalized ventilation with the response to Mixing Ventilation (MV). The air terminal device of the PV system was shaped as a half-cylinder with rectangular opening (240 · 75 mm) for air supply. Perceived air quality, thermal comfort, intensity of Sick Building Syndrome symptoms and performance of subjects were studied during 3h 45min exposures. In case of MV alone the room air temperature was 23°C and 26°C. The PVS supplied outdoor air at 23°C or 20°C or recirculated room air at 23°C when the room temperature was 23°C, and outdoor air at 20°C when the room temperature was 26°C. The personalized ventilation system providing outdoor air improved perceived air quality and decreased Sick Building Syndrome symptoms significantly compared to MV alone. Headache and decreased ability to think clearly were reported as least intense when the PV system supplied outdoor air at 20°C, while the most intense symptoms occurred with MV. PV increased self-estimated performance.

Gong et al. (2005) studied the human perception of locally applied airflow from personalized ventilation air terminal devices by tropically acclimatized people. 24 subjects were exposed to local airflow from the front and towards the face at six air velocities (from 0.15 to 0.9 m/s), at ambient temperatures of 26 and 23.5°C and local air temperature of 26, 23.5 and 21°C. The results showed that the subjects preferred air movement within a certain range, i.e., a higher percentage was dissatisfied at both low and high velocity values. They identified acceptable air velocity ranges from 0.3 up to 0.9 m/s.

Kaczmarczyk et al. (2006) studied the human response of 30 subjects to five different air terminal devices for personalized ventilation operating at two levels of room air temperature (23°C and 26°C). The results showed that the subjects actively used the possibility to change the airflow rate and to adjust the positioning of the air terminal device in regard to the airflow direction. The individual control provided allowed subjects to maintain thermal neutrality with the systems studied, except one, named "Headset" at the higher room temperature of 26°C. The local thermal environment created with personalized ventilation was assessed as acceptable. They also found that the airflow towards the face was preferred to the airflow towards the abdomen.

Niu et al. (2007) studied a chair-based personalized ventilation system that they believe can potentially be applied in theatres, cinemas, lecture halls, aircrafts, and offices. From the human subject tests they found that when the personalized air temperature was lower than the room temperature the air was able to bring ''a cool head'' and increased thermal comfort in comparison with mixing ventilation. They stated that feelings of irritation and local drafts could be eliminated by proper designs.

Sun et al. (2007) studied by human and thermal manikin experiments the performance of a circular perforated panel air terminal device for a personalized ventilation system operating under two levels of turbulent intensity. The PV system was adjusted to deliver treated outdoor air over a range of conditions. The results indicated that PV air supplied at lower turbulent intensity, when compared against that supplied at higher turbulent intensity, achieved a larger range of velocities at the face and a greater cooling effect on the head region. They also showed that the facial cooling effect of the manikin correlated well with the human facial thermal sensation, which implies the manikin measurement serves as an inexpensive alternative to using human subjects.

Yang et al. (2008) developed a ceiling mounted individually controlled personalized ventilation aims for providing each occupant with thermal comfort and clean air without affecting the indoor aesthetics. They used thermal manikin to test the ability of the system to cool people. They found that when the room and personalized air temperature was equal to 23.5°C and the airflow rate varied within the range 4-16 l/s per person, the cooling effect for the head region and the whole-body was equivalent to decrease of the room temperature (without PV) respectively from -1°C to -5°C and from -0.25°C to -1.5°C.

Kaczmarczyk et al. (2008) showed, by human subject laboratory tests, that increasing the personalized supply air to 26°C when the room temperature is 20°C improved thermal comfort and diminished draught discomfort without negative impact on perceived air quality.

According to Zhang (2003) the overall body comfort in warm and cool conditions is determined by the local thermal comfort of feet, hands, and head. Zhang et al. (2008) designed task-ambient conditioning (TAC) system that heats only the feet and hands, and cools only the hands and face, to provide comfort in a wide range of ambient environments. The main purpose of the system is not to provide a clean air through the personal ventilation system but to locally control the thermal micro-environment. The developed TAC uses a maximum of 41 W for cooling and 59 W for heating. They tested the system by human subject laboratory experiments at temperatures ranging between 18-30ºC. The results showed that the TAC system was able to maintain positive comfort levels across the entire temperature range tested, the perceived air quality was significantly improved, even if the air movement was re-circulated room air. By energy simulation they showed significant energy saving potential in hot and dry climate.

1.5.2 Inhaled air quality

Khalifa and Glauser (2006) invented a novel low-mixing PV nozzle that can lengthen the clean air core of a PV jet at low PV clean air flow rates. The nozzle comprises a primary nozzle surrounded by an annular, secondary nozzle. The clean air issues from the primary nozzle, whereas recirculated air issues from the secondary nozzle at roughly the same speed as the primary clean air. With the shear stress at the primary jet's boundary diminished, turbulent mixing is reduced and the jet's core is extended. Khalifa et al. (2008) tested by mean of tracer gas the co-flow air terminal device. The results showed that the new nozzle achieved a ventilation effectiveness close to 4 with an airflow of 2.4 l/s.

Melikov et al. (2007) proposed a solution that incorporates the PV air supply diffusers into the headrest of the user's chair. The results reveal that air supply flow rate, room temperature, the thermal insulation of the seat occupant's clothing and the position of the head rest with the attached air terminal devices are all important for the performance of the system with regard to inhaled air quality. When the room and the supply personalized temperature was equal to 26°C and the clothing insulation was 0.58 clo the percentage of inhaled clean air was already 80% at 7 l/s and 96% at 10 l/s. They suggested that the seat headrest PV can be used in crowded spaces, e.g. theatres, concert halls, aircraft cabins, ground vehicle compartments, etc., where occupants are seated most of the time.

Niu et al. (2007) studied the ventilation performance of a chair-based personalized ventilation system. By comparing eight different air terminal devices it was found that up to 80% of the inhaled air could be composed of fresh personalized air (ventilation effectiveness equal to 5) with a supply flow rate of less than 3.0 l/s. Taking into account both air quality and energy efficiency, they suggested to use a supply flow rate of from 0.8 to 1.6 l/s. In this range the amount of inhaled clean air varied within the range 60-65%.

Nielsen et al. (2007a) proposed a chair with integrated personalized ventilation discharging supply air at very low velocities and relying on the entrainment of this clean PV air from the natural convection flow around the human body. They found that more than 80% of the inhaled air is personalized air (ventilation effectiveness equal to 5) with an airflow rate in most cases equal to 10 l/s. In a following paper Nielsen et al. (2008) reported the results related to the influence by draught in the surroundings to the effectiveness of the system. The results showed that if the direction of the flow in the general air distribution system is controlled, it is possible to obtain a very high level of effectiveness (approx. 90%).

Halvoňová and Melikov (2008) proposed a novel PV concept named "ductless" PV. The main idea behind the "ductless" PV is in the utilization of the clean air supplied over the floor by displacement ventilation system. The "ductless" PV, installed at each desk and comprising an air supply terminal device, a small fan and a short duct, sucks clean air from near the floor at the location of the desk. From laboratory studies they showed that the "ductless" PV provides inhaled air quality similar or better than the quality obtained with DV alone. However the "ductless" personalized flow may increase the mixing of pollution generated in the room and thus decreases the inhaled air quality for the occupants who do not use their "ductless" personalized ventilation systems.

Melikov et al. (2008 a and b) showed, by human subject laboratory tests, that facially applied airflow with elevated velocity (0.3 and 0.6 m/s) significantly improves the acceptability of the air quality at room air temperature of 26°C and relative humidity of 70% and compensates for the negative impact of relative humidity on perceived air quality at low velocity level. The positive impact of elevated velocity on PAQ was larger at 26°C than at 20°C and it was larger at high pollution level than at low pollution level.

Bolashikov et al. (2003) reported on personalized ventilation device with air supply nozzle incorporated with the microphone of headset to supply small amount of clean air directly to the mouth/nose of the user. In reported on high performance of the device based on experiments with breathing thermal manikin. Zhu et al. (2008) studied, by computational fluid dynamic method, the performance of the personalized ventilation device with air supply nozzle incorporated in a headset. They studied the impact of the position of the air supply nozzle on the personalized air distribution at the vicinity of the mouth with circular and ellipse-like supply openings, by placing them either in front of, or below, or sideways to the mouth. The personalized air was supplied at a rate of 0.34 l/s towards the manikin's mouth, with the temperature of 23°C, which was same as the room air temperature. The results identified that the personalized airflow was able to penetrate the free convection airflow around the human body and to provide over 85% of clean air to inhalation.

Watanabe et al. (2008a) studied the performance of an individually controlled system comprising a convection-heated chair, an under-desk radiant heating panel, a floor radiant heating panel, an under-desk air terminal device supplying cool air and a desk mounted personalized ventilation using a thermal manikin at room temperatures of 20°C, 22°C and 26°C. They measured, at air temperature of 20°C, a maximum whole-body heating effect equal to 5.9°C, and at 26°C a maximum cooling effect equals to -0.8°C. Melikov and Knudsen (2005 and 2007) tasted the same system with forty-eight subjects. The subjects were provided with control of the flow rate and direction of the personalized air and under-desk airflow rate, the temperature of the convection flow from the chair, and the surface temperature of the heating panels. The results reveal that the thermal and air quality acceptability was significantly higher with the ICS at all room temperatures compared to the reference condition (a room air temperature of 22°C without the individually controlled system). The ICS could satisfy more than 85% of the subjects in regard to both thermal comfort and inhaled air quality. Watanabe et al. (2008b) analyzed the manikin and human subject data together. They concluded that both the heating and the cooling capacity of the individually controlled system need to be increased in order to satisfy most occupants in practice.

1.5.3 Protection from airborne transmitted diseases

Cermak and Melikov (2007) applied the model for prediction of the risk of airborne transmission of diseases suggested by Rudnick and Milton (2003) to compare the performance of mixing ventilation, underfloor ventilation and personalized ventilation in conjunction with background mixing ventilation. The results indicate that PV is more effective on protection occupants from spreading of influenza than the total volume systems analyzed.

Nielsen et al. (2007c) studied the ability of a personalized ventilation system to improve the protection of people in a room ventilated by an air distribution system based on a textile terminal. The air distribution was characterized as a displacement flow with a downward direction in areas of the room where no thermal load was present. The investigation involved full-scale experiments with two breathing thermal manikins. One manikin was the source and the other the target. The results show that personalized ventilation improves the protection of occupants.

According to Nielsen et al. (2007b) the personalized ventilation systems based on a supply jet have a limitation. The jet entrains air from the surroundings and, therefore, reduces the amount of fresh air which reaches the breathing zone. The entrainment is minimized when the source of clean air is located in the boundary layer close to the breathing zone (face). They tested a pillow and a blanket (mattress) made by a special textile used as a supply opening of fresh air. They obtained a very high protection. When the breathing manikin was lying on his side up to 95% of the inhaled air was clean, and the system in general had effectiveness larger than 50% to 80% while the flow rate was equal to 10 l/s in most cases.

PV has greater potential than total volume air distribution to protect occupants from airborne pathogens. Research in this area started only recently but there is already evidence that PV in conjunction with mixing ventilation can protect occupants from airborne pathogens and is superior to mixing air distribution alone (Bolashikov and Melikov 2007). Moreover, the PV system may be equipped with filter able to reduce the risk of airborne transmission due to their ability to kill the pathogens. Ultraviolet germicidal irradiation (UVGI) is the use of ultraviolet energy to kill or inactivate viral, bacterial and fungal species. Martin et al. (2008) made a literature review on the application of this technology for cleaning the air. A UVGI unit could be included in PV systems that use room air to guarantee that each occupant receives air free from pathogens. This solution may solve some problems related to the difficulties of supplying the personalized air (e.g. duct work) and to the PV energy consumption and it would further improve the PV effectiveness on protecting people. However laboratory and field studies are needed to evaluate the applicability of this technology and its possible drawbacks (e.g. ozone generation).

The researches reported in the previous three sections confirmed the ability of PV to:

- Increase thermal comfort (Gong et al. 2005; Niu et al. 2007; Yang et al. 2008; Kaczmarczyk et al. 2008; Zhang 2003 and 2008; Watanabe et al. 2008 a and b; Melikov and Knudsen 2005 and 2007).
- Increase self reported performance (Kaczmarczyk et al. 2004)
- Increase perceived air quality (Kaczmarczyk et al. 2004; Melikov et al. 2008 a and b).
- Increase ventilation effectiveness (Niu et al. 2007, Melikov et al. 2007, Zhu et al. 2008; Nielsen e al. 2007 a, b and 2008; Halvoňová and Melikov 2008).
- Decrease the risk of spreading of airborne transmitted diseases (Cermak and Melikov 2007; Nielsen et al. 2007c).
- Decrease SBS symptoms (Kaczmarczyk et al. 2004).

The reported researches concluded that:

- Acceptable air velocity for tropical acclimatized subjects ranges from 0.3 up to 0.9 m/s (Gong et al. 2005).
- Subjects actively use the possibility to change the airflow rate and to adjust the positioning of the air terminal device in regard to the airflow direction (Kaczmarczyk et al. 2006).
- Human subjects prefer airflow towards the face to the airflow towards the abdomen (Kaczmarczyk et al. 2006).
- The overall body comfort in warm and cool conditions is determined by the local thermal comfort of feet, hands, and head (Zhang 2003).
- It is possible to achieve a ventilation effectiveness close to 4 with an airflow of 2.4 l/s (Khalifa and Glauser 2006).
- PV may be integrated in chairs, pillows, blankets, and headsets (Niu et al 2007, Melikov et al. 2007, Nielsen et al. 2007a, b and 2008; Zhu et al. 2008).

The reported researches confirmed the advantages of PV summarized by Melikov (2004) in a review of the literature and showed improvements of the system; In particular regarding the ventilation effectiveness and thermal comfort. Systems that provide high ventilation effectiveness at low air flow rates had been developed. System that could satisfy more than 85% of the subjects in regard to both thermal comfort and inhaled air quality have been designed and tested. The described studies are mainly base on laboratory experiments or computer simulated work. In the future, the performance of PV systems installed in real buildings needs to be investigated.

1.5.4 Occupant density

The presence of occupants modifies the indoor environment because each human being emits heat and pollutants (such as water vapor, carbon dioxide, odours, bioeffluents etc.). Occupants also interact with buildings to enhance their personal comfort. For example they will heat, cool or ventilate their environment to improve their thermal comfort, they will adjust lighting systems or blinds to optimize their visual comfort (Page et al. 2008).

The effectiveness of a ventilation system depends on where occupants are (if they are in the ventilated area and in which part of it), for how long and how many they are in a given location at a given time. These information are even more useful for PV system, that generates a strongly non uniform environment, therefore its applicability and effectiveness would depend on the occupant behaviors. Knowing the occupation patterns would be useful to asses and quantify the influence of PV on thermal comfort (Nagareda et al. 2007) and on inhaled air quality (Zhao et al. 2003 and Yang et al. 2004).

PV systems decrease the pollutant concentration mostly in the microenvironment at the workstation, but they may also increase the contaminant in other zone of the room. Therefore, occupant's exposure to pollutant depends on the ratio of the time occupant stays at the workstation over the total time he/she stays in the room. This ratio is named Occupied Density (OD) (Zhao et al. 2003). The occupied density for the *i*th occupant is the ratio of time that occupant stays in a certain region over the time that occupant stays in the room, e.g. if the occupant stays at the desk for 3 hours and the total time he stays in the room is 4 hours, then the occupied density of the desk of that occupant is 0.75. Computational Fluid Dynamic (CFD) was used to apply this concept for studying the contaminant exposure of occupant when PV is used in combination with a total-volume ventilation system (Yang et al. 2004). The results showed that the effect of desk mounted personalized ventilation depends significantly on the type of occupant activity patterns, thus, on occupied density. Thus, the application of PV should be restricted to certain types of space and human activities. The capacity of PV to decrease the pollutant intake depends on, among other parameters, the time the occupant stays at the desk. The longer the occupant stays at the workstation, the higher he/she will benefit the advantages of PV. The main limitation of the index proposed by Zhao et al. (2003) and applied by Yang et al. (2004) is that it can be applied only to Computational Fluid Dynamic simulations and not to real measurement data. In order to apply the occupied density index to full-scale measurement of PV it would be needed to discretized it and to clearly define which are the zones that influences the human contaminant exposure. A new index combining a normalized concentration and a tailored definition of occupied density is proposed for assessment of benefits in regard to inhaled air quality from use of PV in practice is presented in the paragraph 3.3.

1.5.5 Energy-saving potential of personalized ventilation

Little is known about energy use of personalized ventilation. According to one of the main manufacture of personal ventilation system the main barrier to market of the PV system is the lack of knowledge on its energy consumption and on the control strategies that may reduce the energy requirements.

Very few papers tried to evaluate the energy consumption of personalized ventilation. Seem and Braun (1992) studied the energy use characteristic of a system incorporating personal environmental control compared with convectional designs through the use of computer simulations. They simulated the desktop personal environmental control system described by Arens et al. (1990). The system incorporated an electrical radiant panel, two local air distribution fans, a noise generator, a desk light and a workstation occupancy sensor. Their study showed that the effect of personal environmental control ranged between a 7% saving and 15% penalty in building lighting and HVAC electrical use. Bauman et al. (1998) measured the field performance of the same system described above. They reported that the energy consumption of a personal environmental control system follows the occupancy behaviour; the system switches off when occupants leave the workstation, thus allowing energy saving to be measured.

In the literature, information is available about the energy-saving potential of personalized ventilation. The main strategies suggested in the literature to have potential for energy-saving with personalized ventilation are:

- Reducing the outdoor airflow rate due to the higher ventilation effectiveness of PV (Melikov et al. 2002; Bolashikov et al. 2003; Faulkner et al. 2004; Sekhar et al. 2003; Sekhar et al. 2005 and Niu et al. 2007).
- Expanding the room temperature comfort limits by taking advantage of PV's ability to create a controlled microenvironment (Bauman et al. 1993; Sekhar et al. 2003; Kaczmarczyk et al. 2004 and Sekhar et al. 2005).
- Supplying the personalized air only when the occupant is present at the desk (Seem and Braun 1992; Bauman et al. 1993).

1.5.5.1 Reducing the airflow rate

There are several definitions of ventilation effectiveness (Mundt et al. 2004); in this thesis the ventilation effectiveness is defined as the ratio of the concentration of pollution in exhaust air divided by the concentration of pollution in air inhaled by occupants and it is defined by Eq. 1.

$$\varepsilon_V = \frac{c_E - c_S}{c_i - c_S} \qquad \text{Eq. 1}$$

where c_i is the contaminant concentration in the breathing zone [ppm];
c_S is the contaminant concentration in the supply air [ppm];
c_E is the contaminant concentration in the exhaust air [ppm].

This definition of ventilation effectiveness is also named pollutant removal efficiency (Faulkner et al. 2002). Ventilation effectiveness higher than one implies that the ventilation system is more effective than a traditional mixing ventilation system to supply and distribute the outdoor air. According to the European standard CEN EN 13779 (2008) and report CEN CR 1752 (1998), the minimum airflow rate can be reduced by using the ventilation effectiveness (divided by the ventilation effectiveness). The ASHRAE standard 129 (ASHRAE 2002) defines the Air Change Effectiveness (ACE) as the ratio of the age of the exhaust air and the age of the air in the breathing zone. According to ASHRAE standard 62.1 (ASHRAE 2004b) the minimum outdoor air supply rate could be decreased using ACE (multiplied by 1/ACE). Several studies reported a ventilation effectiveness higher than one. Hereafter are reported some significant examples.
Faulkner et al. (2004) studied in chamber experiments the ACE of a task ventilation system with an air supply nozzle located underneath the front edge of a desk. The personalized airflow rate per person (q_V) varied from 3.5 to 6.5 l/s. They reported that the system studied had an ACE equal to 1.5; therefore the minimum outdoor air supply rate could be decreased by one third. Sekhar et al. (2005) found that in a tropical climate, for an ambient temperature of 26°C, and a PV flow rate of 7 l/s per person at a supply air temperature of 23°C or 20°C, the ventilation effectiveness was 1.42. Melikov et al. (2002) studied in chamber experiments the influence of five different air terminal devices on the ventilation effectiveness with the airflow rate varying from 5 l/s up to 23 l/s. The ventilation effectiveness varied within the range 1.30- 2.38. A highly efficient air terminal device providing almost 100% clean and cool personalized air in each inhalation has been developed by Bolashikov et al. (2003). The air terminal device makes it possible to increase the ventilation effectiveness 20 times or more compared with mixing ventilation. Niu et al. (2007) studied the ventilation performance of a chair-based personalized ventilation system. By comparing eight

different air terminal devices it was found that up to 80% of the inhaled air could be composed of fresh personalized air (ventilation effectiveness equal to 5) with a supply flow rate of less than 3.0 l/s. Nielsen et al. (2007a) proposed a chair with integrated personalized ventilation discharging supply air at very low velocities and relying on the entrainment of this clean PV air from the natural convection flow around the human body. They found that more than 70-80% of the inhaled air is personalized air (ventilation effectiveness > 3.5-5) with an airflow rate in most cases equal to10 l/s.

1.5.5.2 Expanding the room operative temperature limits

A personal ventilation system allows for local control the microenvironment around the occupant. Therefore, it is possible, without scarifying the thermal comfort, to extend the indoor temperature limits. This may lead to an energy saving. Bauman et al. (1993) reported that at a high room air temperature (25°C - 27°C), the local cooling effect of the desktop system was able to maintain average temperatures in the occupied zone of one workstation from 0.5°C to 1.5°C below the corresponding temperatures in an adjacent workstation without a desktop system. Kaczmarczyk et al. (2004) in an experiment comprising 60 human subjects showed that at a room temperature of 26°C PV, supplying air at 20°C was able to keep occupants in better thermal comfort (close to neutrality instead of slightly warm) than a mixing ventilation system. Sekhar et al. (2005) showed that human subjects prefer, from a thermal comfort and perceived indoor air quality point of view, an environment with a room temperature of 26°C and PV at 23°C or 20°C rather than a room at 23°C without a PV system. They stated that for a tropical climate, where the common indoor temperature for a conditioned building is 23°C, a significant reduction of energy consumption can be achieved if the room temperature is maintained at 26°C.

1.5.5.3 Demand controlled personalized ventilation

Depending on their activities during working time occupants may spend only a part of the time in the office and even a shorter time at the desk (Bauman et al. 1994; Nobe et al. 2002; Bernard et al. 2003; Johansson et al. 2004; Wang et al. 2005; Nakagawa et al. 2007; Melikov and Hlavaty 2007; Halvarsson et al. 2006; Page et al. 2008); therefore energy-saving may be achieved if the system is able to automatically switch off when occupants are not at the desk. This technique is usually named demand controlled ventilation. According to Seem and Braun (1992) and to Bauman et al. (1993) energy saving may be achieved by supplying the air only when occupants are at the desk. Bauman et al. (1993), by means of laboratory experiments, showed that when this technique is implemented the energy consumption pattern is similar to the occupancy pattern.

2 OBJECTIVES

The main objectives of the present work are:

- To study by mean of computer simulation the energy saving when providing occupants with thermal comfort at elevated room temperature with increased air movement.
- To study by mean of computer simulation the energy consumption of a personalized ventilation system and to find out the most effective energy saving strategies that can be used to control a personal ventilation system.
- To develop and test by mean of laboratory measurements an index for the evaluation of the cooling fan efficiency. The index is the ratio between the cooling capacity of the fan and its energy consumption.

An additional objective of the study is to develop and test an index for evaluation of air quality improvements in room with non-homogeneous contaminant distribution (e.g. rooms with a personalized ventilation system) that takes into account the occupant location pattern.

3 METHODS

3.1 Energy simulation

According to the European Standard CEN EN 15615-2007 the "energy need" is the heat to be delivered to or extracted from a conditioned space to maintain the intended temperature conditions during a given period of time. The energy need is calculated and cannot easily be measured. This term does not take into account the efficiency of the HVAC system. It has been decide to use it instead of the energy use (commonly named energy consumption) or the primary energy because we would like to make the results independent of the HVAC system.

Two groups of simulation have been performed.
In the first one the potential saving of cooling energy by elevated air speed which can offset the impact of increased room air temperature on occupants' comfort was quantified by means of simulations with EnergyPlus software. Fifty-four cases covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem, Athens), three indoor environment categories - I, II and III (according to standard CEN EN 15251-2007) and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated. Air movement increase was generated either by personalized ventilation and cooling fan. In the second group of simulation the influence of the personalized supply air temperature control strategy on energy consumption and the energy-saving potentials of a personalized ventilation system was investigated by means of simulations with IDA-ICE software. GenOpt software was used to determine the optimal supply air temperature. The simulated office room was located in a cold climate.

The building locations and weather data, the description of the office room, the internal temperature, ventilation, infiltration rate and heat gains, the occupancy, the description of the HVAC system and of the simulation software are described in detail in the papers A and B. The European standard CEN EN 15265-2006 recommends a format for reporting the input data of an energy simulation. The input data in the papers A and B complies with the guidance in the standards. In the following paragraphs are described the simulated cases.

3.1.1 Energy analysis of increased air movement

The possibility of increasing the upper operative temperature limit at elevated velocity may reduce the energy consumption without affecting occupants' thermal comfort. The individual control of air movement can be achieved with personalized ventilation systems, task/ambient systems, desk, standing, tower or ceiling fans, and under some conditions with operable windows. The energy consumption for air movement generation by these methods is different. The purpose of these simulations is to quantify the potential savings of energy need for cooling achieved by elevated air speed without reducing occupants' thermal comfort conditions.

From Figure 1, assuming that the air temperature is equal to mean radiant temperature ($\theta_{IDA} = \theta_{mr}$), it is shown that the increase allowed in operative temperature is equal to 1.7°C for an airflow of 0.5 m/s and 2.5°C for an airflow of 0.8 m/s. These values were added to the maximum summer operative temperatures for the three indoor environment categories as specified in CEN EN 15251-2007. The values shown in Figure 1 were obtained for a comfort limit of 26°C, which is the comfortable temperature limit for category II in CEN EN 15251-2007. It is reasonable to assume that the same increments in operative temperature can be applied for the comfortable temperature limits for categories I and III, i.e. 25.5°C and 27°C. In total, fifty-four cases, covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem, Athens), three indoor environment categories (I, II and III) and three air velocities (<0.2, 0.5 and 0.8 m/s) as listed in Table 1, were simulated. The summer design day simulation was performed for fifty-four cases in order to calculate the maximum power needed for providing the comfort conditions. The maximum power is used to size the chiller. The summer design day conditions were taken from ASHRAE Fundamentals Handbook (2005). The cooling design days used in the simulation were characterized by an annual percentile of 1.0% for the dry-bulb temperatures and the mean coincident wet-bulb temperatures. These are suggested for use by ASHRAE Fundamentals Handbook (2005) when sizing cooling equipment such as chillers or air-conditioning units. The cities were chosen in order to describe in a homogeneous way different climate conditions (see Table 1 in the paper A). The focus was on summer conditions. The Cooling Degree Days (ASHRAE Fundamentals Handbook 2005) with a base temperature of 18°C were used as an indicator of the intensity of the summer period.

Table 1. Simulated cases: category of indoor environment, airflow rates, minimum and maximum operative temperatures. The maximum operative temperatures for cooling are increased according to the air velocity.

Category according CEN EN 15251-2007	Airflow per person [l/(s pers)]	Airflow per floor area [l/(s pers)]	Min θop for heating [°C]	Velocity [m/s]	Temperature increase [K]	Max θop for cooling [°C]
I	10	1	21	<0.2	0	25.5
				0.5	1.7	27.2
				0.8	2.5	28
II	7	0.7	20	<0.2	0	26
				0.5	1.7	27.7
				0.8	2.5	28.5
III	4	0.4	19	<0.2	0	27
				0.5	1.7	28.7
				0.8	2.5	29.5

[a] Recommended values from CEN EN 15251-2007 for low polluting buildings (see Annex C of CEN EN 15251-2007).

3.1.2 Energy analysis of a personalized ventilation system

The first purpose of energy simulations of a personal ventilation system in a cold climate (the building was located in Copenhagen) is to investigate the energy need of PV in comparison with a convectional mixing ventilation system for several control strategies of the supply air temperature (see Table 2 from Case 1 to Case 8). The second purpose of the simulation is to explore the strategies having potential for energy-saving listed in the section 1.5.5 (see Table 2 from Case 9 to Case 26). A mixing ventilation system supplying the air at a constant temperature (16°C) throughout the year is the reference case. All the simulated cases are summarised in Table 2 and described in the paragraphs 3.1.2.1 and 3.1.2.2.

3.1.2.1 Supply air temperature control

When the occupants are not provided with control over the temperature of the supplied personalized air, the building manager has to define the supply air temperature (θ_{SUP}) needed to provide the occupants with thermal comfort at a minimal level of energy consumption. In a single duct constant air volume system, θ_{SUP} set-point may be constant, or it may be reset based on the outdoor (θ_{ODA}) or indoor (θ_{IDA}) air temperature. PV supplies the air close to occupants. Therefore the lowest and highest permissible supply air temperatures are limited by thermal comfort issues. In this study it has been chosen that θ_{SUP} may vary in the range 20-26°C. All the θ_{SUP} profiles presented in the following are restricted within this range. In Case 1, 2, 3, θ_{SUP} was constant and equal to 20, 23, 26°C respectively. In Cases 4, 5, 6 (see Figure 2) the θ_{SUP} was reset according to θ_{ODA}. Two of them (Cases 4 and 5) were chosen by the authors and the other one, Case 6, was obtained using GenOpt (this software is discussed later in the paper).

Table 2. Simulated cases with the personalized ventilation.

Case	Control strategy of the supply air temperature	Supply air temperature profile	θ_{UP} [a] [°C]	Airflow rate per person q_V [l/(s person)]	Occupancy from 8:00-17:00
1	Constant	20°C	25.5	20	Full
2	Constant	23°C	25.5	20	Full
3	Constant	26°C	25.5	20	Full
4	Outdoor	Figure 2	25.5	20	Full
5	Outdoor	Figure 2	25.5	20	Full
6	Outdoor	Figure 2	25.5	20	Full
7	Indoor	Figure 3	25.5	20	Full
8	Indoor	Figure 3	25.5	20	Full
9	Constant	20°C	25.5	5	Full
10	Constant	20°C	25.5	10	Full
11	Constant	20°C	25.5	15	Full
12	Indoor	Figure 3	25.5	5	Full
13	Indoor	Figure 3	25.5	10	Full
14	Indoor	Figure 3	25.5	15	Full
15	Constant	20°C	27	20	Full
16	Constant	20°C	28	20	Full
17	Constant	20°C	29	20	Full
18	Constant	20°C	30	20	Full
19	Indoor	Figure 3	27	20	Full
20	Indoor	Figure 3	28	20	Full
21	Indoor	Figure 3	29	20	Full
22	Indoor	Figure 3	30	20	Full
23	Constant	20°C	25.5	20	Figure 4
24	Constant	20°C	25.5	Varying[b]	Figure 4
25	Constant	20°C	25.5	20	Figure 5
26	Constant	20°C	25.5	Varying[b]	Figure 5

[a] The cooling systems tried to keep the room operative temperature below the upper room operative temperature limit.
[b] The airflow varies accordingly to the occupation reported in Figure 4 and Figure 5 . At full occupation the airflow is equal to 20 l/s per person.

Cases 4 and 5 are characterized by supplying the personalized air at 20°C when the θ_{ODA}<20°C in order to minimize the heating energy that the Air Handling Unit (AHU) must provide to the supplied air. When θ_{ODA}>20°C the personalized air is supplied to the room without being conditioned. The profiles are limited in the upper part by a maximum supply air temperature equal to 22 and 26°C respectively. GenOpt software was used to find the optimal supply air temperature profile (Case 6) within the boundaries of the room air temperature given by CEN EN 15251-2007 for category I of the indoor environment. GenOpt was set to minimize the sum of energy needed for heating and cooling of the outdoor supply airflow rate and the room (mathematically named cost function). In order to minimize the cost function, GenOpt changes the θ_{SUP} corresponding to the following fixed outdoor temperatures (-20, 10, 15, 18, 20, 21, 23, 25, 26, 27, 30, 40°C) by choosing an integer value within the range 20-26°C. In Cases 7 and 8 (see Figure 3) the θ_{SUP} was controlled by the θ_{IDA}, which is equal to the return air temperature in a mixing ventilation system. The Case 7 profile aims to maximize occupants' thermal comfort because it supplies hot air when it is chilly in the room and cool air when it is warm; the profile was named "comfort" profile. The authors expect that the "comfort" profile would probably be used by the occupants if they would have the opportunity to control the supply air temperature. In Case 8 the air is supplied isothermally within the range 20-26°C, based on recent findings indicating that elevated velocity at the breathing zone improves inhaled air quality and compensates for the negative impact of increased temperature on perceived air quality (Melikov and Kaczmarczyk 2008 a and b). The profile was named "isothermal" profile.

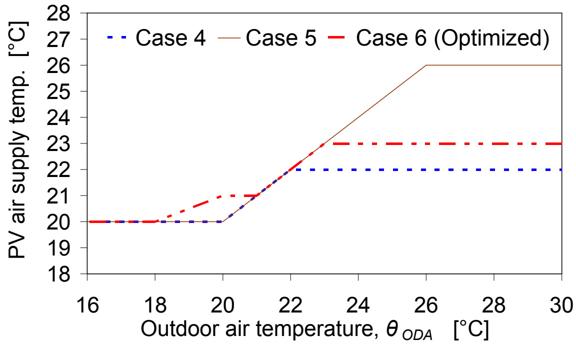

Figure 2. PV supply air temperature profiles as a function of the outdoor air temperature for Cases 4, 5 and 6 (See Table 2).

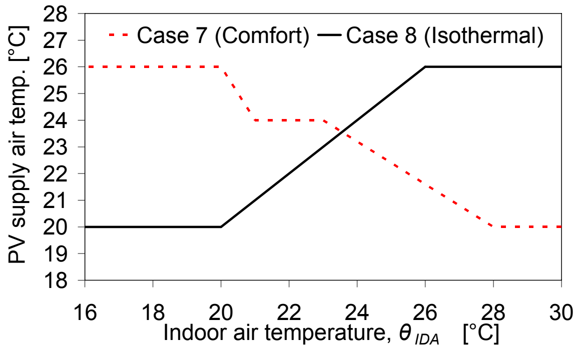

Figure 3. PV supply air temperature profiles as a function of the indoor air temperature for Case 7 and Case 8 (See Table 2).

3.1.2.2 Energy-saving strategies

The three energy-saving strategies presented in the paragraph 1.5.5 were investigated (from Case 9 to Case 26, see Table 2). Two supply air temperature strategies were used: supplying the air at 20°C constantly for the whole year (Case 1) and the "comfort" profile (see Figure 3, Case 7). The former has been chosen because from the simulation it was found that it is the strategy which minimizes the energy need.

From the Case 9 to Case 14 (see Table 2) the effectiveness of reducing the q_V was studied. q_V was reduced to 15, 10, and 5 l/s per person. These values correspond to a ventilation effectiveness of 1.34, 2 and 4 respectively. From the Case 15 to Case 22 (see Table 2) the effectiveness of expanding the θ_{UP} was studied. θ_{UP} was expanded from 25.5°C (corresponding to Category I of the indoor environment according to CEN EN 15251-2007) to 27, 28, 29, and 30°C. The lower room operative temperature was kept equal to 21°C because it was found that reducing it (e.g. to 18°C) does not affect the energy need.

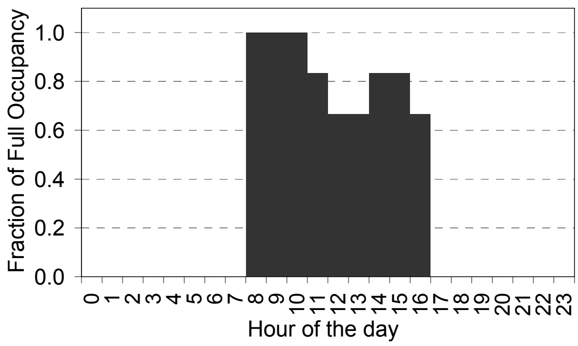

Figure 4. Occupancy profile according to the standard CEN EN 15232-2006.

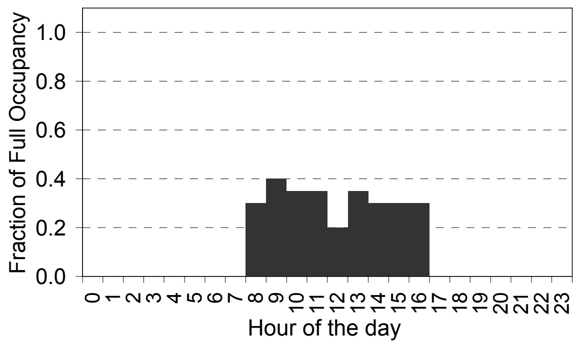

Figure 5. Occupancy profile according to the measured data by Nobe et al. (2002).

From the Case 23 to Case 26 (see Table 2) the effectiveness of supplying the personalized air only when the occupant is present at the desk was studied. Two occupancy behaviour profiles were used. The fraction of full occupancy is defined as the ratio between the actual number of occupants seated at the desk over the maximum number of occupants for whom the room was designed. The first occupancy behaviour profile (shown in Figure 4) has been obtained from the European standard

CEN EN 15232-2006. The second profile (shown in Figure 5) has been extrapolated by the data measured by Nobe et al. (2002) in a Japanese 52-story office building where 240 workstations were monitored for a week. The two profiles were bounded within the office hours (from 8:00 to 17:00). It is assumed that when the occupant is not at his/her desk he/she is out of the office. When the occupant is not at the desk the heat loads generated by him/her and his/her equipment is not taken into account, and in the Cases 24 and 26 the personalized air is switched off.

3.2 Physical measurements

3.2.1 Cooling fan efficiency index

In this paragraph an index that can be used for the evaluation of the efficiency of the cooling fans is defined and described.

The efficiency is usually the ratio of the output to the input. It can be improved by reducing input and/or improving output. In the case of fans, used for cooling people in warm environments by increasing the air velocity around the human body, the input is the electrical energy needed for running the fan (the power requirement of a fan is almost constant and it can be used instead of energy in order to make the input variable time-independent) and the output is the body cooling effect.

The body cooling effect is the result of a complex interaction of many parameters. The body cooling effect produced by a fan depends on generated air velocity and turbulence field, body area exposed to moving air, body posture, air and mean radiant temperature, air humidity, clothing insulation, metabolic rate, humidity, and skin wettedness. Sophisticated thermal manikins with full body size and a complex shape have been developed and used for determination of the heat loss from the human body under different environmental conditions (Tanabe et al. 1994; Tsuzuki et al. 1999; Melikov et al. 2002). A manikin's body is typically divided into several segments. They can be operated to maintain constant heat flux from the body, constant body surface temperature, or to have surface temperature equal to the skin temperature of an average person in a state of thermal comfort under the particular environmental condition of the exposure. Thermal manikins can be used to measure the fan cooling effect and thus to determine the cooling fan efficiency index. Thermal manikins that can measure dry heat loss from the human body are most commonly used today though sweating thermal manikins are under development as well (Psikuta et al. 2008). Therefore at this stage, dry heat loss from the human body can be used for determining the cooling fan efficiency. Clothing thermal insulation and metabolic rate (personal factors that may vary substantially in real life) can be assumed to be constant, while air humidity and skin wettedness are not taken into account.

The equivalent temperature (t_{eq}) is a well-known parameter that can be used for determining the cooling fan efficiency index. In the SAE (1993), equivalent temperature (former Equivalent Homogenous Temperature) is defined as: "The uniform temperature of the imaginary enclosure with air velocity equal to zero in which a person will exchange the same dry heat by radiation and convection as in the actual non-uniform environment". The same definition was used by Nilsson et al. (1999). In the definition it is assumed that the body posture, the activity level and the clothing design and thermal insulation is the same in both environments. The equivalent temperature is a pure physical quantity that in a physically sound way integrates the independent effects of convection and radiation on human body heat loss. t_{eq} does not take into account human perception and sensation or other subjective aspects, but may correlate with them. It is important to notice that t_{eq} is not a temperature that can be measured by a thermometer and that t_{eq} cannot be translated to an air temperature in a complex climate (Bohm et al. 1999).

The body cooling effect achieved by air movement can be quantified by the change in whole-body manikin-based equivalent temperature, t_{eq}, from the reference condition, t_{eq}* (similar indoor environmental conditions but without air movement), i.e. $\Delta t_{eq} = t_{eq} - t_{eq}$*. The concept of Δt_{eq} has been already used by several authors to quantify the whole-body cooling effect of air movement (Tanabe et al. 1994; Tsuzuki et al. 1999; Melikov et al. 2002; Watanabe et al. 2005; Sun et al. 2007). Thus, the Cooling Fan Efficiency (*CFE*) is defined by the following equation:

$$CFE = (-1)\frac{\Delta t_{eq}}{P_f} \qquad \text{Eq. 2}$$

where P_f fan power. It is the input power of the fan (CEN EN 12792-2003) [W].
Δt_{eq} whole-body cooling effect [°C].

The measuring unit of *CFE* is °C/W. Δt_{eq} would be usually negative (the equivalent temperature of the body cooled by a fan would be lower that the temperature without the fan). In order to have an index that is easy to interpret, the ratio between the cooling effect and the fan power has been multiplied by -1. The higher the *CFE* index, the better the fan performance.

Figure 1 shows the cooling fan efficiency as a function of the fan power calculated at cooling effect Δt_{eq} of -0.5, -1, -2, -3 and -4°C. It has been reported that a cooling effect of -4°C obtained by local body cooling can be acceptable for people (Watanabe et al. 2005 and 2008). An internet survey showed that the typical power consumption of cooling fans is lower than 90 W. The figure shows that at constant cooling effect the *CFE* increases with the decrease of the fan power, i.e. fans with different power may have the same cooling effect. The figure also shows that fans with the same air power may have a different cooling effect due to differences in the generated flow, e.g. different target area, velocity and turbulence field, etc.

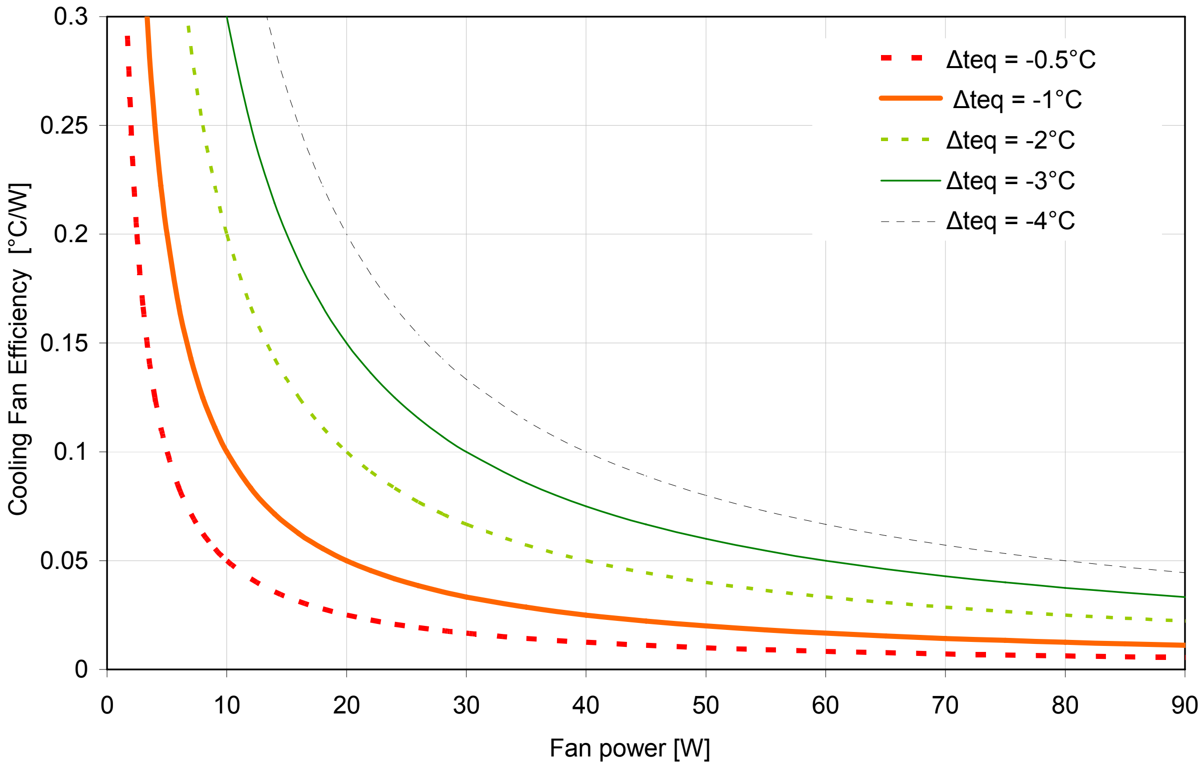

Figure 6. Cooling fan efficiency versus fan power for five cooling effect levels.

Knowing the cooling fan efficiency index (*CFE*) and its cooling effect (Δt_{eq}) will help customers to purchase a better fan, fan designers/manufacturers to assess and develop better products, and policymakers to fix minimum values or classes of fan efficiency as is usually done with other electrical appliances (e.g. air-conditioner, refrigerators, boilers, etc.). HVAC designers may choose the summer maximum allowed room temperature, depending on the cooling capacity of the fan. They may also evaluate the possibility for energy saving based on the strategy of increased air movement at elevated room air temperature.

The cooling fan efficiency index, *CFE*, has been measured in a real office for three temperature levels (25, 27, 30°C) and for the velocity levels of the fans. Experiments were performed with four fans available on the market including a ceiling fan (CF), a desk fan (DF), a standing fan (SF) and a tower fan (TF). A detailed description of the office, of the experimental facilities, of the measuring instruments, of the experimental conditions, of the experimental procedure and of the statistical method used is reported in paper C. The description of the uncertainties of the measured and derived quantities is reported in Appendix B of the paper C.

3.3 Occupant normalized concentration index

In order to describe the different location an occupant can stays in a room ventilated with PV and, at the same time, do not increase too much the number of measurements needed to quantify the assumed locations a modified definition of the occupied density index suggested by Zhao et al. (2003) is developed. The occupied zone of the room is divided in two regions:

1. Workstation region, e.g. occupant working at the desk, characterized by the average values of physical parameters measured at the workstation at the height of 1.1 m above the floor.
2. Background region, characterized by the average values of physical parameters measured at the height of 1.7 m above the floor. It is supposed that the occupant is standing in the office when he/she is not at the workstation.

Thus the ratio of time the occupant is at the workstation over the total time he/she stays in the ventilated room, defines the workstation occupied density index ODW:

$$ODW = \frac{\tau_W}{\tau_{TOT}} \qquad \text{Eq. 3}$$

where τ_{TOT} is the total time the occupant stays in the ventilated room [hour];
τ_W is the time the occupant spends at the workstation [hour];
τ_S is the time the occupant spends standing in the remaining (background) area of the room ($\tau_{TOT} = \tau_W + \tau_S$) [hour].

Similarly, the ratio of time that the occupant spends in the background area of the room over the total time he/she stays in the ventilated room is defined as, the background occupied density index, ODB. It is clear that the sum of ODB and ODW will be equal to 1. The normalized concentration of contaminant $\dot{c}$ is defined by Eq. 4.

$$\dot{c} = \frac{c - c_S}{c_E - c_S} \qquad \text{Eq. 4}$$

where c is the contaminant concentration in a point [ppm];
c_S is the contaminant concentration in the supply air [ppm];
c_E is the contaminant concentration in the exhaust air [ppm].

The normalized concentration is equal to 1 if there is complete mixing of air and contaminants. If the air quality is better than in the exhaust, the normalized concentration is lower than 1 and vice versa. The supply air has a normalized concentration of 0. The reciprocal value of the normalized concentration is known as ventilation effectiveness.

The occupant normalized concentration (C) is the normalized concentration weighed by the workstation occupied density, ODW. i.e. it is the weighed normalized concentration to which the occupant is exposed in average if he/she stays for τ_W at the workstation and for τ_S in the background area. This index is mathematically described by Eq. 5.

$$C = \dot{c}_W \cdot ODW + \dot{c}_S \cdot (1 - ODW) \qquad \text{Eq. 5}$$

where $\dot{c}_W$ is the normalized contaminant concentration in the air inhaled by the occupant at the workstation [-];

$\dot{c}_S$ is the normalized contaminant concentration in the air inhaled by the occupant standing in the background area of the room [-].

The occupant normalized concentration (C) is a linear function of ODW. The occupant normalized concentration is an index which determines the quantity of pollutant in air inhaled by the occupant. The occupant normalized concentration can be used to calculate the average pollutant exposure as a function of the pollutant distribution in a space and of the occupant activity. It can be applied to total-ventilation system and to personal ventilation system. The lower the normalized concentration is, the better the inhaled air quality is.

The index can be used for comparison of different air distribution systems in regard to the quality of air inhaled by occupants performing office work with different type of occupancy. The index can be applied to any type of ventilation system, though it has been developed for personalized ventilation. In the following the index is applied in the case of PV in conjunction with total volume ventilation in order to quantify the advantages of introducing a PV system in an total-volume ventilation system. Three scenarios are considered: first, the performance of only the total-volume ventilation system in operation is characterized by the normalized concentration defined at the workstation ($\dot{c}_{TVW}$) and in the background of the room ($\dot{c}_{TVS}$); second, the performance of the total-volume ventilation operating in conjunction with PV which efficiently protects the occupant and provides clean air in inhalation is characterized by the normalized concentration at the workstation ($\dot{c}_{PVpW}$), and by the normalized concentration in the background ($\dot{c}_{PVS}$); third, the performance of the total-volume ventilation operating in conjunction with PV which does not provide clean air to inhalation (or may be turned off) and does not protect the occupant from air pollution present in the room air is characterized by the normalized concentration at the workstation ($\dot{c}_{PVnpW}$),, and by the normalized concentration in the background ($\dot{c}_{PVS}$). The defined normalized concentrations are used to calculate the occupant normalized concentration, in the case of total volume ventilation alone (C_{TV}), total volume ventilation in conjunction with personalized ventilation protecting the occupant (C_{PVp}), and total volume ventilation in conjunction with PV which does not protect the occupant efficiently or is turned off (C_{PVnp}). The normalized concentrations, $\dot{c}_{TVW}$, $\dot{c}_{TVS}$, c_{PVpW}, $\dot{c}_{PVnpW}$ and $\dot{c}_{PVS}$ are function of the type of the total-volume and the personalized ventilation systems adopted and of the pollution source considered; the occupant normalized concentrations C_{TV}, C_{PVp} and C_{PVnp} are also function of the ODW. The lower the occupant normalized concentration is the better the inhaled air quality will be because the amount of inhaled pollution will be lower.

The usefulness of the occupant normalized concentration index is demonstrated with data collected during full-scale measurements of personalized ventilation in conjunction with total volume ventilation system (mixing and displacement) and total volume ventilation performing alone as reported in (Cermak 2004 and Cermak et al. 2006). The results are reported in the paragraph 4.4.

4 RESULTS

4.1 Energy saving by increased air movement

The energy need for cooling ($E_{N,C}$) of the room when located in each of the selected six cities for the three categories (Table 1) at the three levels of velocity (0.2, 0.5, and 0.8 m/s) and the corresponding operative temperatures (Table 1) is listed in Table 3. The energy need for cooling is the annual amount of cooling energy that must be supplied to the room to keep the operative temperature below the maximum summer operative temperature limit. The cooling energy for the control of humidity and the energy losses in the system are not included.

The heating energy need is not affected by the air velocity increase. It depends on the outdoor conditions (climate zone), on the building characteristics, on the heat loads and on the required category of the indoor environment. The maximum heating energy need is in Helsinki for category I (83 kWh/m^{2}y). In Rome, Jerusalem and Athens the heating demand is covered by the internal heat load, and there is therefore no need for a heating system.

Table 3. Energy need for cooling ($E_{N,C}$) per unit of floor and fan operating hours at the three velocity levels for the three categories of indoor environment when the room is located in the six cities with different outdoor climate conditions. The energy saved due to the increase of air velocity (or relative increase of upper operative temperature limits) is listed.

		Velocity< 0.2 m/s Reference case	Velocity = 0.5 m/s				Velocity = 0.8 m/s			
			Energy		Fan		Energy		Fan	
City	C. [a]	$E_{N,C}$ [b]	$E_{N,C}$ [b]	Saved [c]	$h_{0.5}$ [d]	$h_{0.5}/h_{tot}$ [e]	$E_{N,C}$ [b]	Saved [c]	$h_{0.8}$ [d]	$h_{0.8}/h_{tot}$ [e]
Hel-sinki	I	18	12	34%	636	31%	9	48%	645	31%
	II	21	15	29%	765	37%	12	41%	788	38%
	III	24	18	24%	859	41%	16	35%	867	42%
Berlin	I	24	16	32%	814	31%	13	45%	826	31%
	II	26	19	28%	848	37%	16	40%	864	38%
	III	27	21	23%	907	41%	18	34%	916	42%
Bor-deaux	I	39	28	27%	1080	52%	24	38%	1091	52%
	II	41	31	24%	1184	57%	27	34%	1204	58%
	III	42	33	21%	1345	65%	29	31%	1368	66%
Rome	I	52	40	23%	1300	63%	35	33%	1308	63%
	II	53	42	21%	1406	68%	37	30%	1420	68%
	III	53	43	19%	1499	72%	38	27%	1509	73%
Jeru-salem	I	65	51	21%	1483	71%	45	30%	1491	72%
	II	66	52	20%	1722	83%	47	29%	1746	84%
	III	66	54	19%	1909	92%	48	27%	1928	93%
Athens	I	75	61	18%	1419	68%	56	25%	1439	69%
	II	74	61	17%	1555	75%	56	25%	1579	76%
	III	73	61	17%	1888	91%	55	24%	1921	92%

[a] C. = Category according CEN EN 15251-2007.
[b] $E_{N,C}$ = Energy need for cooling [kWh/(m^{2}y)].
[c] Saved. = Percentage of the saved energy need for cooling compared to the reference case.
[d] h_i = Annual number of hours that the fan is operating for increasing the air velocity. (i=0.5 m/s or 0.8 m/s)
[e] h_i/h_{tot} = Annual number of hours that the fan is operating (h_i) over yearly occupant working hours (h_{tot}).

The fan operation total hours (h_i) are shown in Table 3 as well. It is assumed that when the indoor operative temperature is higher than the maximum operative temperature limit (without any increase of the air velocity) the occupant switches on the fan. Thus the fan operation hours were calculated as the sum of hours during which the operative temperature was higher than the maximum operative temperature limit and the occupant was in the room, e.g. an hour is counted if the occupant was in the room and the room operative temperature was above 25.5°C for category I, or above 26°C for category II, or above 27°C for category III. The total number of hours that the fan is in operation is proportional to the energy consumption of the fan. In Table 3 the ratio between the fan operation hours and the total yearly occupant working hours is reported. The total occupant working hours (h_{tot}) per year (260 working days) is 2080. The maximum cooling power per unit of floor area and the percentage of time that the relative humidity requirements are fulfilled are shown in Table 4 of the paper A.

4.2 Energy consumption of a personalized ventilation system

To better understand the results the "energy need" has been divided in four parts. It is the sum of energy for heating (AHU Heating) and cooling (AHU Cooling) the supplied air in order to obtain the desired θ_{SUP} and for heating (Room Heating) and cooling (Room Cooling) the conditioned space in order to maintain the indoor operative temperature within the designed range during a given period of time (from 6:00 to 17:00).

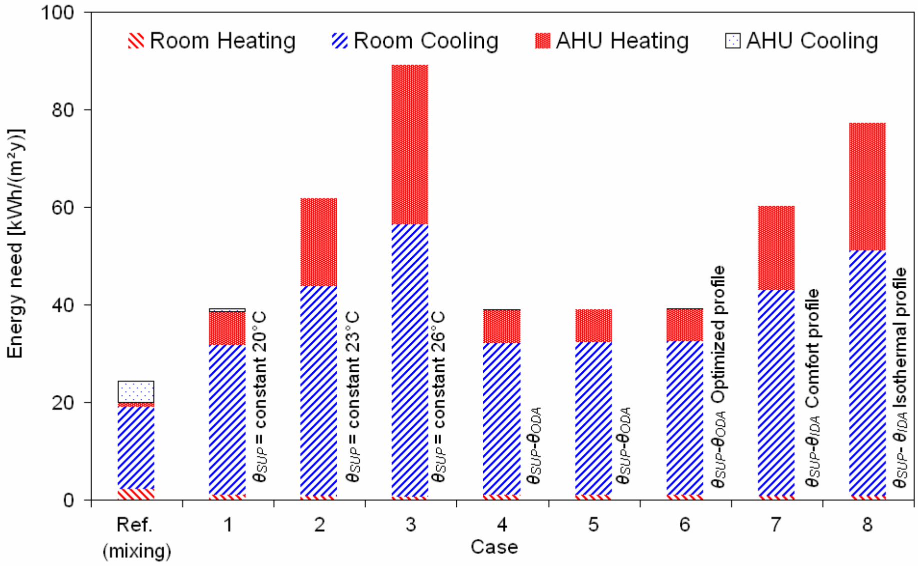

Figure 7. The energy need for several control strategies of the personalized supply air temperature, θ_{SUP} (see Table 2).

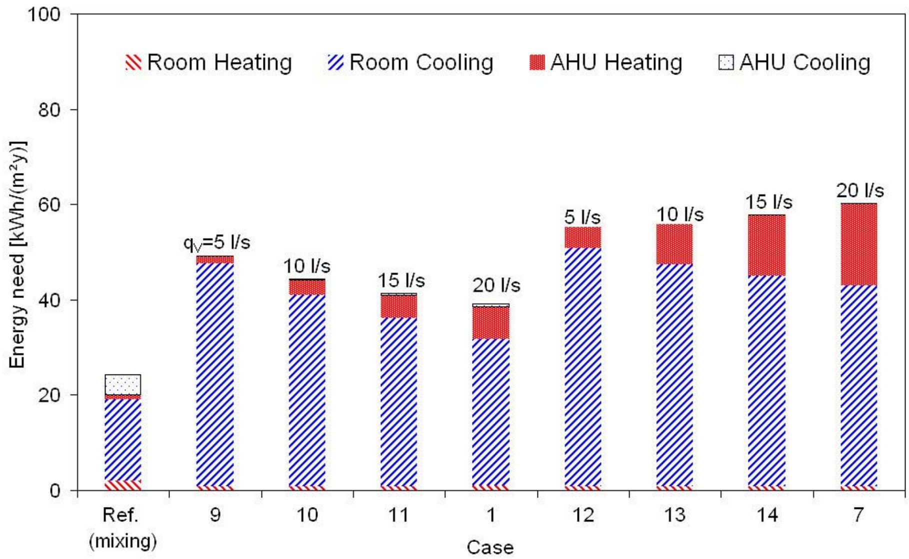

Figure 8. The energy need for the reduced outdoor airflow rates, q_V, for θ_{SUP} constant and equal to 20°C (Cases 9, 10, 11, 1) and for θ_{SUP} following the comfort profile shown in Figure 3 (Cases 12, 13, 14, 7).

The energy need for several θ_{SUP} control strategies (Table 2, Cases 1 - 8) is shown in Figure 7. The energy need for the reduced outdoor airflow rates (Table 2, Cases 9 - 14) is shown in Figure 8. The energy need for the expanded upper room operative temperature limits (Table 2, Cases 15 - 22) is shown in Figure 9. The energy need for personalized air supplied only when the occupant is present at the desk (Table 2, Cases 23 - 26) is shown in Figure 10.

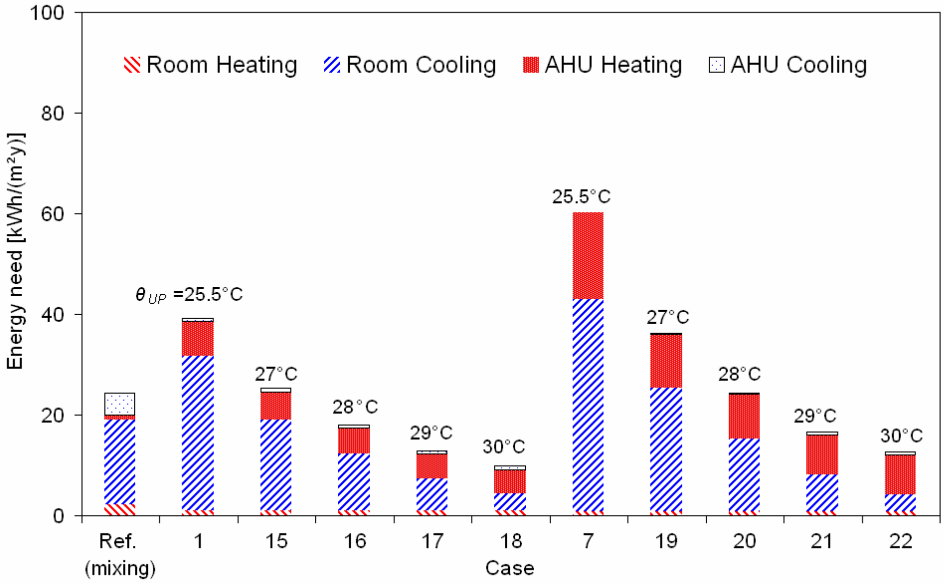

Figure 9. The energy need for the expanded room temperature comfort limits, θ_{UP}, rates for θ_{SUP} constant and equal to 20°C (Cases 1, 15, 16, 17, 18) and for θ_{SUP} following the comfort profile shown in Figure 3 (Cases 7, 19, 20, 21, 22).

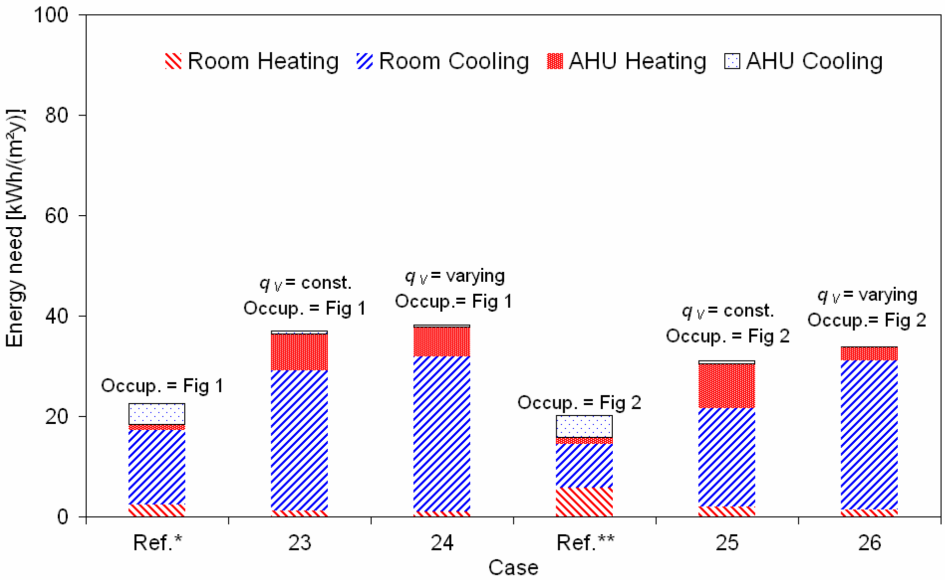

Figure 10. The energy need for personalized air supplied only when the occupants are present at the desk for the occupancy profile shown in Figure 4 and for the one shown in Figure 5. The airflow rate is constant in Case 23 and 25, and it varies according to the occupancy profile in Case 24 and 26. Ref* and Ref ** are respectively the energy need for the reference case (mixing ventilation) when the occupancy and the relative heat loads are varied according to the profiles shown in Figure 4 and Figure 5.

4.3 Cooling fan efficiency index

The cooling effect, Δt_{eq}, the fan power, P_f, and the cooling fan efficiency index, *CFE,* were obtained for each of the four fans under the experimental conditions studied. These are listed in Table C-1, Appendix C of the paper C. The results identify a large variation in the whole-body cooling effect (between -3.2 and -0.4°C), in the fan power (between 15.6 and 49.3 W) and in the *CFE* index (between 0.009 and 0.177°C/W).

The results obtained with the four fans at the room air temperatures and velocity levels studied are compared in Figure 11. The desk fan has the highest *CFE* index (*CFE* varies between 0.095 and 0.177°C/W) and the smallest power consumption (P_f varies between 16 and 20 W). The ceiling, the standing and the tower fans have similar results: *CFE* and P_f for the ceiling fan, the standing fan and the tower fan varied respectively in the ranges 0.018 - 0.079°C/W and 37 - 48 W, 0.038 - 0.058°C/W and 33 - 40 W and 0.009 - 0.066°C/W and 37 - 49 W. The results also indicate that the *CFE* of the desk fan is substantially more sensitive to the changes in the room air temperature and velocity level than the *CFE* of the other three fans. The *CFE* of the standing fan is least affected by the change of the room air temperature and fan velocity.

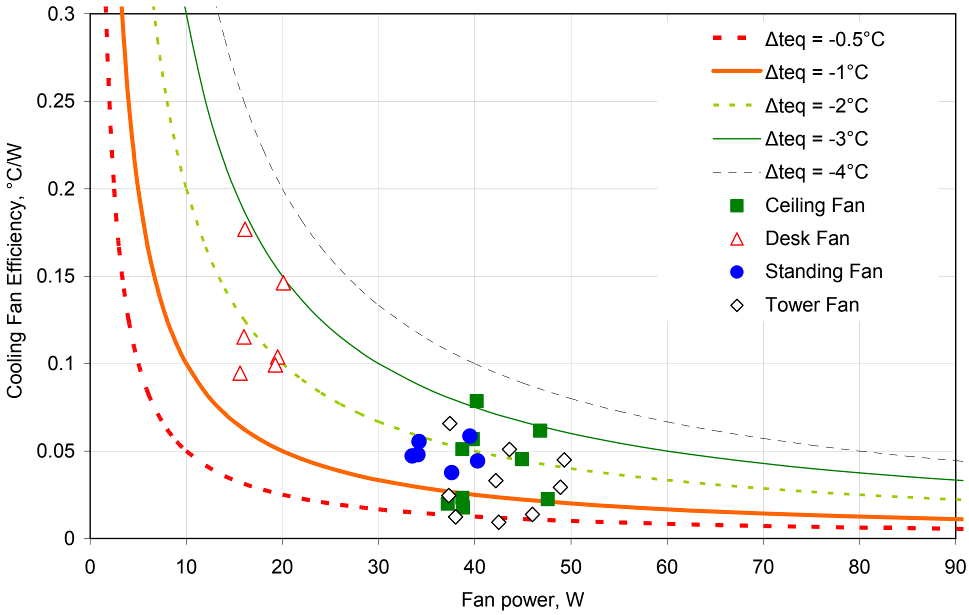

Figure 11. Fan power versus cooling fan efficiency index for the ceiling fan (CF) the desk fan (DF), the standing fan (SF), and the tower fan (TF). Lines with constant cooling effect (Δt_{eq}) are plotted.

The average of the cooling fan efficiency obtained for different room air temperatures and velocity levels with each of the fans was calculated. It is compared in Figure 12. The sample standard uncertainty of the index is equal to ±0.009°C/W (see Appendix B of paper C). The desk fan is the most effective cooling device; its cooling fan efficiency (*CFE*=0.123°C/W) being more then double the index of the other fans (between *CFE*=0.032-0.048°C/W). The tower fan is the least efficient cooling device. The efficiency of the DF is significantly ($p<0.01$) higher than the *CFE* of the remaining three fans. No significant difference in efficiency of these three fans was found (except that the *CFE* of the SF is higher than the *CFE* of the TF).

It is useful for the interpretation of the results to plot the whole-body cooling effect versus the fan power. An example is shown in Figure 13 and the conclusions that can be drawn from the figure are described hereafter. In the Figure 13 the whole-body cooling effect determined is plotted versus the fan power measured when the room temperature was equal to 25°C. The relative uncertainties are shown. The whole-body cooling effect of the desk fan and the ceiling fan is almost the same (around -2.5°C). However, the desk fan needs less than half of the electrical power used by the ceiling fan (around 20 W compared to 40 W). The DF and CF have a higher cooling effect than the TF and the SF. The SF has the lowest cooling effect, lower than -2°C, with a fan power that varies in the range 35-40 W. For the TF an increase of the velocity level implies a slight reduction of the cooling effect with an increase of the needed power. Increasing the velocity level always implies an increase of the power requirement but this does not always causes a higher cooling effect. From the results shown in Figure 13 it can be concluded that changing the velocity level is not an effective way of controlling the cooling effect.

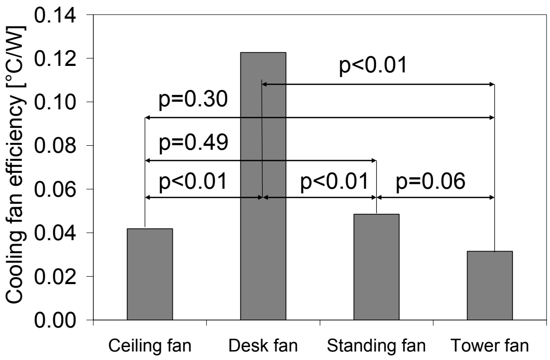

Figure 12. Averaged (over the velocity levels and room air temperature) cooling fan efficiency index for the ceiling, desk, standing and tower fan.

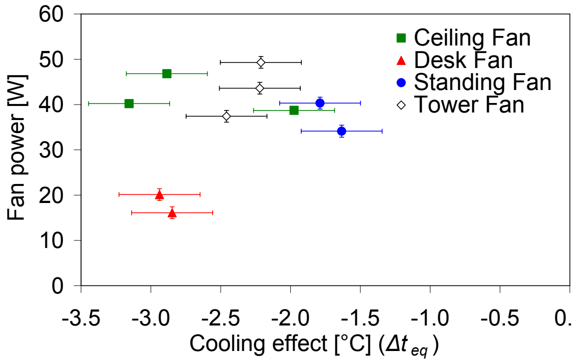

Figure 13. Cooling effect versus fan power for the ceiling, desk, standing and tower fan for the tested velocity levels when the room temperature was set to 25°C.

The influence of the room air temperature on the *CFE* was analyzed. The average of the *CFE* index obtained with the four fans at the tested velocities was calculated for each of the room air temperatures. The results are compared in Figure 14. From a heat transfer point of view the room air temperature has an influence on the cooling effect, and thus should have an influence on the cooling fan efficiency index. A significant ($p<0.01$) difference was found between the *CFE* determined at 25°C and the *CFE* at 30°C. The results (Table C-1 of Appendix C in paper C) also reveal that the room air temperature has no effect on the power consumption of the fan.

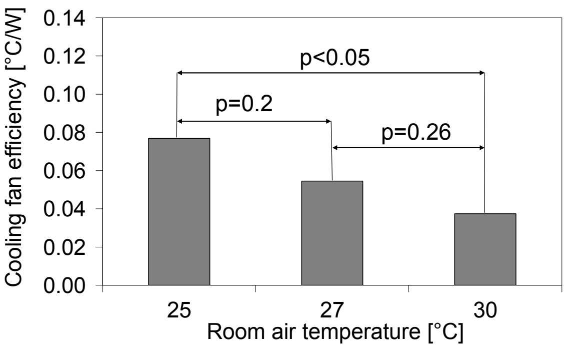

Figure 14. Averaged (over the type of cooling fan and velocity level) cooling fan efficiency for three room air temperatures.

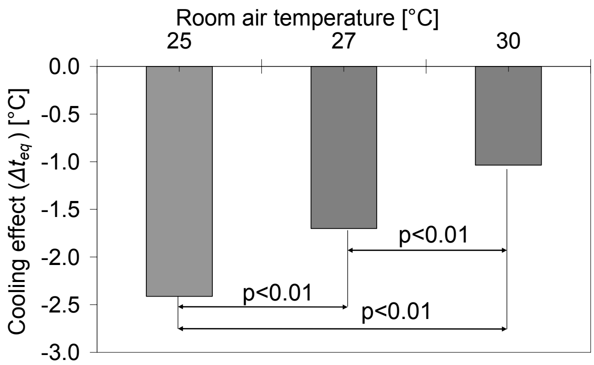

Figure 15. Averaged (over the type of cooling fan and velocity level) cooling effect for three room air temperatures.

4.3.1 Cooling effect

The cooling effect depends on the room air temperature and on the velocity levels. For the tested conditions, the cooling effect of the Ceiling Fan (CF) varied between -3 and -0.5°C, of the DF between -3 and -1.5°C, of the SF between -2.5 and -1.5°C and of the TF between -2.5 and 0.5°C. The cooling effect of the SF was least affected by the change in the experimental conditions.

The influence of the room air temperature on the cooling effect was analyzed. The average of the cooling effect obtained with the four fans at the tested velocities was calculated for each of the room air temperatures. The results are compared in Figure 15. As expected, the cooling effect increase significantly ($p<0.01$) with the decrease of the room air temperature.

The whole-body cooling effect (Δt_{eq}) discussed so far it is the weighted average of the cooling effect of each body segment. The cooling of the body segments depends on the local flow field generated by the fans. Analyses of the local cooling effect obtained by the tested fans for each body segment were performed. In the following, the results obtained at a room air temperature of 25°C are shown and discussed because the conclusions were rather similar for the results obtained at 27°C and 30°C.

The local cooling effect of the four fans on each of the 22 body segments of the manikin is shown in Figure 16. The desk and standing fan had two velocity levels, while the ceiling and the tower fan had three levels. The cooling effect increases with the increase of the velocity level. However, the exposure to the airflow has much a stronger effect. The body segments exposed directly to the flow are cooled much more than for those in "shadow". The impact of the velocity level on the cooling effect is greater for the exposed body segments than the segments in "shadow".

A detailed analysis of the local cooling effect of the body segment for each fan is reported in the paper C, here it is just underlined that the fans generated a strongly non-uniform cooling of the manikin's body.

The flow field generated by the fans was non-uniform and therefore caused non-uniform local cooling of the manikin's body. The asymmetric cooling on the body areas was investigated further for the results obtained when the room temperature was equal to 25°C. The average cooling effect for the upper body segments (right hand (left hand was broken), forearm (right and left), upper arm (right and left), chest (right and left), and back) and for the head (skull, face (right and left), back of neck) was determined. The total area of the upper body segments was 0.68 m^2, of the head it was 0.13 m^2 and of the whole-body it was 1.48 m^2. The results are compared in Figure 17.

The cooling effect of the ceiling fan was the most uniform. The difference in the whole-body cooling effect of the four types of fans is less than 2°C. The cooling effect of the upper body parts is always higher than the cooling effect of the head and the whole-body. The desk and the standing fan generate the largest non-uniformity in the local cooling effect. The head and the upper body parts are substantially cooler than the whole-body. The head is much cooler than the reference condition (11°C for the DF and between 9°C and 10°C for the SF) and it is cooler than the whole-body (8°C for the DF and between 7.5°C and 8.5°C for the SF). The tower fan causes a quite uniform but weak cooling of the body. The whole-body and the upper parts are cooler than the head (between 1 and 2°C cooler). The velocity level does not affect significantly the whole-body cooling effect except for the ceiling fan. The impact of the velocity level on the cooling of the upper body part and the head is also smaller in comparison with the effect of exposure to the flow.

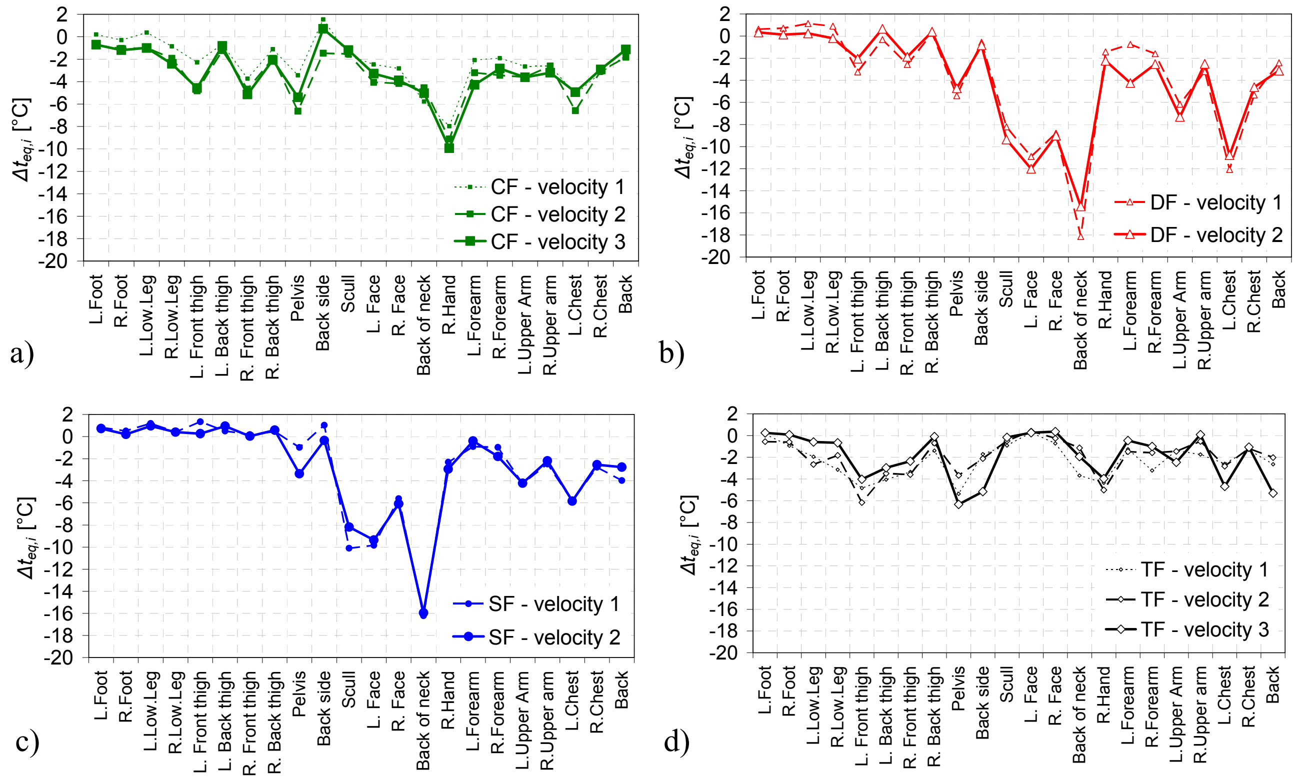

Figure 16 Changing in manikin-based equivalent temperature ($\Delta t_{eq,i}$) on each body part from the reference condition (room temperature equals to 25°C and no devices used for move the air) for the a) ceiling fan (CF); b) desk fan (DF); c) standing fan (SF); and d) tower fan (TF). Step-change control of the fan velocity is possible.

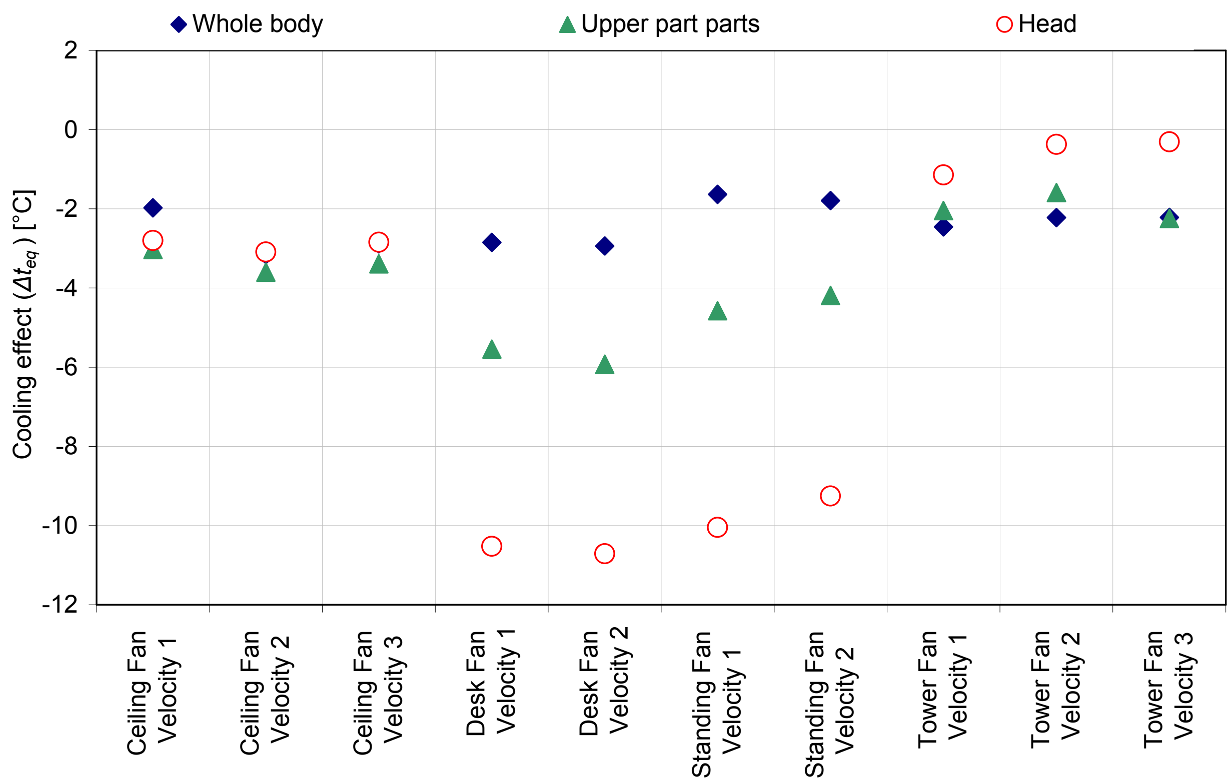

Figure 17. Cooling effect for the whole-body (22 body segments), the upper body part (12 body segments), and the head (4 body segments) for the ceiling, desk, standing and tower fan when the room temperature was set to 25°C.

The air velocities measured at 0.2, 0.6, 1.1 and 1.7 m height above the floor with the CF and the TF are shown in Figure 18-a and with the DF and the SF in Figure 18-b. The air velocity field generated by the four fans is different. The CF generates downward airflow from the ceiling to the floor. The highest velocity (2.2 m/s) is measured at the floor level. Therefore it may be expected that the generated flow will cool mostly the lower part of the manikin (legs and feet). This, however, is not seen from the results of the segmental cooling effect because the air velocity was measured while the manikin was moved away from the desk. Furthermore, the manikin was seated in front of the desk with its legs under the table far from the location of the velocity measurement. The blocking effect of the manikin's body and the interaction between the fan flow and the thermal plume generated by the thermal manikin may have had an impact on the cooling of the body segments. The TF also causes air movement mainly in the lower part of the room. The highest velocity (3.2 m/s) was measured at the flow level.

The desk and the standing fans generated similar air velocity profiles. In both cases the maximum air velocity (2.4 m/s for the DF and 1.8 m/s for the SF) was recorded at 1.1 m above the floor, i.e. the height of the manikin's head. The high velocity at the head level caused the strong non-uniform cooling of the body segment (Figure 17) as already discussed.

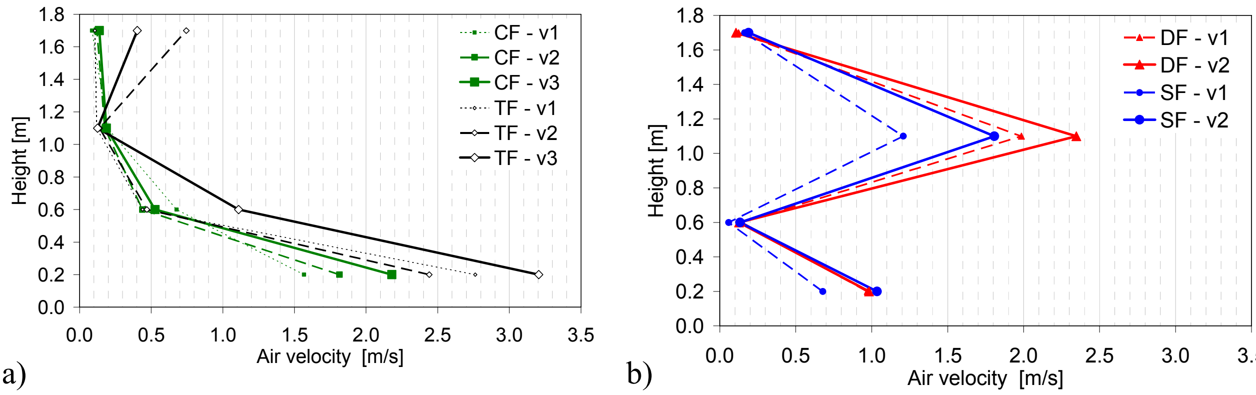

Figure 18 Air velocity measured at 0.2, 0.6, 1.1 and 1.7 m height above the floor where the manikin was located during the experiments for the ceiling and the tower fan (Figure 18-a) and for the desk and the standing fan (Figure 18-b) when the room temperature was set to 25°C.

4.4 Validation of the occupant normalized concentration index

The descriptions of the experimental method and instruments and of the data selected for the validation of the index are described in attached paper F. Hereafter only the results are reported.

Data from two types of total-volume systems (mixing and displacement), an active and concentrate pollution sources, and a PV system using round movable panel as air terminal device were taken from large number of experiments in order to show the potential of the occupant normalized concentration (C) index. The measured normalized concentrations, listed in Table 4, were used in Eq. 5 to calculate the occupant normalized concentrations C_{TV}, C_{PVp}, C_{PVnp} as function of ODW.

An example is shown in Figure 19, when the total-volume system used was mixing ventilation. Previous analyses of this experimental data compared the normalized concentration for ODW=1, i.e. when occupants are steady exposed to the personal ventilation flow (Melikov et al. 2003). With the occupant normalized concentration is possible to quantify the occupant exposure for the whole range of ODW values, from 0 till 1. In Figure 19, can be seen that the introduction of PV does not influence significantly the contaminant distribution in the room and the inhaled air quality of the unprotected occupant does not change appreciably. The PV is able to reduce the contaminant concentration of the occupants protected by PV. The occupant normalized concentration index makes it possible to show and quantify that, due to the higher concentration of pollutant outside the personal airflow, the occupant exposure to contaminant increase with the reduction of ODW.

Table 4. Normalized concentration of human-produced contaminant (SF_6) for mixing ventilation and displacement ventilation. Round movable panel was used as air terminal device.

Normalized Concentration	Mixing	Displacement
c_{TVW}	0.93	0.15
c_{TVS}	1.06	0.76
c_{PVpW}	0.13	0.03
c_{PVnpW}	0.98	0.85
c_{PVS}	1.07	0.9

In Figure 20 is shown the occupant normalized concentrations versus the ODW when total-volume system used was displacement ventilation. The comparison of the results in the figure show that the occupant normalized concentration for displacement ventilation alone at ODW=0.5 is three times higher than at ODW=1, and four time higher than at ODW=0.3. This means that the benefits of displacement ventilation will be lower for minor values of ODW.

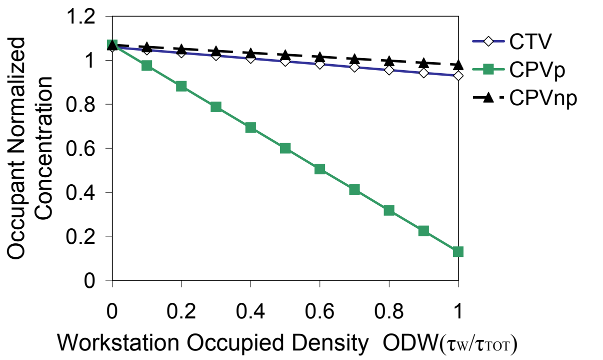

Figure 19. Occupant normalized concentration (C_{TV}, C_{PVp}, C_{PVnp}) versus workstation occupied density (ODW) when the total-volume system used was mixing ventilation.

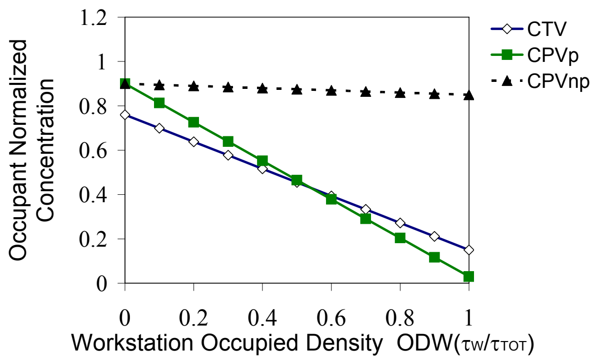

Figure 20. Occupant normalized concentration (C_{TV}, C_{PVp}, C_{PVnp}) versus workstation occupied density (ODW) when the total-volume system used was displacement ventilation.

When ODW=1, the normalized concentration (c_{TVW}) to which a sitting occupant is exposed if only displacement ventilation is used is 0.15 and in the case of combined PV and displacement systems the normalized concentration (c_{PVpW}) of a protected occupant is 0.03. The PV has a ventilation effectiveness that is 5 times higher than the ventilation effectiveness of displacement ventilation and therefore PV is able to provide a better inhaled air quality than displacement ventilation alone. For ODW=0.5 the occupant normalized concentration is the same for the two systems, but the normalized concentration will be almost 2 times higher if the occupant does not use it PV system, i.e. unprotected occupant. For lower values of ODW, displacement ventilation appears to be more effective in providing the occupant better inhaled air quality. These analyses have been performed under steady state conditions, i.e. without disturbance of the displacement pattern due to occupants' walking.

5 DISCUSSION

5.1 Energy saving by increased air movement

From the results of the energy simulation (section 4.1) can be deduced that increasing the air velocity implied a reduction of the energy consumption (Table 3). A saving of the energy need for cooling between 17% and 48% was obtained. The highest percentage of energy saving was obtained in Helsinki for category I of the indoor environment. The lowest percentage of energy saving was obtained in Athens for category III of the indoor environment. The energy savings decreases when the quality of the indoor environment category decreases, e.g. in Bordeaux for category I the saving was 27% and for category III it was 21%. The energy savings decreases with the increase of the cooling degree days (ASHRAE Fundamentals Handbook 2005). The savings increases when the air velocity increases. In fact, the higher savings have been obtained for an air velocity equal to 0.8 m/s. These conclusions can be drawn from Figure 21. In summary, increasing the air velocity to compensate for the higher room temperature is an energy-saving solution that gives a higher performance in high quality indoor environment offices located in a cold climate. This results is in accordance with previously published results (Sekhar 1995, Aynsley 2005, and Atthajariyakul and Lertsatittanakorn 2008). It is interesting to note that, in Helsinki, Berlin and Bordeaux, the energy need for cooling increased with the reduction of the quality of the indoor environment due to the free cooling effect of the outdoor air.

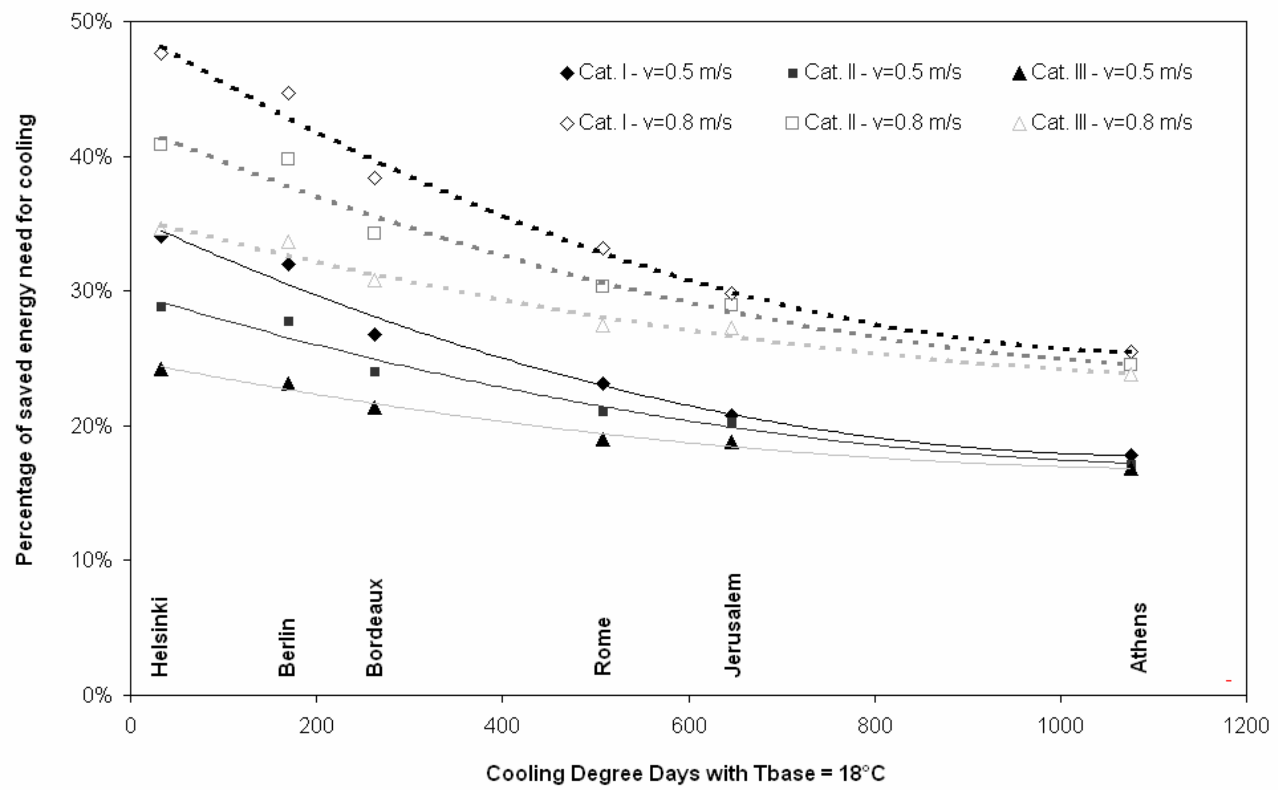

Figure 21. Percentage of saved energy need for cooling vs. cooling degree days. The points are the values obtained from the simulations. The lines are second order polynomial interpolations of the calculated data. The reference case for each category and city is the one without any increase in air velocity (<0.2 m/s).

The fan operation hours are listed in Table 3. The fan operation hours increase with an increase in the number of cooling degree days and with a reduction of the indoor environment category. The fan operation hours are almost independent of the increase of air velocity. In Table 3 the ratio between the fan operating hours and the yearly occupant working hours is shown. The ratio varies between 31% and 93%. High values of the ratio mean that the fan would work also during winter time, when it is presumed that people dress with a clothing insulation equal to 1 clo. In this case the graph, as shown in Figure 1, cannot be applied. However, the fan is working during winter-time in warm climates (Jerusalem and Athens), where the occupant would probably have lighter clothing. Moreover, during winter-time, it is reasonable to think that other techniques would be used to cool the room, such as night free-cooling, or increasing the shading capacity or the thermal mass of the building. The discussion about the humidity level in the room and the reduction of the maximum cooling power can be found in the section "discussion" of the paper A.

5.1.1 Energy consumption of the fan

The air movement increase can be produced by ceiling fans (common nameplate power consumptions around 70W), standing fans (50W), tower fans (40W), desk fans (30W), personal ventilation systems and under certain conditions with operable windows. Measurements of several fans, performed during this project, confirm that the effective input fan power is equal to the value stated on its nameplate.

In order to check whether the electrical consumption of the fan is a critical factor for energy saving, the difference between the saved (in the chiller) and consumed (by the fan) energy is calculated. The saved electrical energy for running the chiller is named $E_{el,Cool}$ and the electrical energy consumed by the fan is named $E_{el,Fan}$. The difference between $E_{el,Cool}$ and $E_{el,Fan}$ is hereafter named net electrical energy saved ($E_{el,Net}$). The saved electrical energy for running the chiller ($E_{el,Cool}$) depends on the saved energy need for cooling (see $E_{N,C}$ in Table 3), on the energy losses from emission, distribution and storage (taken into consideration in the calculations by η) and on the Coefficient Of Performance (COP) of the chiller. COP and η depend on the type of cooling system used and on the building characteristics. The electrical energy consumed by the fan ($E_{el,Fan}$) depends on the electrical input power of the fan (*Pf*) and on the number of fan operating hours (h_i). The net electrical energy saved ($E_{el,Net}$) is defined by Eq. 6.

$$E_{el,Net} = E_{el,Cool} - E_{el,Fan} = \frac{(E^{v\leq 0.2m/s}{}_{N,C} - E^{v=i}{}_{N,C})(1+\eta)}{COP} - 10^{-4} P_f h_i \quad (i\text{=0.5 or 0.8 m/s}) \qquad \text{Eq. 6}$$

Where $E_{el,Net}$ is the net electrical energy saved [kWh/(m^{2}y)];
$E^{v=i}{}_{N,C}$ is the energy need for cooling ($E_{N,C}$) obtained when the air velocity is $i \leq 0.2$ or $i = 0.5$ or $i = 0.8$ m/s [kWh/(m^{2}y)];
P_f is the electrical input power of the fan [W];
h_i is the number of hours that the fan is operating (Table 3) [hour];
η is the ratio between the energy need for cooling and the thermal energy that the chiller has to produce [-];
COP is the coefficient of performance of the chiller [-].

Practical experience shows that the COP can vary within the range between 2.5 and 4.5 with a best guess value of 3.5 and the η can vary within the range between 0 and 0.15 with a best guess value of 0.05. The influence of these two parameters on the net electrical energy saved, $E_{el,Net}$, was calculated for Helsinki in the case of the indoor environment category I for velocity elevated to 0.5 m/s and 0.8 m/s. From the results shown in Figure 22 it can be seen that $E_{el,Net}$ varies as a function of the COP and η for the two air velocities.
The results in Figure 22 reveal that COP has a significant influence on the net electrical energy saved, and η has less impact. Moreover, it can be seen that $E_{el,Net}$ is lower for higher values of COP, is due to the fact that the required electrical energy for producing a certain amount of cooling energy decreases with the increase of the COP.

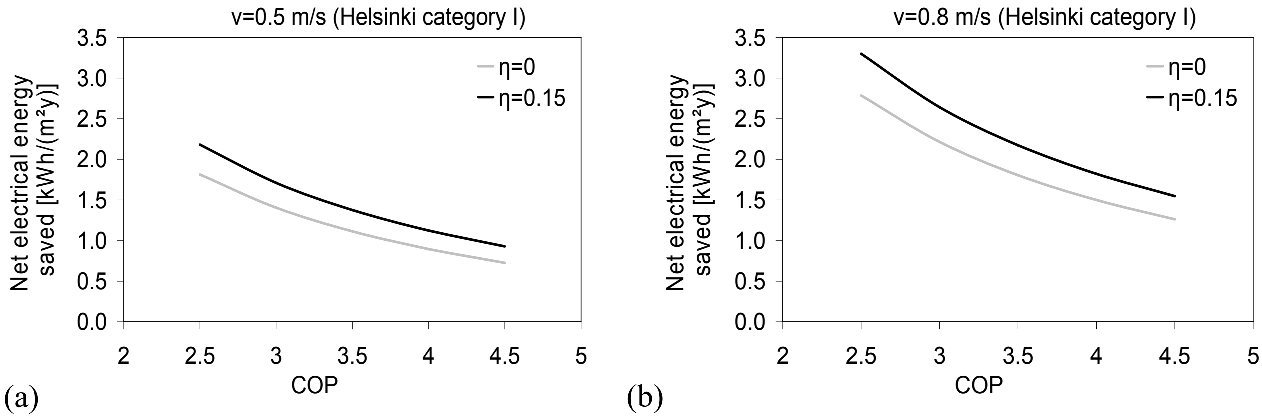

Figure 22. The net electrical energy saved ($E_{el,Net}$) calculated for Helsinki for category I versus the COP for η equal to 0 or 0.15 for air velocity of 0.5 m/s (a) and 0.8 m/s (b).

Easy-to-use graphs for checking, as a rule of thumb, how much energy can be saved as a function of the fan input power are shown in Figure 23.

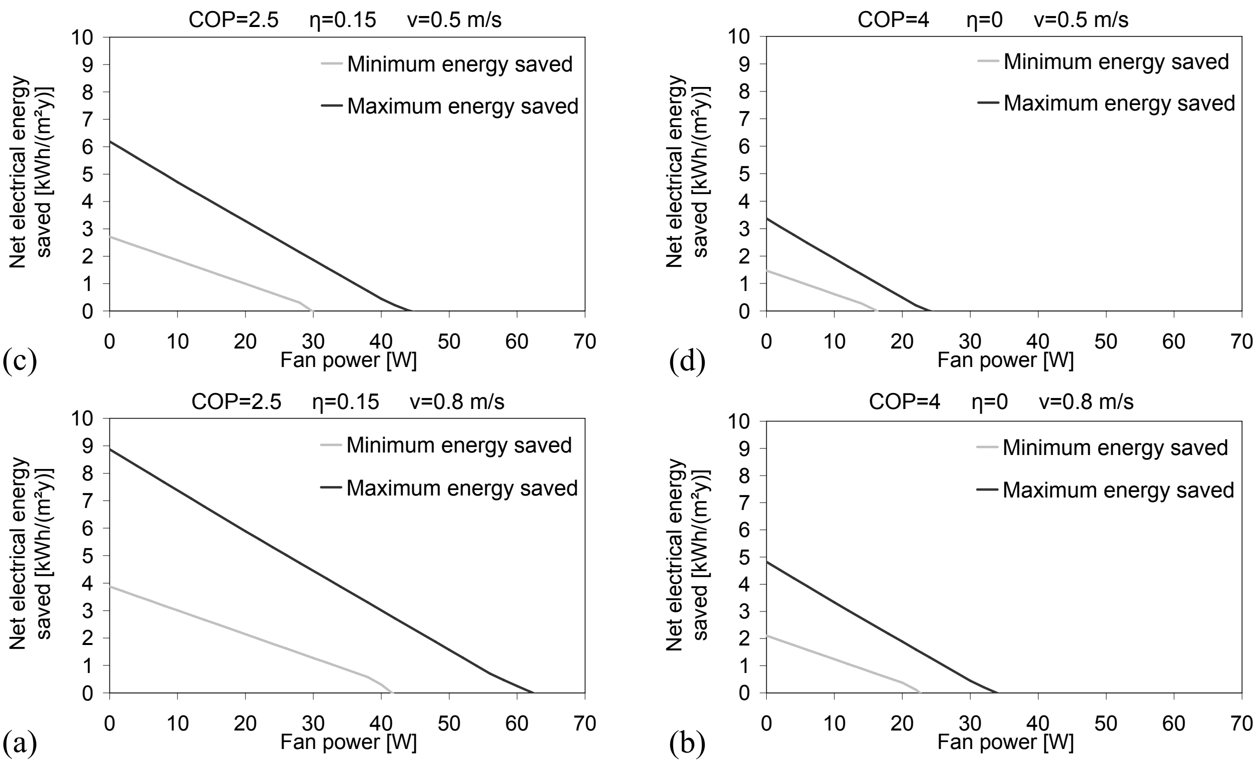

Figure 23. The net electrical energy saved vs fan input power when: a) COP=2.5, η=0.15 and air velocity=0.8 m/s; b) COP=4, η=0 and air velocity=0.8 m/s; c) COP=2.5, η=0.15 and air velocity=0.5 m/s; and d) COP=4, η=0 and air velocity=0.5 m/s.

In Figure 23 four cases are reported, including two air velocities (0.5 and 0.8 m/s) and two combinations of COP and η. The combinations of COP and η were chosen in order to calculate the extreme cases. With COP=2.5 and η=0.15 the $E_{el,Net}$ is the highest, while with COP=4 and η=0 the $E_{el,Net}$ is the lowest. The net electrical energy saved ($E_{el,Net}$) was calculated for a fan input power within the range 2-70 W for all the fifty-four simulated cases. The maximum and minimum values for each fan input power has been plotted. The use of these graphs is explained in the following example. If the input power of the fan is 20 W, the COP is equal to 2.5, η=0.15 and the air velocity is 0.8 m/s (Figure 23a), the expected net electrical energy saved is then at minimum 2.1 kWh/(m^2y) and at maximum 5.9 kWh/(m^2y). On the other hand, with the same fan input power, if the COP is equal to 4, η=0 and the air velocity remains the same (Figure 23b), the expected net electrical energy saved is then at minimum 0.4 kWh/(m^2y) and at maximum 1.9 kWh/(m^2y). If the input power of the fan is still 20 W, the COP is equal to 4, η=0 and the air velocity is 0.5 m/s (Figure 23d), the expected net electrical energy saved is then at maximum 0.5 kWh/(m^2y). In this case, the minimum is not plotted because there is no energy saving but energy waste. The values plotted in Figure 23 were obtained from computer simulations where the human behaviour was not modelled. The human behaviour (e.g. leaving the fan switched on when the occupant is out of the office) affects the possibility of saving energy by using the technological solution studied in this paper. The main advantage of the presentations in Figure 23 is that the graphs are independent of the location and of the indoor environment category and can therefore give a first estimation of the saving. For example, if the fan power input is 60 W, then it can be easily seen that energy savings cannot be achieved. From the figures, it can be concluded that traditional systems, such as ceiling fans (70W) and standing fans (50 W), cannot be used to save energy on the basis of assumptions made in this study. From Figure 23 it can be seen that for the conditions considered in this study (outdoor climate, indoor environment category, air velocity increase) and for the range of COP and η used, it is never possible to reach a net energy saving with a fan input power higher than 60 W. On the other hand, it is always possible to save energy if the input power is lower than 15 W. Calculations made for the best guess values for COP and η, respectively 3.5 and 0.05, reveal that energy savings will not be achieved with fans using more than 20 W. This can be done using a small desk fan or a personal ventilation system. The main conclusion is that the fan input power is a critical factor for the applicability of this solution in practice.

5.2 Personalized ventilation energy consumption

5.2.1 Influence of the supply air temperature on energy need

The results shown in Figure 7 reveal that the simulated building needs mainly cooling. Room Heating is needed only for the reference case (mixing ventilation supplying air at 16°C). The building has a good insulation and air tightness and the internal heat gains are sufficient to maintain the required operative temperature. The supplied personalized air needs to be cooled only sporadically; in fact AHU Cooling is equal to zero except for the reference case. The supply personalized air temperature and its control strategy have a marked influence on energy consumption. The energy need for the simulated cases is in the range 39.0-89.2 kWh/(m^2y). The energy need for the reference case is 24.3 kWh/(m^2y); it means that by using PV the energy need increases from 61% to 268%. This is mainly due to the fact that the lowest supply air temperature for the PV system was limited to 20°C for comfort reasons. In the reference case the air is supplied at 16°C. The building needs mainly cooling and the need for warming the personalized supplied air up to 20°C is a heat load (AHU Heating) that later has to be removed by the cooling system (Room Cooling). This can be seen in Figure 7 by subtracting the AHU Heating from the Room Cooling; the remaining Room Cooling is almost constant in the range between 23.2 and 25.2 kWh/(m^2y). To supply the air at an elevated temperature of 23°C or 26°C (Cases 2 and 3) required a greater amount of energy than to supply at 20°C (Case 1). The energy needs for Cases 1, 4, 5, and 6 are almost equal, i.e. the different supply air temperature control strategies do not differ with regard to the energy need. The reason can be understood by analysing the outdoor air temperature cumulative profile. In Copenhagen the outdoor air temperature is higher than 20°C only 3.2% of the time in one year, higher than 22°C only 1.3%, higher than 24°C only 0.5%, and higher than 26°C only 0.1% of the year. Therefore, controlling the supply air temperature, θ_{SUP}, based on the outdoor temperature, θ_{ODA}, using profiles that differ only for θ_{ODA}>20°C, does not make any significant difference with regard to energy need. Controlling the θ_{SUP} by the indoor air temperature, θ_{IDA} (Case 7 and Case 8) implies high energy consumption. Case 7 has an energy need almost equal to Case 2, where θ_{SUP} = 23°C, but from a thermal comfort point of view, it would perform better. For the simulated building

and for the assumptions made in this paper, the best supply air temperature control strategy is to provide air constantly at 20°C, the minimum permissible supply temperature.
The supply air temperature of a personal ventilation system has a marked influence on the energy consumption because it may become a significant heat load that needs to be removed. In a mixing ventilation system the outdoor air, after been conditioned, can be mixed with the recirculated air to reach the desired supply air temperature. This cannot be done with a PV system if its main aim is to improve significantly the inhaled air quality and to reduce the risk of spread of diseases.

5.2.2 Analysis of the energy-saving strategies

The energy-saving strategies with personalized ventilation were studied with Cases 9 – 26 (Table 2), as defined in the Method section, sub-section "Energy-saving strategies". The results are shown in Figure 8, Figure 9, Figure 10.

The influence of reducing the personalized flow rate, q_V, thanks to the higher ventilation effectiveness on the energy need, is shown in Figure 8. In Cases 9, 10, 11 and 1 q_V is equal to 5, 10, 15 and 20 l/s per person respectively, and the θ_{SUP} is in all cases constant and equal to 20°C. In Cases 12, 13, 14 and 7 q_V is equal to 5, 10, 15 and 20 l/s per person respectively, and the θ_{SUP} is a function of the θ_{IDA} and varies according to the "comfort" profile (see Figure 3, Case 7). In all cases the energy need is determined mainly by the AHU Heating and the Room Cooling. From Figure 8 it can be deduced that reducing q_V implies: a reduction of AHU Heating because the amount of outdoor air that needs to be heated is reduced and an increase of the Room Cooling because the outdoor air has a free cooling effect. Therefore, reducing q_V is beneficial only when the decrement in AHU Heating is higher than the increment in the Room Cooling. This is valid for the Cases 12, 13, 14, and 7 but not for the Cases 9, 10, 11, and 1 because the supply air does not need to be warmed up more than 20°C. When the θ_{SUP} is kept constant and equal to 20°C (Cases 1, 11, 10, 9) the energy need increases from 39.2 kWh/(m^{2}y) to 49.3 kWh/(m^{2}y) with the decrease of q_V from 20 to 5 l/s per person which corresponds to 26% of energy penalty. In this case it is not an advantage to reduce the airflow because the supplied air has a free cooling effect. When θ_{SUP} follows the "comfort" profile the energy need slightly decreases from 60.2 kWh/(m^{2}y) to 55.2 kWh/(m^{2}y) with the decrease of q_V from 20 to 5 l/s per person. In this case energy is reduced by 8% and it is an advantage to reduce the airflow. In conclusion, in a cold climate, reducing the personalized airflow rate does not always lead to a reduction of energy need because the outdoor air may have a free cooling effect. PV requires more energy than the reference case (mixing ventilation) even if the temperature of the supplied personalized air follows the applied "comfort" profile. However, it is believed that reducing q_V would always lead to energy-saving in hot and humid climates.

The influence of extending the upper room operative temperature, θ_{UP}, on energy need is shown in Figure 9. In Cases 1, 15, 16, 17 and 18 θ_{UP} is equal to 25.5, 27, 28, 29, and 30°C respectively, and the personalized supply air temperature, θ_{SUP}, is constant and equal to 20°C. In Cases 7, 19, 20, 21 and 22 θ_{UP} is equal to 25.5, 27, 28, 29, and 30°C respectively, and the θ_{SUP} follows the "comfort" profile. Also in these cases the energy need is determined mainly by the AHU Heating and the Room Cooling. From Figure 9 it can be deduced that increasing θ_{UP} implies a significant decrease of the Room Cooling, but it does not affect substantially the AHU Heating. Therefore, extending the upper room operative temperature limit is always beneficial. Independently of the θ_{SUP} strategies, the extension of θ_{UP} leads to energy need reduction, and when θ_{UP} is equal or higher than 28°C, using the personal ventilation system implies less energy need than the reference case of mixing ventilation.

The results in Figure 9 show that when the θ_{SUP} is kept constant and equal to 20°C and θ_{UP} is increased from 25.5 to 30°C, the energy need decreases from 39.2 kWh/(m^{2}y) to 9.9 kWh/(m^{2}y), corresponding to 75% of energy-saving (Cases 1, 15, 16, 17, 18). When θ_{SUP} follows the "comfort" profile (Figure 3) and θ_{UP} is increased from 25.5 to 30°C the energy need decreases from 60.2 kWh/(m^{2}y) to 12.7 kWh/(m^{2}y), corresponding to 79% of energy-saving (Cases 7, 19, 20, 21, 22). This energy-saving strategy is an effective way of reducing the energy need. However, it can be recommended only in the working environment where the occupants spend most of their time at their workstation in a comfortable thermal environment achieved by personalized ventilation.

The influence of supplying the personalized air only when the occupant is at the desk is shown in Figure 10. Ref.* and Ref.** are the energy needs for the reference case (mixing ventilation) when the internal heat load generated by occupants and equipment follows the occupancy profiles reported in Figure 4 and Figure 5 and the ventilation airflow is constant. This leads to an energy decrease from 24.3 kWh/(m^{2}y) to 22.6 kWh/(m^{2}y) for the Ref.* case and to 20.2 kWh/(m^{2}y) for the Ref.** case. This means that the reduction of the internal heat load generated by occupants and equipment implies a reduction of 7% and 17% respectively for the occupancy profiles shown in Figure 4 and Figure 5. The energy need for the reference case was recalculated in order to be comparable (same internal heat load) with the energy need with the PV. Supplying the personalized air only when the occupant is at the desk implies lower airflow rates. As in the previous cases (Cases 9-14) the reduction of the airflow rate causes two effects, a reduction of the AHU Heating (less outdoor air needs to be warmed up) and an increase of the Room Cooling (reduced free cooling). From Figure 10 it can be seen that for both occupancy profiles it is not effective to supply the airflow rate only when people are at the desk. When the airflow rate is adjusted according to the occupancy profile shown in Figure 4, the energy need slightly increases from 37 kWh/(m^{2}y) to 38.3 kWh/(m^{2}y), corresponding to 3% of energy penalty (Cases 23 and 24). When the airflow rate is adjusted according to the occupancy profile shown in Figure 5, the energy need increases slightly from 31.1 kWh/(m^{2}y) to 33.9 kWh/(m^{2}y), corresponding to 9% of energy penalty (Cases 25 and 26). This energy-saving strategy is not effective for reducing the energy need.

In conclusion, the energy consumption with personalized ventilation may increase substantially (between 61% and 268%) compared to mixing ventilation alone if energy-saving strategies are not applied. Among the studied energy-saving strategies the most effective way of saving energy with personalized ventilation is to increase the maximum permissible room temperature (saving up to 60% compared to the mixing ventilation). Reducing the airflow rate does not always imply a reduction of energy consumption because the outdoor air may have a free cooling effect. Supplying the personalized air only when occupants are at the desk is not an effective energy-saving strategy.

5.3 Cooling fan efficiency index

In the paragraph 5.1.1 it is discussed the influence of the energy consumption of the cooling fan on the energy saving obtained by increased room temperature limits. Due to different design, installation and use the performance of cooling fans with regard to their cooling effect can be quite different. As Figure 13 show, at the same cooling effect the power consumption of different fans can be different as well. The cooling fan efficiency index makes it possible for the first time to evaluate and compare cooling fans. This index combines in a single value the fan performance with regard to its cooling effect and its energy use. The experiments performed with four cooling fans of different design available on the market, i.e. ceiling, desk, tower and floor standing fans, document that the cooling fan efficiency index is sensitive enough in identifying differences in the performance of the cooling devices. The body cooling effect caused by the fans was different. The ceiling fan and the desk fan had a rather similar cooling effect which was substantially higher than the cooling effect of the floor standing fan and the tower fan. However, the electrical power used by the desk fan was twice as low as that used by the ceiling fan, and the desk fan therefore had a significantly higher cooling fan efficiency index than the remaining three fans. The *CFE* index can be used by HVAC engineers and policymakers as well as for classifying fans according to their performance.

The desk fan was found to have the highest efficiency index of the four tested fans (Figure 11 and Figure 12). The whole-body cooling effect of this fan was largest. The non-uniformity of the local cooling effect of this fan was also greatest, with the head region being mostly cooled. It may be suggested to use the head cooling effect together with the cooling fan efficiency index when assessing the performance of cooling fans because the head is an active heat dissipater and in warm environments the whole-body thermal sensation follows the head region thermal sensation closely (Melikov et al. 2004 a and b; Arens et al. 2006; Watanabe et al. 2008a). Thus, at the same efficiency, the performance of the fan that provides greater cooling of the head may be considered to be better. However, these selection criteria may fail to be correct in practice because human response to airflow from the front and from the back is different.

In this thesis, the cooling effect of air movement has been quantified by measuring the dry heat loss. The evaporative heat loss has not been taken into account because the thermal manikin used cannot sweat. Several studies have used dry heat loss measured by a thermal manikin to quantify the cooling effects of air movement on the human body. Tsuzuki et al. (1999) studied the performance of three designs of task ambient air-conditioning systems and found that the cooling effect of the combined evaporative and sensible cooling may double the total whole-body cooling rate due to dry heat loss alone when 20% of the surface was wet. The cooling effect of the evaporative heat loss will increase with the increase of the room temperature. In the future, the determination of fan efficiency can be made more accurately by sweating thermal manikins. The sweat glands are not uniformly distributed over the human body. Therefore, use of the thermal manikins available today with simulated sweat glands on the surface areas corresponding to the site of the human skin where they are most dense can be considered.

A considerable number of studies focused on the use of fans to cool people in a warm environment (McIntyre 1978 and 1979, Rohles et al. 1983; Jones et al. 1986; Tanabe and Kimura 1987; Scheatzle et al. 1989; Bauman et al. 1993; Melikov et al. 1994 a and b; Fountain et al. 1994; Arens et al. 1998; Szokolay 1998; Tsuzuki et al. 1999; Khedari et al. 2000; Hayashi et al. 2004; Sekhar et al. 2005; Aynsley 2005 and 2007; Atthajariyakul and Lertsatittanakorn 2008; Sun et al. 2007 and 2008; Watanabe et al. 2008a and b). Only in one study was the fan power reported (Sun et al. 2008). The power consumption of cooling fans is considered negligible (usually less then 90 W) and therefore it is not reported. However, as already discussed in 5.1.1, it has been demonstrated that the required power input of cooling fans is a critical factor for an energy-saving strategy used in warm environments.

In the section 5.1.1 it has been shown that in some buildings the use of cooling fans with power input of more than 20 W will actually increase the energy consumption compared to the energy consumption needed to cool the whole building. For the same cooling effect the power input of the desk fan tested in the present study was 16-20 W, i.e. twice as low as the power input of the ceiling fan (approx. 40 W) and therefore its cooling fan efficiency index was twice as high. Nevertheless, one should be cautious when recommending the use of the desk fan instead of the ceiling fan. The ceiling fan may provide cooling to several occupants while the desk fan provides cooling to only one occupant. Individual control with a ceiling fan is difficult to achieve in practice when it aims to provide cooling to several occupants who may have different preferences with regard to the air movement. The development of desk fans with a strong cooling capacity and low energy consumption of a few watts, as for example the fans used by Watanabe et al. (2008a) and Sun et al. (2008), is recommended.

The convection heat loss from the body with cooling fans is mainly based on the velocity and the turbulence intensity of the generated flow. As discussed, the fans tested in the present study generated a non-uniform flow. The velocity distribution at the location of the thermal manikin was rather different as well. The CF and the TF generated flow with the highest velocity near the floor, up to 0.6 m above the floor, while the highest velocity generated by the DF and the SF was measured at the head region. The indoor climate standards recommend individual control of the airflow at elevated velocity. Velocity control at two or three levels was provided for the fans tested. The control, however, affected the flow mostly in the high velocity region, i.e. near the floor for the CF and the TF and at the head region for the DF and the SF, and therefore resulted mainly in an increase of the local cooling of the body segments exposed to the flow and affected only slightly the whole-body cooling (Figure 17). In this respect the layout, furniture arrangement, etc. were also factors affecting the local air distribution around the manikin's body.

5.3.1 Limitation of thermal comfort standard

As already discussed (see paragraph 1.3), elevated air speed under individual control is recommended in the present indoor climate standards (ASHRAE 55-2004; ISO 7730-2005; CEN 15251-2007) for providing occupants with thermal comfort in warm environments. The relationship between the air speed and the upper operative temperature limits (see Figure 1) provided in the standards is described in the section 1.3. The relationship is based on the assumption that a uniform air velocity field hit the human body.

Table 5. SET* calculated with the data collected for determining the CFE of the four cooling fans.

Measuring height [m]	Air velocity [m/s]	Type of Fan	SET* [°C]	Cooling effect [°C]
0.2	1.35	Ceiling fan	23.4	3.1
0.6	0.32	Ceiling fan	25.5	1
1.1	0.14	Ceiling fan	26.5	0
1.7	0.13	Ceiling fan	26.5	0
0.2	0.74	Desk fan	24.3	2.2
0.6	0.1	Desk fan	26.5	0
1.1	1.76	Desk fan	23	3.5
1.7	0.11	Desk fan	26.5	0
0.2	1.27	Standing fan	23.5	3
0.6	0.18	Standing fan	26.4	0.1
1.1	1.77	Standing fan	23	3.5
1.7	0.12	Standing fan	26.5	0
0.2	3.27	Tower fan	22.4	4.1
0.6	0.77	Tower fan	24.2	2.3
1.1	0.27	Tower fan	25.7	0.8
1.7	0.12	Tower fan	26.5	0
	0.15	**none**	**26.5**	

The relationship is not easily usable in practice when cooling fans are applied because, as shown in Figure 16 and Figure 18, the body cooling by such fans is non-uniform due to large non-uniformity in the generated velocity field. The velocity field and its direction cannot be described with a single value. Therefore, it is not clear how to apply in practice the recommendations in the standards. Other methods for quantification of the cooling effect of air movement have been suggested as well (Szokolay 1998; Aynsley 2007). Aynsley (2007) proposed to use the SET* index (Gagge et al. 1971) since it includes the impact of humidity and the thermal insulation of clothing which are not considered in the relationship for elevated velocity included in the present standards. However, this approach has the same limitation, namely that there is no unique velocity which can describe the complex air velocity field generated by cooling fans. This is demonstrated with the following example, based on the data collected for determining the *CFE* of the four cooling fans. The air velocity values of the four tested fans, measured when the mean radiant and air temperatures were equal to 27°C and the velocity level was one, were used to calculate the SET* index. The measured relative humidity was equal to 26%, the clothing thermal insulation of the manikin was 0.62 clo (including the thermal insulation of the chair) and the activity level was 1.1 met. The results of the calculations are listed in Table 5. The SET* calculations were performed with ASHRAE's thermal comfort program (Fountain and Huizenga 1994).

The SET* calculated with the velocities measured at different heights with each of the fans is substantially different (up to 4.1°C in the case of the tower fan). The indoor climate standards specify using measurements at 0.6 m height (sedentary person) in order to predict occupants' thermal comfort (PMV-PPD index, etc.). At this height the SET* values for the desk and standing fans are almost null (because the desk shades the occupant's body at that height), but their whole-body cooling effects measured with the thermal manikin (Table C-1 in Appendix C of paper C) are strong (varied between -1.4 and -2.9°C). It is clear that the approach recommended in the present standards, as well as the SET*, cannot be used directly in practice. This issue needs to be carefully considered and addressed in the standards.

5.4 Occupant normalized concentration index

Many studies focused on the measurement and/or on statistical modeling of the time an occupant stay in a room over the working time (Bauman et al. 1994; Nobe et al. 2002; Bernard et al. 2003; Johansson et al. 2004; Wang et al. 2005; Nakagawa et al. 2007; Nagareda et al. 2007; Melikov and Hlavaty 2007; Halvarsson et al. 2006; Page et al. 2008). To the knowledge of the authors only four studies reported on the time occupants in office buildings spend at the workstation over the time they stay in the office (Nobe et al. 2002, Nakagawa et al. 2007; Nagareda et al. 2007; and Melikov and Hlavaty 2007). Usually, the time spent in the office and the working time are not coincident. Among the four studies the one of Nobe et al. (2002) has the highest number of data. They measured the average seat occupancy rate in a large scale office in Japan. 240 workstations were monitored, during weekday office hours for the attendant occupants only (the outing persons were removed). The results were classified in a function of the type of occupants' activity. It was obtained that for clerical work the average value of ODW was equal to 0.47, for technical work ODW was equal to 0.37, for business work ODW was equal to 0.31. This indicates that occupants stay at the workstation less often than away from it. Moreover the time an occupant spent at the desk was found to depend on the type of job, e.g. the ODW could be related to the type of human activity.

Figure 19 and Figure 20 show that the occupant exposure to pollutant depends also on the occupied density. Comparing only the performance of a total-volume and PV for ODW=1 is not enough. In order to accurately assess the performance of PV the concentration of pollution at the workstation (typically in inhaled air) as well as in the rest of the room should be reported. In this way, it will be possible to accurately assess the occupant's exposure to contaminants considering also ODW.

Values of ODW lower than 0.5 indicate a strong influence of the pollution concentration in the room away from the workstation on the occupant's exposure. Therefore, the performance of PV with regard to inhaled air quality should be evaluated based on at least two criteria: first its ability to provide 100% clean air in inhalation (ODW=1) and second, on its ability to avoid an increase of pollutant concentration in the background region, measured at 1.7 m, compared to the total-volume system alone (ODW<0.5). For example, in the case of Figure 20, Melikov et al. (2003) underlined that PV generate an higher concentration of pollutant at 1.7 m than displacement ventilation alone because it promotes mixing of contaminants located in its vicinity. When ODW is lower than 0.5, the occupant exposure will be lower for displacement ventilation alone than with the personal ventilation system. For ODW=0.3, corresponding to business work according to Nobe et al. (2002), the occupant normalized concentration of displacement ventilation alone is 0.54 while for the PV system is 0.64. The occupant normalized concentration index makes it possible to assess more realistically occupants' exposure in a room based on non-uniformity in pollution distribution in the room and occupant activity.

The database providing occupant density as a function of occupant activity is so far limited. The considerations reported in this thesis are based on the data collected by Nobe et al. (2002). The database should be expanded and based on the country because the work values different significantly from country to country.

The occupied density concept, proposed in this thesis, has been applied to the evaluation of the inhaled air quality in room with non-uniform pollutant concentration. It may be interesting to apply the occupied density concept to the assessment of the thermal comfort in room characterized by non-uniform thermal conditions.

5.5 Disadvantages of personalized ventilation

Among the several positive aspects related to personalized ventilation, there are some facets that need to be considered when a designer wants to install a personal ventilation system.

The main aim of personalized ventilation is to supply clean air close to the occupants. This means that the outdoor air should be transported through ducts to the personal ventilation Air Terminal Device (ATD). The installation of the ducts from the Air Handling Unit to the ATDs may be difficult and it may increase the installing costs. In addition, it may increase the pressure loss and it could make the refurbishment (changes in the office layout) more complex and costly. Moreover, there could be also problems related to the aesthetic (it may not be easy to integrate the ducts in the office layout). Some solutions to this problem have been developed. Coupling personalized ventilation with an Under-Floor Air Distribution (UFAD) system may be a solution if the outdoor air is provided to the workstation using ducted system. The recirculated and conditioned air is distributed in the room by a pressurized plenum, and the outdoor is air is supply to the ATD through ducts located in the plenum. The main limitations of this solution are related to the initial costs and to UFAD system. Even if UFAD is a growing technology (estimated 6% of new offices building in North America are equipped with UFAD system), it is still not yet a mature solution. Problems related to plenum air leakage and thermal decay, and inadequate control and operating strategies need to be solved. Another solution has been proposed by Halvoňová and Melikov (2008). This solution has been described in the paragraph 1.5.2. It is a novel idea and it needs further developments and field studies.

PV is a quite new ventilation strategy, there are few installations in the world, and therefore, there is a lack of design, operation and maintenance knowledge. According to some PV manufacturers and from the experience from the first marketed PV systems the aesthetic of the air terminal devices and their integration into the desk are problems that need to be considered carefully.

According to the author the main problems related to PV could be due to its integration with the HVAC system, and thus, to the energy consumption. When the PV system has to supply only outdoor air without any recirculation, it is unavoidable to install another system to control the indoor temperature and humidity because the PV system will not be able to manage all the sensible and latent loads. Using two independent HVAC systems is a common practice in Europe, where the indoor temperature is usually controlled with a water-based system (e.g. radiant floor/ceiling panels, two/four pipes fan coils, radiators, split unit, etc.). In US and Asia this is not a common practice, as

all-air systems are generally applied. Using only one system reduces the initial costs but it increases the energy consumption. Where all-air systems are used, the installation of another air handling unit dedicated to personalized ventilation may significantly increase the initial cost. Even if a water-based system is already installed there could be the need of installing two AHUs for providing clean air to the occupants that are not often at the desk. Cermak (2004) proposed as a design strategy the integration of personalized ventilation with a total-volume system (mixing, or displacement or UFAD). Also in this case two AHUs would be needed.
The initial cost is not the only problem related to use two AHUs or an AHU working with a water based system, energy problem could be bigger. As described in the section 5.2.2 (simultaneously heating and cooling), the interaction of two systems may lead to an energy waste, e.g. when a system is heating the supplied air the other it is trying to cool the room air, or vice versa. In some case this could be avoided with proper control strategies, but as in the results shown in section 5.2.2 sometimes is unavoidable. Personalized ventilation gives the opportunity to occupants to control their thermal microenvironment by changing the air velocity and jet direction, some PV solutions may be equipped with local electrical devices for heating and/or cooling the air as thermoelectric cooling (i.e. Peltier cooler) or electric heater. Electrical devices may be highly energy consuming. Depending on the country electrical energy production system, the primary energy use may be 2-4 times higher than the produced unit of electrical energy. When possible, it would be better to cool or heat the air with a coil and with a heat recovery unit. Moreover, if the PV users are not properly trained they may control the system disregarding the energy consumption of the overall system (e.g. local cooling in winter or local heating in summer), this also may lead to higher energy consumption.

6 CONCLUSIONS

The objectives of the present project were to study, by mean of computer simulation, the energy saving of increased air movement and the energy consumption of a personalized ventilation system; to develop and test, by means of laboratory measurements, an index for the evaluation of the cooling fan efficiency and another index for the evaluation of air quality improvements in rooms with personalized ventilation taking into account the occupant location pattern.

The effectiveness of increasing the maximum allowed indoor temperature thanks to the cooling effect of elevated air speed has been studied for a wide range of climates, indoor environment categories and air velocity levels. Cooling energy savings in the range of 17-48% have been obtained. The percentage of savings increases when: the air velocity increases, the indoor environment category level increases, and the number of cooling degree days decreases. A reduction of the maximum cooling power in the range 10-28% has also been obtained. The results reveal that the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature. Under the assumptions of this study, the energy saving may not be achieved with the methods for air speed increase, such as ceiling, standing, tower and desk fans widely used today when the power of the fan is higher than 20 W. More efficient (high cooling effect and low energy consumption) fans should be developed.

Computer energy simulations of a personal ventilation system installed in a high quality Scandinavian building located in a cold climate have been performed. The results showed that the control strategy of the supplied personalized air temperature has a significant influence on energy consumption. The energy consumption with personalized ventilation increases substantially (between 61% and 268%) compared to mixing ventilation alone when energy-saving strategies are not applied. Among the studied energy-saving strategies the most effective way of saving energy with personalized ventilation is to increase the maximum permissible room temperature (saving up to 60% compared to the mixing ventilation) but it can be applied only in offices where occupants spend most of their time at the desk. Reducing the airflow rate does not always imply a reduction of energy consumption because the outdoor air may have a free cooling effect. Supplying the personalized air only when occupants are at the desk is not an effective energy-saving strategy. By using an optimization software it was obtained that the best supply air temperature control strategy is to provide air constantly at 20°C, the minimum permissible supply temperature.

A new index, named “cooling fan efficiency” defined as the ratio between the cooling effect (measured with a thermal manikin) of the used device and its power consumption has been introduced for evaluation of the performance of cooling fans. The index was determined for a ceiling fan, a desk fan, a standing fan and a tower fan in a real office at three room air temperatures and at different fan velocity levels. The results revealed that the index is sensitive enough to identify differences in the performance of the cooling devices. It has been measured a large variation in the whole-body cooling effect (between -3.2 and -0.4°C), in the fan power (between 15.6 and 49.3 W), and in the cooling fan efficiency index (between 0.009 and 0.177°C/W). The desk fan had a significantly ($p<0.01$) higher efficiency than the other three fans tested. The cooling effect of fans decreases with the increase of the room temperature. A standard method for testing the performance of cooling fans with regard to their cooling effect and power input needs to be developed.

The air velocity field and the local cooling effect for body segments caused by the fans were strongly non-uniform. A single value cannot summarize the complex air velocity field. This makes the recommendation in the standards (see Figure 1) for elevated velocity in warm environments difficult to use in practice. The present thermal comfort standards need to be revised to better address the issue of elevated air velocity in warm environments.

The personal ventilation decreases the pollutants concentration mostly in the microenvironment at the workstation. Therefore, occupant's exposure to pollutant depends on the ratio of time occupant stays at the workstation over total time he/she stays in the room. In this study an index has been developed, named "occupant normalized concentration", which makes it possible to assess more realistically occupant's exposure in a room characterized by a non-uniform pollution distribution. The index can be used to compare and quantify the variation in terms of inhaled pollution by occupant in a room with PV in conjunction with a total-volume ventilation system. The results of the application of the index to data collected during full-scale room measurements showed that it can be used at the design stage for assessing the benefits of PV when applied in practice for office buildings with different occupation patterns. It has been demonstrated that displacement ventilation alone was able to provide to the occupant better inhaled air quality than displacement ventilation in conjunction with PV when occupied density is lower than 0.5.

From the results reported in this work the following recommendations can be outlined. The recommendations are valid within the assumptions made in this work.

- The power of a fan used to cool people by increasing air movement should be in general less than 20 W.
- The supply air temperature control strategy for a personal ventilation system should be taken in care consideration because it may strongly affect the energy consumption in cold climate.
- The best supply air temperature control strategy for a personal ventilation system placed in a cold climate is to provide air constantly at 20°C.
- The desk fan used in this project is more effective to cool people that other tested fans (ceiling, standing and tower).
- The tracer gas concentration at 1.7 m should be measured (far from the personalized ventilation station) and reported when the ventilation effectiveness is tested in order to check if the PV system increases the pollution concentration in the room. This value can be used in the occupant normalized concentration index.

This work attempts to make a contribution to the solution of some problems related to personalized ventilation and cooling fans. Due to the limitations of this study, further research is requested on following topics.

- In this research, it was studied the energy behaviour of a personalized ventilation system installed in a high quality Scandinavian building located in a cold climate. The energy simulations are strongly sensitive to the climate and to building features. The results obtained in the simulation cannot be extrapolated and applied to different boundary conditions. Energy simulations in other climates and for different building characteristics should be performed. It is thought that the energy saving could be much bigger in hot and humid climate (e.g. Singapore).
- When the occupant has the responsibility to control his/her microenvironment with a personal ventilation system the energy consumption could be affected significantly. There are not available large enough field measured data of occupants' behaviour (e.g. chosen supply air temperatures, airflow rates, air velocities and directions). Those data should be

collected and used in energy simulations to study the influence on the energy consumption of the delegation of thermal environment control to the occupants.

- In this work, the cooling fan efficiency (*CFE*) has been developed and measured under well defined conditions, based on assumptions of their use in practice. Its advantages have been described. It is now essential to develop a standard measuring method. Test standard conditions should be fixed taking into consideration the influence of air velocity and turbulence field (relative distance between the body and the fan), body area exposed to moving air, body posture, air and mean radiant temperature, air humidity, clothing insulation, metabolic rate, humidity, skin wettedness on the cooling capacity of a fan. Typical fans utilization conditions should be decided based on measured data. A standard procedure for testing the *CFE* index should be developed considering also other factors, such as number of occupants who can benefit from one cooling fan, maximum non-uniformity of body cooling which will be acceptable for the occupants, maximum velocity limitations to avoid blowing of paper, and non-thermal discomfort such as eye blinking, etc.
- For a better prediction of the cooling effect caused by increased air movement, the influence of the latent heat loss should be included, in this way a more correct *CFE* index could be calculated. A sweating manikin should be used to calculate the latent heat loss, but up to now, sweating manikins are expensive and rare.
- In this thesis, the occupation density concept has been used to better calculate occupant's exposure in a room characterized by a non-uniform pollution distribution. The same concept could be applied to better calculated occupant's thermal comfort in a room characterized by a non-uniform thermal conditions distribution.
- Computer fans can be used as cooling fans because their power input is extremely low while the generate flow rate is quite high. More research work is required to evaluate their cooling fan efficiency and their optimal desk locations.

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SPREAD OF KNOWLEDGE

To spread the research results I obtained during the PhD, I participated to the following international conferences (listed in Table 6) and workshops (listed in Table 7).

Table 6. List of the international conferences.

Conference	Function	Date
The 29th International AIVC Conference (Advanced building ventilation and environmental technology for addressing climate change issues), Kyoto, Japan.	Poster presentation (Best Poster)	Oct. 14 -16, 2008
The 11th International Conference on Indoor Air Quality and Climate. Copenhagen, Denmark.	Oral presentation	Aug. 17-22, 2008
The 46th International Conference AICARR-Expocomfort, Milan, Italy.	Oral presentation	Mar. 12-13, 2007
The 10th International Conference on Air Distribution in Rooms, Roomvent 2007. Helsinki, Finland.	Oral presentation	June 13-15, 2007

Table 7. List of the workshops.

Conference	Function	Date
Short Oral Presentation: "Energy saving strategies of personalized ventilation". Forum on "Developments in Personalized Ventilation" Indoor Air 2008. Copenhagen Denmark.	Oral presentation	Aug. 21, 2008
Short Oral Presentation: "Energy saving strategies of personalized ventilation" at 3rd workshop on PECS, EXHAUSTO. Denmark.	Oral presentation	Aug. 15, 2008
Lecture: "Personalized Ventilation" at the summer course: "Integrated design of HVAC systems in buildings". Venice, Italy. 7-11/07/2008.	Oral presentation	Jul. 10, 2008
Lecture: "Energy Saving with Increased Air Movement" at EuroAdemy: "Integrated Analysis of Building Envelope and Indoor Environment". Pamporovo, Bulgaria. 06-11/05/2008.	Oral presentation	May 09, 2008
Oral Presentation: "Saving energy with increased air velocity" at two days workshop DTU-IBP-TU Muchen-Fraunhofer PhD student meeting. Lyngby, ICIEE, DTU, Denmark. 3-4/03/2008.	Oral presentation	March 3, 2008
Oral Presentation: "Saving energy with increased air movement" The workshop was organized by TNO and ICIEE. Lyngby, ICIEE, DTU, Denmark. 9-10/10/2007.	Oral presentation	Oct. 9, 2007
Lecture: "Occupied Density" at EuroAdemy: "Individually Controlled Environment". Pamporovo, Bulgaria. 6-13/05/2007.	Oral presentation	May 11, 2007

ORIGINAL PUBLICATIONS

PAPER A

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Energy saving and improved comfort by increased air movement

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ABSTRACT

In this study, the potential saving of cooling energy by elevated air speed which can offset the impact of increased room air temperature on occupants' comfort, as recommended in the present standards (ASHRAE 55 2004, ISO 7730 2005 and EN 15251 2007), was quantified by means of simulations with EnergyPlus software. Fifty-four cases covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem and Athens), three indoor environment categories I, II and III (according to standard EN 15251 2007) and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated. The required cooling/heating energy was calculated assuming a perfectly efficient HVAC system. A cooling energy saving between 17 and 48% and a reduction of the maximum cooling power in the range 10–28% has been obtained. The results reveal that the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature. Under the assumptions of this study, the energy saving may not be achieved with the methods for air speed increase, such as ceiling, standing, tower and desk fans widely used today when the power consumption of the fan is higher than 20 W.

1. Introduction

According to the 2007 report of the Intergovernmental Panel on Climate Change [1], warming of the climate system is unequivocal. Therefore, a reduction of greenhouse gases emission is needed. The building sector plays an important role in this challenge. The report states that the residential and commercial building sectors have the greatest global potential for emission reduction among all sectors studied in the report. Energy efficiency options for new and existing buildings can reduce CO_2 emissions considerably with net economic benefit. Energy efficient buildings, while limiting the increase of CO_2 emissions, can also improve indoor and outdoor air quality, improve social well-being and enhance energy security [2].

1.1. Air velocity and maximum operative temperature

In the present international indoor climate standards [3–5] the operative temperature comfort limits are based on an air speed limit of 0.20 m/s. However, according to the standards, elevated air speed can offset the indoor temperature rise and provide occupants with thermal comfort. An air speed increase is necessary in order to maintain the heat exchange between the human body and the environment, this being a prerequisite for thermal comfort. The relationship between the air speed and the upper operative temperature limits, as included in the present standards [3,5], is shown in Fig. 1. The recommended speed increase, as shown in Fig. 1, depends not only on the air temperature but also on the difference between mean radiant temperature (t_{mr}) and air temperature (t_a). When the mean radiant temperature is lower than the air temperature, the elevated air speed is less effective for increasing the heat loss from the body. Conversely, elevated air velocity is more effective for increasing the heat loss when the mean radiant temperature is higher than the air temperature. Fig. 1 is based on a theoretical calculation; however, the neutral curve ($t_a = t_{mr}$) has been verified in human subject experiments [6].

The conditions defined in Fig. 1 may be applied only to a lightly clothed person with a clothing insulation between 0.5 and 0.7 clo (0.08–0.1 m^2 K/W) who is engaged in near sedentary physical activity with metabolic rates between 1.0 and 1.3 met (58.15–75.6 W/m^2). The effect of elevated speed on the heat loss from the human body increases at high activity and lighter clothing [3]. Moreover, the increase in operative temperature cannot be higher than 3.0 °C above the values for the comfort zone and the elevated air speed must not be higher than 0.8 m/s. Large individual differences exist between people with regard to the preferred air speed [7]. Therefore the standards require personal control over the speed, the benefit of which was also confirmed in [8]. Thus it

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Nomenclature

COP	coefficient of performance of the chiller
$E_{N,C}^{v=i}$	energy need for cooling ($E_{N,C}$) obtained when the air velocity is $i \leq 0.2$ or $i = 0.5$ or $i = 0.8$ m/s (kWh/(m^2 y))
$E_{N,C}$	energy need for cooling (kWh/(m^2 y))
$E_{el,Cool}$	electrical energy consumed by the chiller (kWh/(m^2 y))
$E_{el,Fan}$	electrical energy consumed by the fan (kWh/(m^2 y))
$E_{el,Net}$	net electrical energy saved (kWh/(m^2 y))
h_i	annual number of hours that the fan is operating for increasing the air velocity. It is calculated for an air velocity of 0.5 m/s ($h_{0.5}$) and 0.8 m/s ($h_{0.8}$) (h)
h_{tot}	the total occupant working hours (h)
P_{Fan}	the electrical input power of the fan (W)
t_a	air temperature (°C)
t_{mr}	mean radiant temperature (°C)
t_{op}	operative temperature (°C)

Greek symbol

η	energy losses from emission, distribution and storage for cooling. It is the ratio between the energy need for cooling and the thermal energy that the chiller has to produce

may not be appropriate to offset a temperature increase by increasing the air speed within a centrally controlled air system [8].

The possibility of increasing the upper operative temperature limit may reduce the energy consumption without significantly affecting occupants' thermal comfort. The individual control of air movement can be achieved with personalized ventilation systems, task/ambient systems, desk, standing, tower or ceiling fans, and under some conditions with operable windows. The energy consumption for air movement generation by these methods is different. The purpose of this study is to quantify, by means of simulations with EnergyPlus software, the potential savings of energy need for cooling (defined in EN 15615 [9]) achieved by elevated air speed without reducing occupants' thermal comfort conditions.

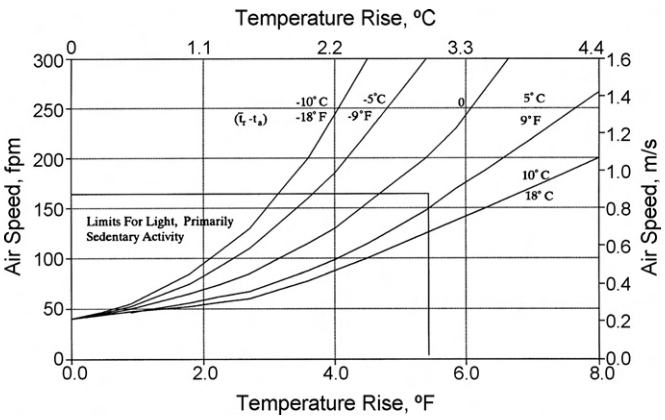

Fig. 1. Air speed required to offset increased temperature (Fig. 5.2.3 from ASHRAE [3]).

2. Methods

The European standard 15265 [10] recommends a format for reporting the input data of an energy simulation. The following presentation of input data complies with the guidance in the standards.

2.1. Building locations and weather data

The energy simulations were performed for the same single office sited in six European and Mediterranean cities listed in Table 1. The cities were chosen in order to describe in a homogeneous way different climate conditions. The focus was on summer conditions. The Cooling Degree Days [11] with a base temperature of 18 °C were used as an indicator of the intensity of the summer period. The ASHRAE IWEC Weather Files were used as input data in the simulation model.

2.2. Description of the office room

The single office room has a floor surface area of 4 m by 2.5 m. The room height is 3 m. The external walls are constructed with 20 mm of plaster, 100 mm of glasswool, 240 mm of brick and 10 mm of internal plaster. The window has an external low-emissivity glass pane (thickness 6 mm), 13 mm of air and an internal glass pane (thickness 6 mm). It has a *U*-value equal to 1.72 W/(K m^2) and a g-factor or Solar Heat Gain Coefficient equal to 0.56. The window has a total area of 2.4 m^2 (24% of the floor area, height of 1.2 m and width of 2 m). The window faces south. There is an external shading device. It has a shading coefficient of 0.48 (g-factor equal to 0.43), and it is activated when the total irradiance on the windows is higher than 400 W/m^2. The internal walls, floor and ceiling are adiabatic. The effect of thermal mass is taken into account.

2.3. Internal temperature, ventilation and infiltration rate

The thermal comfort conditions and ventilation specifications were chosen in order to guarantee the values defined in EN 15251 [4] for the categories I, II and III for indoor environment in the room during occupation. From 7:00 till 18:00 the heating and cooling system kept the internal operative temperature within a range between the minimum operative temperature below which heating is required (Min t_{op} for heating) and the maximum operative temperature above which cooling is required (Max t_{op} for cooling). The minimum and maximum operative temperatures are shown in Table 2. During weekends and night-time the temperature set-back was 12 °C in winter and 40 °C in summer. The design ventilation rates are shown in Table 2. The design airflow rate was supplied during occupation hours. The airflow rates during unoccupied periods were 7% of the design values, i.e. from 0.06 to 0.14 l/s m^2 (the standard suggests a minimum airflow rate for

Table 1
Cities where the office is sited

City	Country	Latitude	Cooling degree day $-t_{base}18$ °C
Helsinki	Finland	60°19′	33
Berlin	Germany	52°28′	170
Bordeaux	France	44°49′	263
Rome	Italy	41°47′	508
Jerusalem	Israel	31°46′	647
Athens	Greece	37°54′	1076

The intensity of the summer period is described using the Cooling degree days with a base temperature of 18 °C.

Table 2
Simulated cases: category of indoor environment, airflow rates, minimum and maximum operative temperatures

Category according EN 15251 2007	Airflow per person (l/(s person))	Airflow per floor area[a] (l/(s person))	Min t_{op} for heating (°C)	Velocity (m/s)	Temperature increase (K)	Max t_{op} for cooling (°C)
I	10	1	21	<0.2	0	25.5
				0.5	1.7	27.2
				0.8	2.5	28
II	7	0.7	20	<0.2	0	26
				0.5	1.7	27.7
				0.8	2.5	28.5
III	4	0.4	19	<0.2	0	27
				0.5	1.7	28.7
				0.8	2.5	29.5

The maximum operative temperatures for cooling are increased according to the air velocity.
[a] Recommended values from Annex C of EN 15251 [4] for low polluting buildings.

unoccupied hours in the range 0.1–0.2 l/s m^2). The infiltration is considered null.

2.4. Internal heat gains, occupancy and description of the HVAC system

One occupant was present in the room (10 m^2 per person). She/he contributed to both sensible and latent heat loads. The activity level of the occupant was 1.2 met (1 met = 58.15 W/m^2), and the total heat produced per occupant was thus around 125 W. The balance between sensible and latent heats was calculated by the software used. The occupant was present in the room from Monday to Friday, from 9:00 to 18:00 with an hour as break at noon. Saturday and Sunday were free days and no public holidays were involved. The heat load due to office equipment was 5.4 W/m^2. According to ASHRAE [11], this value corresponds to a "light load office". The loads follow the schedules of the occupant. The lighting load was 6 W/m^2, a common value used in practice for an office. The lighting load was at 90% of its capacity from 9:00 to 10:59, at 70% from 11:00 to 12:59 and from 14:00 to 15:59, and at 100% from 16:00 to 17:59. In the other hours the light was switched off. The energy needed was calculated assuming a perfectly efficient HVAC system. The airflow network and the heating and cooling plants were not modelled; therefore the airflow needed was supplied at outdoor conditions. The humidity level was monitored but not controlled.

2.5. Simulated cases

From Fig. 1, assuming that the air temperature is equal to mean radiant temperature ($t_a = t_{mr}$), it is shown that the increase allowed in operative temperature is equal to 1.7 °C for an airflow of 0.5 m/s and 2.5 °C for an airflow of 0.8 m/s. These values were added to the maximum summer operative temperatures for the three categories as specified in EN 15251 [4]. The values shown in Fig. 1 were obtained for a comfort limit of 26 °C, which is the comfortable temperature limit for category II in EN 15251 [4]. It is reasonable to assume that the same increments in operative temperature can be applied for the comfortable temperature limits for categories I and III, i.e. 25.5 and 27 °C. In total, 54 cases, covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem and Athens), three indoor environment categories (I, II and III) and three air velocities (<0.2, 0.5 and 0.8 m/s) as listed in Table 2, were simulated. The summer design day simulation was performed for 54 cases in order to calculate the maximum power needed for providing the comfort conditions. The maximum power is used to size the chiller. The summer design day conditions were taken from ASHRAE [11]. The cooling design days used in the simulation were characterized by an annual percentile of 1.0% for the dry-bulb temperatures and the mean coincident wet-bulb temperatures. These are suggested for use by ASHRAE [11] when sizing cooling equipment such as chillers or air-conditioning units.

2.6. Simulation software

A robust building energy simulation program, EnergyPlus, was used for the simulations. This software allows for performing simulations of the building and the HVAC system as a whole. It calculates the thermal loads to be satisfied and defines the system strategy needed to fulfil the required comfort conditions. In the present research, EnergyPlus is used mainly in order to predict the energy need for keeping the room operative temperature within the comfort limits (specified in Table 2).

3. Results

The energy need for cooling ($E_{N,C}$) [9] of the room when located in each of the selected six cities for the three categories (Table 2) at the three levels of velocity (0.2, 0.5 and 0.8 m/s) and the corresponding operative temperatures (Table 2) is listed in Table 3. The energy need for cooling is the annual amount of cooling energy that must be supplied to the room to keep the operative temperature below the maximum summer operative temperature limit. The cooling energy for the control of humidity and the energy losses in the system are not included.

The heating energy need is not affected by the air velocity increase. It depends on the outdoor conditions (climate zone) and the required category of the indoor environment. The maximum heating energy need is in Helsinki for category I (83 kWh/m^2 y). In Rome, Jerusalem and Athens the heating demand is covered by the internal heat load, and there is therefore no need for a heating system.

The fan operation total hours (h_i) are shown in Table 2 as well. It is assumed that when the indoor operative temperature is higher than the maximum operative temperature limit (without any increase of the air velocity) the occupant switches on the fan. Thus the fan operation hours were calculated as the sum of hours during which the operative temperature was higher than the maximum operative temperature limit and the occupant was in the room, e.g. an hour is counted if the occupant was in the room and the room operative temperature was above 25.5 °C for category I, or above 26 °C for category II, or above 27 °C for category III. The total number of hours that the fan is in operation is proportional to the energy consumption of the fan. In Table 3 the ratio between the fan operation hours and the total yearly occupant working hours is reported. The total occupant working hours (h_{tot}) per year (260 working days) is 2080.

Table 3
Energy need for cooling ($E_{N,C}$) per unit of floor and fan operating hours at the three velocity levels for the three categories of indoor environment when the room is located in the six cities with different outdoor climate conditions

City	C[a]	Velocity								
		<0.2 m/s	0.5 m/s				0.8 m/s			
		Reference case	Energy		Fan		Energy		Fan	
		$E_{N,C}$[b]	$E_{N,C}$[b]	Saved[c] (%)	$h_{0.5}$[d]	$h_{0.5}/h_{tot}$[e] (%)	$E_{N,C}$[b]	Saved[c] (%)	$h_{0.8}$[d]	$h_{0.8}/h_{tot}$[e] (%)
Helsinki	I	18	12	34	636	31	9	48	645	31
	II	21	15	29	765	37	12	41	788	38
	III	24	18	24	859	41	16	35	867	42
Berlin	I	24	16	32	814	31	13	45	826	31
	II	26	19	28	848	37	16	40	864	38
	III	27	21	23	907	41	18	34	916	42
Bordeaux	I	39	28	27	1080	52	24	38	1091	52
	II	41	31	24	1184	57	27	34	1204	58
	III	42	33	21	1345	65	29	31	1368	66
Rome	I	52	40	23	1300	63	35	33	1308	63
	II	53	42	21	1406	68	37	30	1420	68
	III	53	43	19	1499	72	38	27	1509	73
Jerusalem	I	65	51	21	1483	71	45	30	1491	72
	II	66	52	20	1722	83	47	29	1746	84
	III	66	54	19	1909	92	48	27	1928	93
Athens	I	75	61	18	1419	68	56	25	1439	69
	II	74	61	17	1555	75	56	25	1579	76
	III	73	61	17	1888	91	55	24	1921	92

The energy saved due to the increase of air velocity (or relative increase of upper operative temperature limits) is listed.
[a] C = category according EN 15251 [4].
[b] $E_{N,C}$ = energy need for cooling (kWh/(m^2 y)).
[c] Saved = percentage of the saved energy need for cooling compared to the reference case.
[d] h_i = annual number of hours that the fan is operating for increasing the air velocity.
[e] h_i/h_{tot} = annual number of hours that the fan is operating (h_i) over yearly occupant working hours (h_{tot}).

The maximum cooling power per unit of floor area and the percentage of time that the relative humidity requirements are fulfilled when the occupant is in the room are shown in Table 4.

4. Discussion

In all simulated cases, increasing the air velocity implied a reduction of the energy consumption (Table 3). A saving of the

Table 4
Maximum cooling powers per square metre and percentage of time that the relative humidity requirements are fulfilled at the three velocity levels for the three categories of indoor environment when the room is located in the six cities with different outdoor climate conditions

City	C[a]	Velocity							
		<0.2 m/s (reference case)		0.5 m/s			0.8 m/s		
		Max power[b]	RH percentage[c]	Max power[b]	Saved[d] (%)	RH percentage[c]	Max power[b]	Saved[d] (%)	RH percentage[c]
Helsinki	I	49	65	42	15	65	38	22	65
	II	48	94	41	13	93	39	19	92
	III	45	100	41	10	100	38	15	100
Berlin	I	55	82	47	14	83	44	20	82
	II	51	98	45	11	98	42	17	98
	III	46	100	42	10	100	39	15	100
Bordeaux	I	60	71	52	13	77	49	18	78
	II	54	95	48	10	95	45	16	95
	III	47	100	43	9	100	41	14	100
Rome	I	60	57	53	12	67	50	18	72
	II	55	96	49	10	96	46	16	96
	III	48	100	44	10	100	41	14	100
Jerusalem	I	56	90	49	13	88	46	18	85
	II	50	99	45	11	98	42	16	97
	III	44	100	40	10	100	38	15	100
Athens	I	73	74	66	10	80	63	14	81
	II	65	97	59	9	96	56	13	96
	III	56	100	51	8	100	49	12	100

[a] C = category according EN 15251 [4].
[b] Max power = maximum cooling power (W/m^2).
[c] RH percentage = percentage of time that the relative humidity requirements are fulfilled when the occupant are in the room.
[d] Saved = percentage by which the maximum cooling power is reduced compared to the reference case.

energy need for cooling between 17 and 48% is obtained. The highest percentage of energy saving was obtained in Helsinki for category I of the indoor environment. The lowest percentage of energy saving was obtained in Athens for category III of the indoor environment. The percentage of savings decreases when the quality of the indoor environment category decreases, e.g. in Bordeaux for category I the saving was 27% and for category III it was 21%. The percentage of savings decreases with the increase of the cooling degree days (defined in Section 2.1). The percentage of savings increases when the air velocity increases. In fact, the higher savings have been obtained for an air velocity equal to 0.8 m/s. These conclusion can be drawn from Fig. 2. In summary, increasing the air velocity to compensate for the higher room temperature is an energy-saving solution that gives a higher performance in high quality indoor environment offices located in a cold climate. It is interesting to note that, in Helsinki, Berlin and Bordeaux, the energy need for cooling increased with the reduction of the quality of the indoor environment due to the free cooling effect of the outdoor air.

The fan operation hours are listed in Table 3. The fan operation hours increase with an increase in the number of cooling degree days (defined in Section 2.1) and with a reduction of the indoor environment category. The fan operation hours are almost independent of the increase of air velocity. In Table 3 the ratio between the fan operating hours and the yearly occupant working hours is shown. The ratio varies between 31 and 93%. High values of the ratio mean that the fan would work also during winter-time, when it is presumed that people dress with a clothing insulation equal to 1 clo. In this case the graph, as shown in Fig. 1, cannot be applied. However, the fan is working during winter-time in warm climates (Jerusalem and Athens), where the occupant would probably have lighter clothing. Moreover, during winter-time, it is reasonable to think that other techniques would be used to cool the room, such as night free-cooling, or increasing the shading capacity or the thermal mass of the building.

The relative humidity in the environment was not controlled by the system but it was monitored. From Table 4 it can be seen that for all simulated cases with category III the requirements for indoor relative humidity (20% < RH < 70%) were always fulfilled. For cases with category II the requirements (25% < RH < 60%) were fulfilled from 92 to 99% of the time, depending on the outdoor conditions and the relative increase of air velocity. For the cases with category I the requirements (30% < RH < 50%) were fulfilled from 57 to 90% of the time, the rest of the time the humidity conditions were mostly within the range 25% < RH < 60%. A humidification and dehumidification system would be needed to keep the relative humidity always in accordance with the requirements in the standards for category I and II.

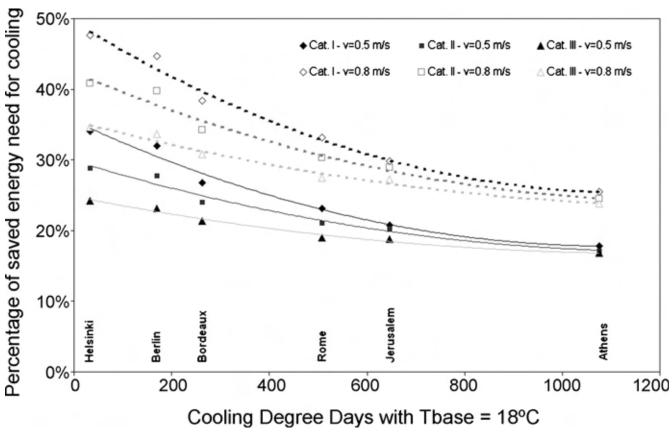

Fig. 2. Percentage of saved energy need for cooling vs cooling degree days. The points are the values obtained from the simulations. The lines are second order polynomial interpolations of the calculated data. The reference case for each category and city is the one without any increase in air velocity (<0.2 m/s).

The maximum cooling power per unit of floor area is shown in Table 4. The reduction of the maximum cooling power due to the increase of air movement is in the range 8–22%. It is higher for an air velocity equal to 0.8 m/s, for the cold climates and for higher quality level of indoor environment. The most effective parameter is the level of air velocity. As a consequence, smaller chillers may be installed, which will lead to a reduction of the initial (investment) costs.

4.1. *Energy consumption of the fan*

The air movement increase can be produced by ceiling fans (common nameplate power consumptions around 70 W), standing fans (50 W), tower fans (40 W), desk fans (30 W), personal ventilation systems and under certain conditions with operable windows. Measurements of several fans, performed in this study, confirm that the effective input fan power is equal to the value stated on its nameplate.

In order to check whether the electrical consumption of the fan is a critical factor for energy saving, the difference between the saved (in the chiller) and consumed (by the fan) energy is calculated. The saved electrical energy for running the chiller is named $E_{\mathrm{el,Cool}}$ and the electrical energy consumed by the fan is named $E_{\mathrm{el,Fan}}$. The difference between $E_{\mathrm{el,Cool}}$ and $E_{\mathrm{el,Fan}}$ is hereafter named net electrical energy saved ($E_{\mathrm{el,Net}}$). The saved electrical energy for running the chiller ($E_{\mathrm{el,Cool}}$) depends on the saved energy need for cooling (see $E_{\mathrm{N,C}}$ in Table 3), on the energy losses from emission, distribution and storage (taken into consideration in the calculations by η) and on the coefficient of performance (COP) of the chiller. COP and η depend on the type of cooling system used and on the building characteristics. The electrical energy consumed by the fan ($E_{\mathrm{el,Fan}}$) depends on the electrical input power of the fan (P_{Fan}) and on the number of fan operating hours (h_i). The net electrical energy saved ($E_{\mathrm{el,Net}}$) is defined by Eq. (1).

$$E_{\mathrm{el,Net}} = E_{\mathrm{el,Cool}} - E_{\mathrm{el,Fan}} = \frac{(E_{\mathrm{N,C}}^{v\le 0.2\,\mathrm{m/s}} - E_{\mathrm{N,C}}^{v=i})(1+\eta)}{\mathrm{COP}} - 10^{-4} P_{\mathrm{Fan}} h_i \quad (i = 0.5\ \mathrm{or}\ 0.8\ \mathrm{m/s}) \tag{1}$$

where $E_{\mathrm{el,Net}}$ is the net electrical energy saved (kWh/(m^2 y)); $E_{\mathrm{N,C}}^{v=i}$ is the energy need for cooling ($E_{\mathrm{N,C}}$) obtained when the air velocity is $i \le 0.2$ or $i = 0.5$ or $i = 0.8$ m/s (kWh/(m^2 y)); P_{Fan} is the electrical input power of the fan (W); h_i is the number of hours that the fan is operating (Table 3) (h); η is the ratio between the energy need for cooling and the thermal energy that the chiller has to produce; COP is the coefficient of performance of the chiller.

Practical experience shows that the COP can vary within the range between 2.5 and 4.5 with a best guess value of 3.5 and the η can vary within the range between 0 and 0.15 with a best guess value of 0.05. The influence of these two parameters on the net electrical energy saved, $E_{\mathrm{el,Net}}$, was calculated for Helsinki in the case of the indoor environment category I for velocity elevated to 0.5 and 0.8 m/s. From the results shown in Fig. 3 it can be seen that $E_{\mathrm{el,Net}}$ varies as a function of the COP and η for the two air velocities.

The results in Fig. 3 reveal that COP has a significant influence on the net electrical energy saved, and η has less impact. Moreover, it can be seen that $E_{\mathrm{el,Net}}$ is lower for higher values of COP, is due to the fact that the required electrical energy for producing a certain amount of cooling energy decreases with the increase of the COP.

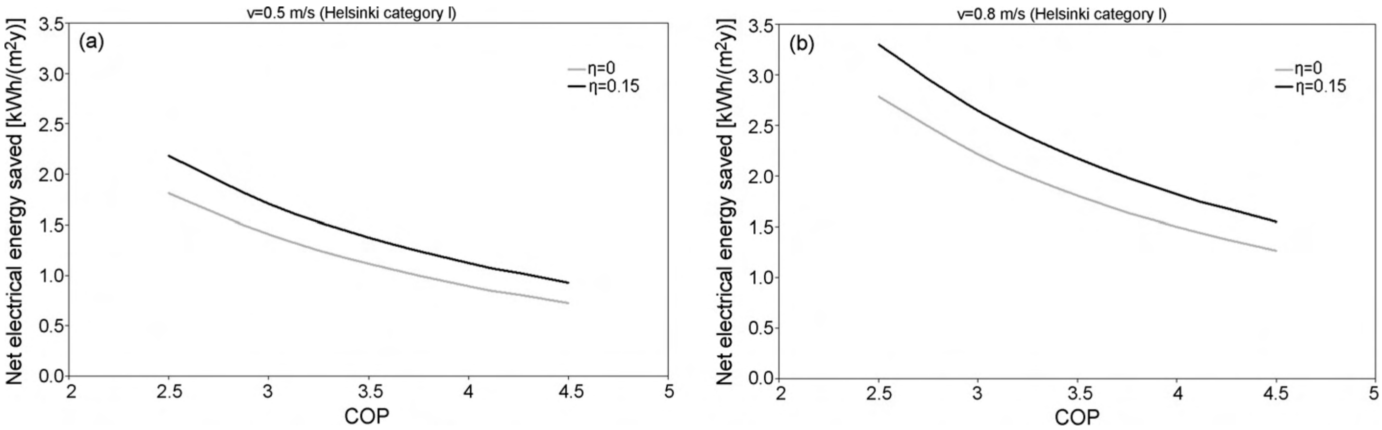

Fig. 3. The net electrical energy saved ($E_{el,Net}$) calculated for Helsinki for category I vs the COP for η equal to 0 or 0.15 for air velocity of 0.5 m/s (a) and 0.8 m/s (b).

Easy-to-use graphs for checking, as a rule of thumb, how much energy can be saved as a function of the fan input power are shown in Fig. 4. Four cases are reported, including two air velocities (0.5 and 0.8 m/s) and two combinations of COP and η. The combinations of COP and η were chosen in order to calculate the extreme cases. With COP = 2.5 and η = 0.15 the $E_{el,Net}$ is the highest, while with COP = 4 and η = 0 the $E_{el,Net}$ is the lowest. The net electrical energy saved ($E_{el,Net}$) was calculated for a fan input power within the range 2–70 W for all the 54 simulated cases. The maximum and minimum values for each fan input power has been plotted. The use of these graphs is explained in the following example. If the input power of the fan is 20 W, the COP is equal to 2.5, η = 0.15 and the air velocity is 0.8 m/s (Fig. 4a), the expected net electrical energy saved is then at minimum 2.1 kWh/(m^2 y) and at maximum 5.9 kWh/(m^2 y). On the other hand, with the same fan input power, if the COP is equal to 4, η = 0 and the air velocity remains the same (Fig. 4b), the expected net electrical energy saved is then at minimum 0.4 kWh/(m^2 y) and at maximum 1.9 kWh/(m^2 y). If the input power of the fan is still 20 W, the COP is equal to 4, η=0 and the air velocity is 0.5 m/s (Fig. 4d), the expected net electrical energy saved is then at maximum 0.5 kWh/(m^2 y). In this case, the minimum is not plotted because there is no energy saving but energy waste. The values plotted in Fig. 4 were obtained from computer simulations where the human behaviour was not

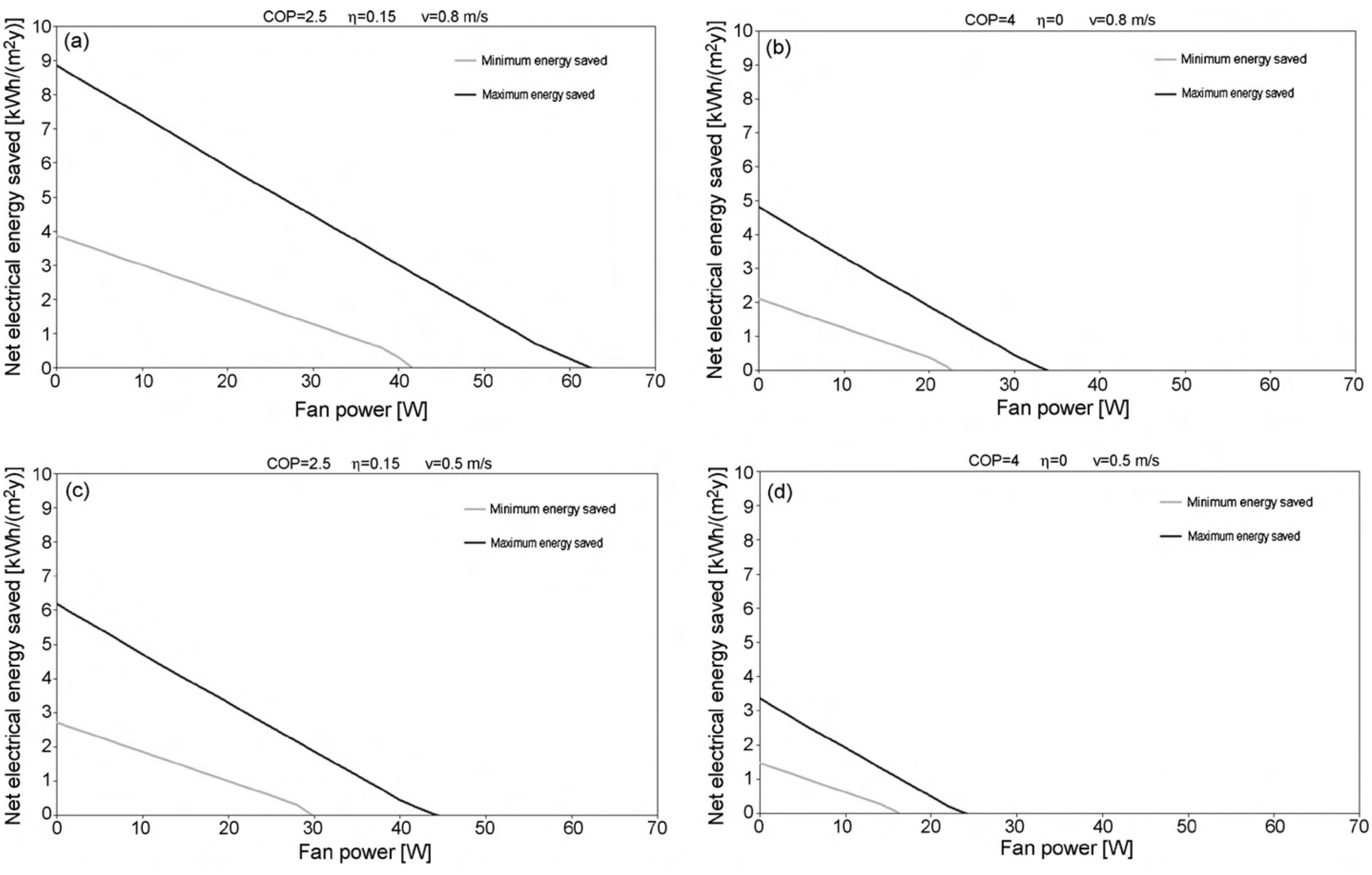

Fig. 4. The net electrical energy saved vs fan input power when: (a) COP = 2.5, η = 0.15 and air velocity = 0.8 m/s; (b) COP = 4, η = 0 and air velocity = 0.8 m/s; (c) COP = 2.5, η = 0.15 and air velocity = 0.5 m/s; and (d) COP = 4, η = 0 and air velocity = 0.5 m/s.

modelled. The human behaviour (e.g. leaving the fan switched on when the occupant is out of the office) affects the possibility of saving energy by using the technological solution studied in this paper. The main advantage of the presentations in Fig. 4 is that the graphs are independent of the location and of the indoor environment category and can therefore give a first estimation of the saving. For example, if the fan power input is 60 W, then it can be easily seen that energy savings cannot be achieved. From the figures, it can be concluded that traditional systems, such as ceiling fans (70 W) and standing fans (50 W), cannot be used to save energy on the basis of assumptions made in this study. From Fig. 4 it can be seen that for the conditions considered in this study (outdoor climate, indoor environment category, air velocity increase) and for the range of COP and η used, it is never possible to reach a net energy saving with a fan input power higher than 60 W. On the other hand, it is always possible to save energy if the input power is lower than 15 W. Calculations made for the best guess values for COP and η, respectively 3.5 and 0.05, reveal that energy savings will not be achieved with fans using more than 20 W. This can be done using a small desk fan or a personal ventilation system. The main conclusion is that the fan input power is a critical factor for the applicability of this solution in practice.

The results in Fig. 1 were obtained and verified with an airflow over the whole body [6] while personal ventilation systems or desk fans typically provide cooling only to the upper part of the body. Nevertheless, the authors believe that the difference would not be significant, because most of the heat loss occurs in the upper part of the body (the head is a strong dissipater of heat). Another advantage of the personal ventilation system is that it will increase the inhaled air quality and this will improve occupants' health and productivity [12].

4.2. Limitations of the study

The HVAC system was not modelled; therefore the interaction between the building and the system could not be predicted. The moisture control was not modelled either. These simplifications may change the range of saved energy need for cooling. Sensitivity analyses for internal and external heat loads and behaviour of the occupant have not been performed.

5. Conclusions

The main conclusions of this study are:

- Cooling energy savings in the range of 17–48% have been obtained in the case of increased room temperature and elevated velocity. The percentage of savings increases when: the air velocity increases, the indoor environment category level increases, and the number of cooling degree days decreases.
- The required power input of the fan is a critical factor. Traditional systems, such as ceiling, standing, tower and desk fans may not be applied to save energy under the assumptions made in this study.

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PAPER B

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Energy-saving strategies with personalized ventilation in cold climates

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ABSTRACT

In this study the influence of the personalized supply air temperature control strategy on energy consumption and the energy-saving potentials of a personalized ventilation system have been investigated by means of simulations with IDA-ICE software. GenOpt software was used to determine the optimal supply air temperature. The simulated office room was located in a cold climate. The results reveal that the supply air temperature control strategy has a marked influence on energy consumption. The energy consumption with personalized ventilation may increase substantially (in the range: 61-268%) compared to mixing ventilation alone if energy-saving strategies are not applied. The results show that the best supply air temperature control strategy is to provide air constantly at 20°C. The most effective way of saving energy with personalized ventilation is to extend the upper room operative temperature limit (saving up to 60% compared to the reference case). However, this energy-saving strategy can be recommended only in a working environment where the occupants spend most of their time at their workstation. Reducing the airflow rate does not always imply a reduction of energy consumption. Supplying the personalized air only when the occupant is at the desk is not an effective energy-saving strategy.

KEYWORDS

Energy analysis; Energy saving; Personalized ventilation; Supply air temperature control; Ventilation; Personal Environmental Control System.

INTRODUCTION

Personalized Ventilation (PV) aims to supply clean and cool air at low velocity and turbulence direct to the breathing zone of occupants. Each occupant may be provided with control of the supplied flow rate and/or supplied air temperature. Control of the airflow direction may be available as well. Thus, beside its ability to decrease the level of pollution in inhaled air and the risk of infection transmission [1, 2], PV improves occupants' thermal comfort. PV may thus increase occupants' satisfaction, decrease Sick Building Syndrome (SBS) symptoms and sick leave, and increase work performance [3].

Little is known about energy use of personalized ventilation. Seem and Braun [4] studied the energy use characteristic of a system incorporating personal environmental control compared with convectional designs through the use of computer simulations. They simulated the desktop personal environmental control system described by Arens et al. [5]. The system incorporated an electrical radiant panel, two local air distribution fans, a noise generator, a local task lighting and a workstation occupancy sensor. Their study showed that the effect of personal environmental control ranged between a 7% saving and 15% penalty in building lighting and HVAC electrical use. Bauman et al. [6] measured the field performance of the same system described above. They reported that the energy consumption of a personal environmental control system follows the occupancy behaviour; the system switches off when occupants leave the workstation, thus allowing energy saving to be measured.

Energy-saving potential

In the literature, information is available about the energy-saving potential of personalized ventilation. The main strategies suggested in the literature to have potential for energy-saving with personalized ventilation are:

- Reducing the outdoor airflow rate due to the higher ventilation effectiveness of PV [7, 8, 9, 10, 11, 12].
- Expanding the room temperature comfort limits by taking advantage of PV's ability to create a controlled microenvironment [6, 10, 11].
- Supplying the personalized air only when the occupant is present at the desk [4, 6].

There are several definitions of ventilation effectiveness [13]; in this paper the ventilation effectiveness is defined as the ratio of the concentration of pollution in exhaust air divided by the concentration of pollution in air inhaled by occupants. According to the European standard EN 13779 [14] and report CR 1752 [15], the minimum airflow rate can be reduced by using the ventilation effectiveness (divided by the ventilation effectiveness). The ASHRAE standard 129 [16] defines the Air Change Effectiveness (ACE) as the ratio of the age of the exhaust air and the age of the air in the breathing zone. For the ASHRAE standard 62.1 [17] the minimum outdoor air supply rate could be decreased using ACE (multiplied by 1/ACE). Several studies reported high ventilation effectiveness or ACE associated with personalized ventilation. Faulkner et al. [9] studied in chamber experiments the ACE of a task ventilation system with an air supply nozzle located underneath the front edge of a desk. The personalized airflow rate per person (q_V) varied from 3.5 to 6.5 l/s. They reported that the system studied had an ACE equal to 1.5; therefore the minimum outdoor air supply rate could be decreased by one third. Sekhar et al. [11] found that in a tropical climate, for an ambient temperature of 26°C, and a PV flow rate of 7 l/s per person at a supply air temperature of 23°C or 20°C, the ventilation effectiveness was 1.42. Melikov et al. [7] studied in chamber experiments the influence of five different air terminal devices on the ventilation effectiveness with the airflow rate varying from 5 l/s up to 23 l/s. The ventilation effectiveness varied within the range 1.30- 2.38. A highly efficient air terminal device providing almost 100% clean and cool personalized air in each inhalation has been developed by Bolashikov et al. [8]. The air terminal device makes it possible to increase the ventilation effectiveness 20 times or more compared with mixing ventilation. Niu et al. [12] studied the ventilation performance of a chair-based personalized ventilation system. By comparing eight different air terminal devices it was found that up to 80% of the inhaled air could be composed of fresh personalized air (ventilation effectiveness equal to 5) with a supply flow rate of less than 3.0 l/s. Nielsen et al. [2] proposed a chair with integrated personalized ventilation discharging supply air at very low velocities and relying on the entrainment of this clean PV air from the natural convection flow around the human body. They found that more than 70-80% of the inhaled air is personalized air (ventilation effectiveness > 3.5-5) with an airflow rate in most cases equal to10 l/s.

Bauman et al. [6] reported that at a high room air temperature (25°C - 27°C), the local cooling effect of the desktop system was able to maintain average temperatures in the occupied zone of one workstation from 0.5°C to 1.5°C below the corresponding temperatures in an adjacent workstation without a desktop system. Kaczmarczyk et al. [18] in an experiment comprising 60 human subjects showed that at a room temperature of 26°C PV, supplying air at 20°C was able to keep occupants in better thermal comfort (close to neutrality instead of slightly warm) than a mixing ventilation system. Sekhar et al. [11] showed that human subjects prefer, from a thermal comfort and perceived indoor air quality point of view, an environment with a room temperature of 26°C and PV at 23°C or 20°C rather than a room at 23°C without a PV system. They stated that for a tropical climate, where the common indoor temperature for a conditioned building is 23°C, a significant reduction of energy consumption can be achieved if the room temperature is maintained at 26°C. Even if the air is supplied isothermally the personalized air is able to cool the occupant. According to the present international indoor climate standards [19, 20, 21], elevated air speed can offset the indoor temperature rise and provide occupants with thermal comfort. A relationship between the air speed and the upper operative temperature limits can be found in the above mentioned standards. The relationship is based on a theoretical calculation; however, it has been verified in human subject experiments [22]. Individual differences exist between people with regard to the preferred air speed [22, 23]. Therefore, the standards require personal control over the speed.

Depending on their activities during working time occupants may spend only a part of the time in the office and even a shorter time at the desk [18, 24, 25, 26, 27, 28, 29, 30, 31, 32]; therefore energy-saving may be achieved if the system is able to automatically switch off when occupants are not at the desk.

The purpose of this study is to analyse, by means of simulations with IDA-ICE software, the influence of the personalized supply air temperature control strategy on energy consumption and the energy-saving potentials of a personalized ventilation system in a cold climate.

METHODS

The European standard 15265-2006 [33] recommends a format for reporting the input data of an energy simulation. The following presentation of input data complies with the guidance in the standards.

Building locations and weather data

An office in a building located in Copenhagen (Denmark) was simulated. The weather is characterized by a cold climate. The ASHRAE IWEC Weather File for Copenhagen is used as input data in the simulation model.

Description of the office room

The open-space office has a floor surface area of 6 x 20 m. The room height is 3 m. The external walls are constructed with 20 mm of plaster (thermal conductivity, λ=0.6 $WK^{-1}m^{-1}$), 150 mm of glasswool (λ=0.036 $WK^{-1}m^{-1}$), 240 mm of clay brick (λ=0.57 $WK^{-1}m^{-1}$) and 10 mm of internal plaster; the overall U-value of the external wall is 0.2 $WK^{-1}m^{-2}$. The window is composed of an external glass pane (thickness 6 mm), 15 mm of argon (90%) and an internal low-emissivity glass pane (thickness 6 mm). It has an overall U-value of 1.2 $WK^{-1}m^{-2}$, a g-factor or Solar Heat Gain Coefficient equal to 0.61, and a light transmittance equal to 0.77. The window has a total area of 36 m^2 (20% of the floor area, height = 1.8 m and width = 20 m). The window faces south. There is a shading device composed of blinds between the window panes. It has a multiplier for a total shading coefficient equal to 0.39. It is activated when the incident light on the windows is higher than 200 W/m^2. The internal walls, floor and ceiling are adiabatic. The effect of thermal mass is taken into account.

Internal temperature, ventilation and infiltration rate

The thermal comfort conditions and ventilation specifications were chosen in order to comply with the values defined in EN 15251 [19] for the category I of the indoor environment in the room during occupation. From 6:00 till 17:00 the heating and cooling systems kept the indoor operative temperature within a range between 21°C (lower room operative temperature limit) and 25.5°C (upper room operative temperature limit). During weekends and night-time the temperature set-back was 12°C in winter and 40°C in summer. The upper room operative temperature limit, θ_{UP}, was expanded in the cases shown in Table 1 for studying the influence of this strategy on the energy need. The design airflow rate was supplied during occupation hours. The airflow rate is calculated according to the European standard EN 15251 [19]. The total airflow rate, q_V, is the sum of the required ventilation rate per person and per floor area. EN 15251 [19] recommends 10 l/s person as ventilation rate per person for the indoor environment category I and 1 $l/(sm^2)$ when the building is considered to be low- polluting. The floor area per occupant is 10 m^2. Therefore, the total airflow rate is equal to 20 l/s per person during occupation hours. The total airflow rate is more than double that required in the ASHRAE standard 62.1 [17]. The European standard requires a higher ventilation rate than the ASHRAE standard. At full occupancy, 12 occupants were present in the room (10 m^2 per person); thus the total outdoor airflow rate is 240 l/s. The airflow rate was reduced in the cases shown in Table 1 (cases 9-14) in order to study the influence of this strategy on the energy need. From Case 23 and Case 26 the occupancy varied according to Figure 1 and Figure 2. The standard EN 15251[19] suggests supplying a minimum value of 0.1 to 0.2 $l/(sm^2)$ during unoccupied hours. This part is not covered by the ventilation system but by the infiltration. The Equivalent Leakage Area [34] is equal to 0.0093 m^2 (0.2 $l/(sm^2)$ when the pressure difference is 4 Pa).

Internal heat gains, occupancy and description of the HVAC system

The 12 occupants contribute to both sensible and latent heat load in the room. The activity level of the occupants was 1.2 met (1 met = 58.15 W/m^2), and the total heat produced per occupant was thus around 125 W. The balance between sensible and latent heat loads is calculated by the software. The occupants were present in the room from Monday to Friday, from 8:00 to 17:00 with a break of one at noon. Saturday and Sunday were free days and no public holidays were involved. The heat load due to office equipment was 6 W/m^2. According to ASHRAE [35], this value corresponds to a “light load office”. The loads follow the schedules of the occupants. The lighting load was 10 W/m^2 during working hours (8:00-17:00). Outside these hours the light was switched off.

In practice, it will be difficult to use the personalized ventilation alone to condition an entire room if the PV system supplies only outdoor air. For comfort reasons there are limitations for the maximum airflow rate and the temperature of the supplied personalized air. Therefore it is not possible to adapt the flow rate (as in the variable air volume system) or the supply temperature (as in the constant air volume system) of the personalized air to the levels needed for heating or cooling of the whole room. In this study two independent systems were modelled. Four-pipe fan coil units were used to control the operative room temperature. The required outdoor airflow rate was conditioned in the design conditions by an AHU with a heat recovery exchanger (efficiency of 0.7). The humidity was not controlled during the simulations since this is not common practice in Denmark. A free-cooling strategy during night-time (from 18:00-6:00) from 1 May to 30

September was used. The supplied airflow was 3 l/(sm^2). The free-cooling starts when the outdoor air temperature is at least 5°C cooler then indoor air and the indoor air temperature is at least 25°C. It stops if the indoor air temperature is lower than 21°C or the difference between indoor and outdoor is less than 3°C.The overall quality of the building (wall thermal insulation, type of windows, shading control, HVAC system, free cooling, high efficiency heat recovery) may be considered high.

Simulation software

IDA Indoor Climate and Energy (ICE) is a tool for simulation of thermal comfort, indoor air quality and energy consumption in buildings. It covers a range of advanced phenomena such as integrated airflow and thermal models, CO_2 modelling, and vertical temperature gradients. The mathematical models are described in terms of equations in a formal language named Neutral Model Format (NMF). This makes it easy to replace and upgrade program modules [36]. GenOpt is an optimization program designed for finding the values of user-selected design parameters that minimize a so-called objective function (or cost function), such as annual energy use, leading to optimal operation of a given system. The minimization of a cost function is evaluated by an external energy simulation program. GenOpt can be coupled to any simulation program (e.g. EnergyPlus, IDA-ICE, TRNSYS, etc.) that reads its input from text files and writes its output to text files [37].

SIMULATED CASES

The first purpose of the paper is to investigate the energy need of a personalized ventilation system in comparison with a convectional mixing ventilation system for several control strategies of the supply air temperature (see Table 1 from Case 1 to Case 8). The second purpose of the paper is to explore the strategies having potential for energy-saving listed in the introduction (see Table 1 from Case 9 to Case 26). A mixing ventilation system supplying the air at a constant temperature (16°C) throughout the year is the reference case. All the simulated cases are summarised in Table 1 and described below.

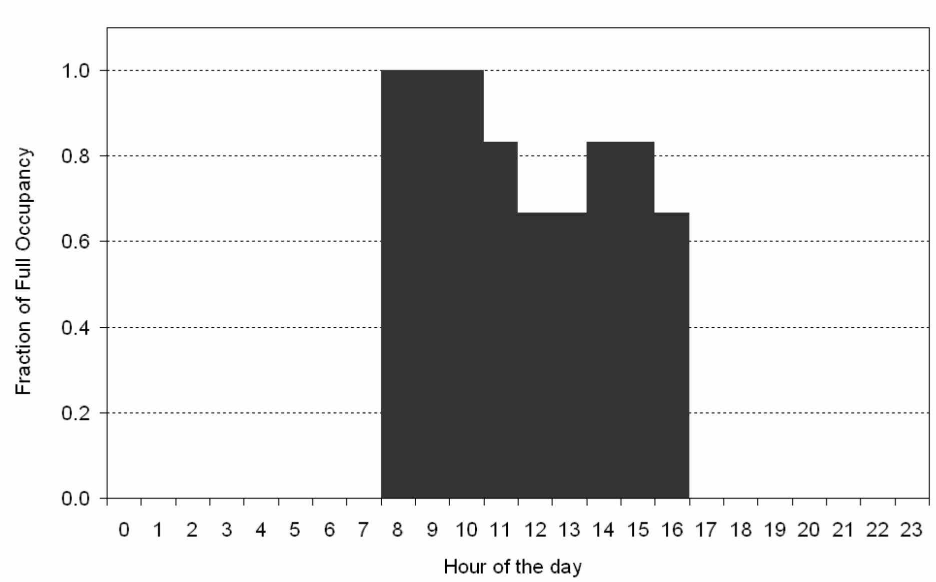

Figure 1 Occupancy profile according to the standard EN 15232 [39].

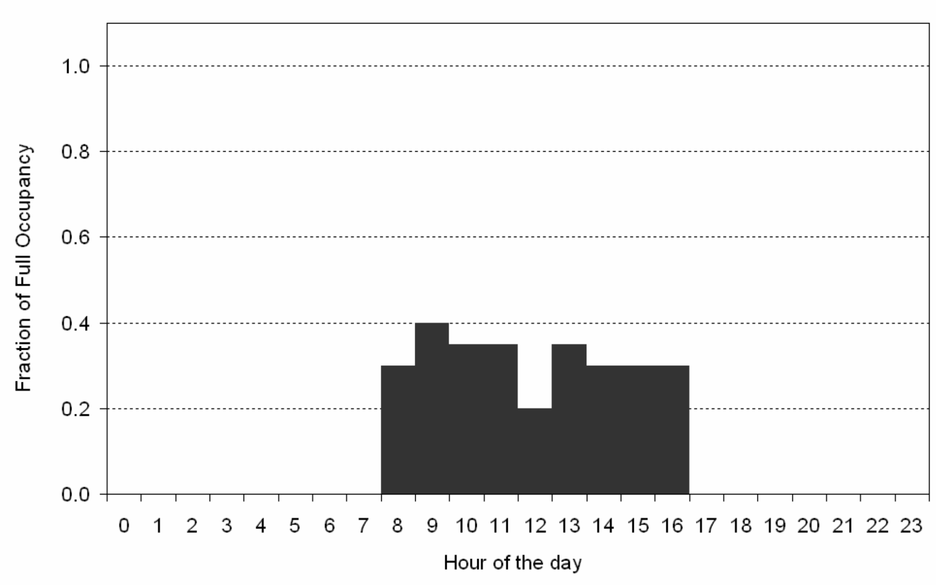

Figure 2 Occupancy profile according to the measured data by Nobe et al. [25].

Table 1 Simulated cases with personalized ventilation.

Case	Control strategy of the supply air temperature	Supply air temperature profile	θ_{UP} [a] [°C]	Airflow rate per person q_V [l/(s person)]	Occupancy from 8:00-17:00
1	Constant	20°C	25.5	20	Full
2	Constant	23°C	25.5	20	Full
3	Constant	26°C	25.5	20	Full
4	Outdoor	Figure 3	25.5	20	Full
5	Outdoor	Figure 3	25.5	20	Full
6	Outdoor	Figure 3	25.5	20	Full
7	Indoor	Figure 4	25.5	20	Full
8	Indoor	Figure 4	25.5	20	Full
9	Constant	20°C	25.5	5	Full
10	Constant	20°C	25.5	10	Full
11	Constant	20°C	25.5	15	Full
12	Indoor	Figure 4	25.5	5	Full
13	Indoor	Figure 4	25.5	10	Full
14	Indoor	Figure 4	25.5	15	Full
15	Constant	20°C	27	20	Full
16	Constant	20°C	28	20	Full
17	Constant	20°C	29	20	Full
18	Constant	20°C	30	20	Full
19	Indoor	Figure 4	27	20	Full
20	Indoor	Figure 4	28	20	Full
21	Indoor	Figure 4	29	20	Full
22	Indoor	Figure 4	30	20	Full
23	Constant	20°C	25.5	20	Figure 1
24	Constant	20°C	25.5	Varying [b]	Figure 1
25	Constant	20°C	25.5	20	Figure 2
26	Constant	20°C	25.5	Varying [b]	Figure 2

[a] The cooling systems tried to keep the room operative temperature below the upper room operative temperature limit.
[b] The airflow varies according to the occupation reported in Figure 1 and Figure 2. At full occupation the airflow is equal to 20 l/s per person.

Supply air temperature control (Case 1 - Case 8)

When the occupants are not provided with control over the temperature of the supplied personalized air, the building manager has to define the supply air temperature (θ_{SUP}) needed to provide the occupants with thermal comfort at a minimal level of energy consumption. In a single duct constant air volume system, θ_{SUP} set-point may be constant, or it may be reset based on the outdoor (θ_{ODA}) or indoor (θ_{IDA}) air temperature. PV supplies the air close to occupants. Therefore the lowest and highest permissible supply air temperatures are limited by thermal comfort issues. In this study it has been chosen that θ_{SUP} may vary in the range 20-26°C. All the θ_{SUP} profiles presented in the following are restricted within this range. In Case 1, 2, 3, θ_{SUP} was constant and equal to 20, 23, 26°C respectively. In Cases 4, 5, 6 (see Figure 3) the θ_{SUP} was reset according to θ_{ODA}. Two of them (Cases 4 and 5) were chosen by the authors and the other one, Case 6, was obtained using GenOpt (this software is discussed later in the paper). Cases 4 and 5 are characterized by supplying the personalized air at 20°C when the θ_{ODA}<20°C in order to minimize the heating energy that the Air Handling Unit (AHU) must provide to the supplied air. When θ_{ODA}>20°C the personalized air is supplied to the room without being conditioned. The profiles are limited in the upper part by a maximum supply air temperature equal to 22 and 26°C respectively. GenOpt software was used to find the optimal supply air temperature profile (Case 6) within the boundaries of the room air temperature given by EN 15251 [19] for category I of the indoor environment. GenOpt was set to minimize the sum of energy needed for heating and cooling of the outdoor supply airflow rate and the room (mathematically named cost function). In order to minimize the cost function, GenOpt changes the θ_{SUP} corresponding to the following fixed outdoor temperatures (-20, 10, 15, 18, 20, 21, 23, 25, 26, 27, 30, 40°C) by choosing an integer value within the range 20-26°C. In Cases 7 and 8 (see Figure 4) the θ_{SUP} was controlled by the θ_{IDA}, which is equal to the return air temperature in a mixing ventilation system. The Case 7 profile aims to maximize occupants' thermal comfort because it supplies hot air when it is chilly in the room and cool air when it is warm; the profile was named "comfort" profile. The authors expect that the "comfort" profile would probably be used by the occupants if they would have the opportunity to control the supply air temperature. In Case 8 the air is supplied isothermally within the range 20-26°C, based on recent findings indicating that elevated velocity at the breathing zone improves inhaled air quality and compensates for the negative impact of increased temperature on perceived air quality [38]. The profile was named "isothermal" profile.

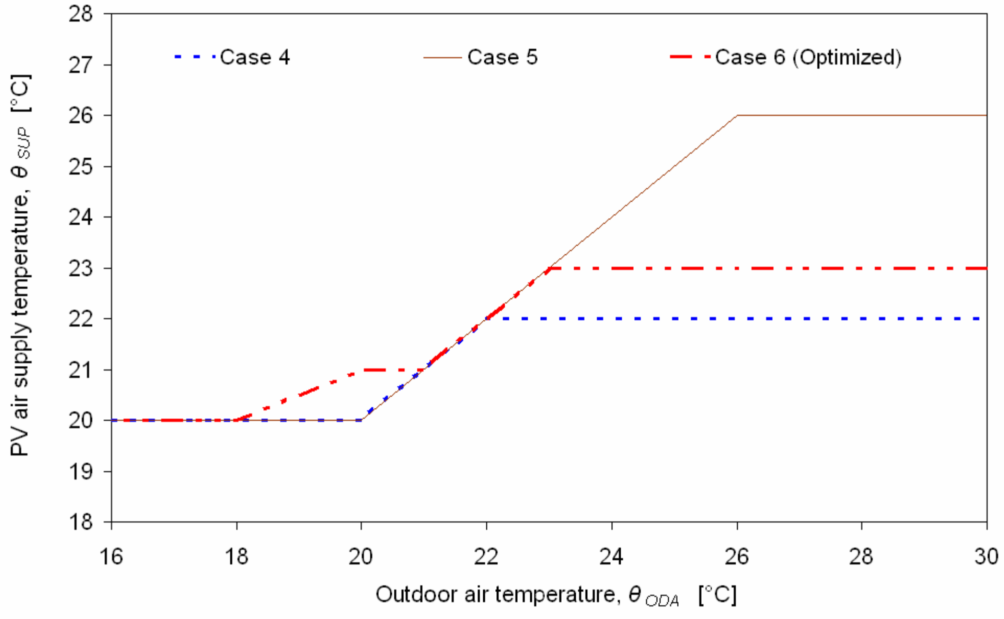

Figure 3 PV supply air temperature profiles as a function of the outdoor air temperature for Cases 4, 5 and 6 (See Table 1).

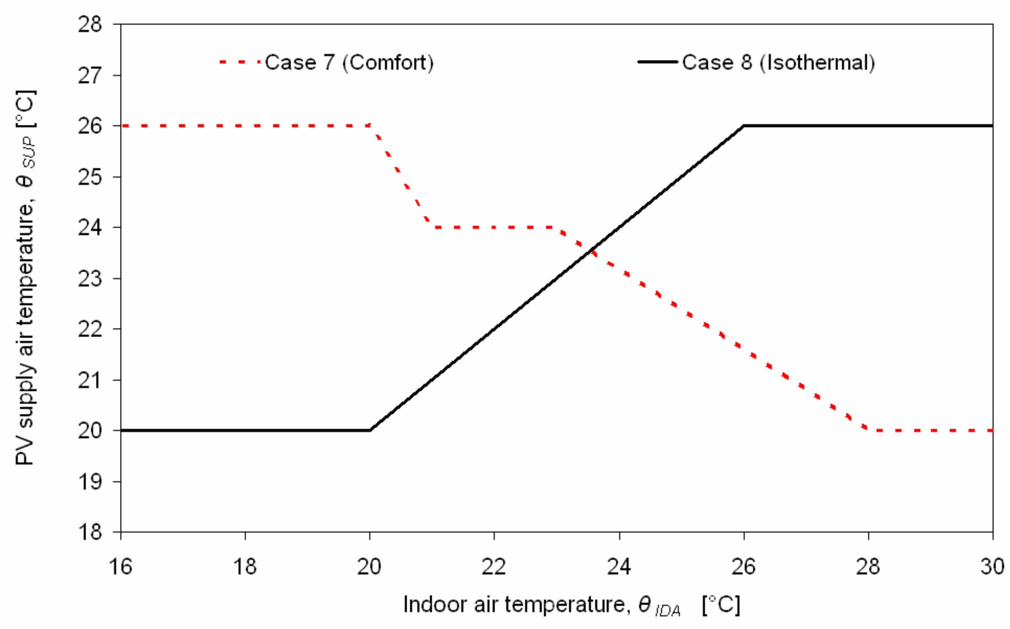

Figure 4 PV supply air temperature profiles as a function of the indoor air temperature for Case 7 and Case 8 (See Table 1).

Energy-saving strategies (Case 9 - Case 26)

The three energy-saving strategies presented in the paragraph "Energy-saving potentials" were investigated (from Case 9 to Case 26, see Table 1). Two supply air temperature strategies were used: supplying the air at 20°C constantly for the whole year (Case 1) and the "comfort" profile (see Figure 4, Case 7). The former has been chosen because from the simulation it was found that it is the strategy which minimizes the energy need.

From the Case 9 to Case 14 (see Table 1) the effectiveness of reducing the q_V was studied. q_V was reduced to 15, 10, and 5 l/s per person. These values correspond to a ventilation effectiveness of 1.34, 2 and 4 respectively. From the Case 15 to Case 22 (see Table 1) the effectiveness of expanding the θ_{UP} was studied. θ_{UP} was expanded from 25.5°C (corresponding to Category I of the indoor environment according to EN 15251 [19]) to 27, 28, 29, and 30°C. The lower room operative temperature was kept equal to 21°C because it was found (not reported in this paper) that reducing it (e.g. to 18°C) does not affect the energy need.

From the Case 23 to Case 26 (see Table 1) the effectiveness of supplying the personalized air only when the occupant is present at the desk was studied. Two occupancy behaviour profiles were used. In this paper the fraction of full occupancy is defined as the ratio between the actual number of occupants seated at the desk over the maximum number of occupants for whom the room was designed. The first occupancy behaviour profile (shown in Figure 1) has been obtained from the European standard EN 15232 [39]. The second profile (shown in Figure 2) has been extrapolated by the data measured by Nobe et al. [25] in a Japanese 52-story office building where 240 workstations were monitored for a week. The two profiles were bounded within the office hours used for previous simulations (from 8:00 to 17:00). In this study it is assumed that when the occupant is not at his/her desk he/she is out of the office. When the occupant is not at the desk the heat loads generated by him/her and his/her equipment is not taken into account, and in the Cases 24 and 26 the personalized air is switched off.

RESULTS

The "energy need" is the sum of energy for heating (AHU Heating) and cooling (AHU Cooling) of the supplied air in order to obtain the desired θ_{SUP} and for heating (Room Heating) and cooling (Room Cooling) of the conditioned space in order to maintain the indoor operative temperature within the designed range during a given period of time (from 6:00 to 17:00). The definition is in accordance with the European standard EN 15615 [40]. The energy need for several θ_{SUP} control strategies (Table 1, Cases 1 - 8) is shown in Figure 5. The energy need for the reduced outdoor airflow rates

(Table 1, Cases 9 - 14) is shown in Figure 6. The energy need for the expanded upper room operative temperature limits (Table 1, Cases 15 - 22) is shown in Figure 7. The energy need for personalized air supplied only when the occupant is present at the desk (Table 1, Cases 23 - 26) is shown in Figure 8.

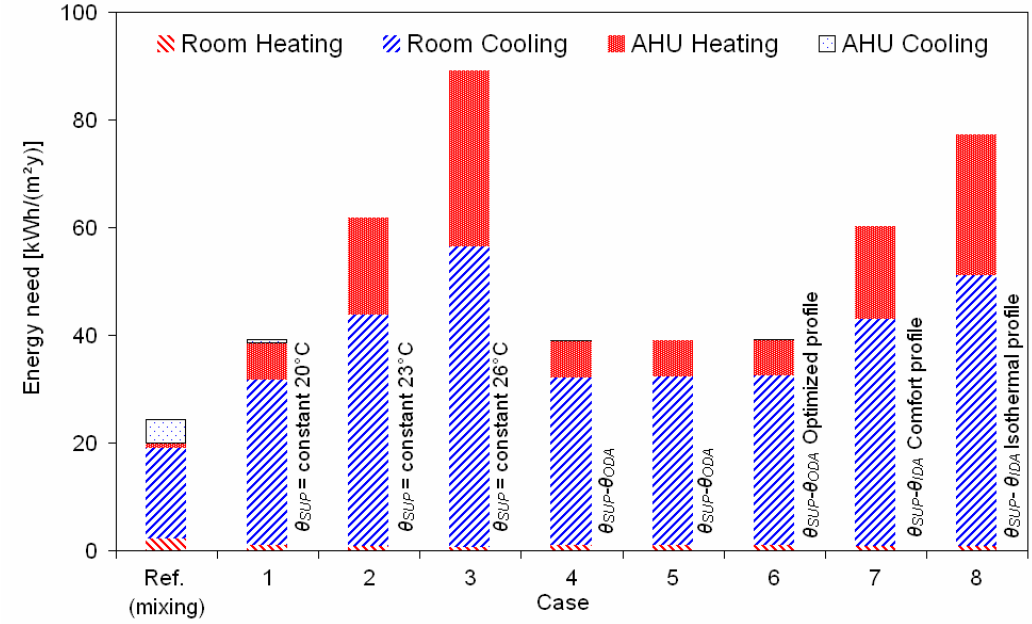

Figure 5 The energy need for several control strategies of the personalized supply air temperature, θ_{SUP} (see Table 1).

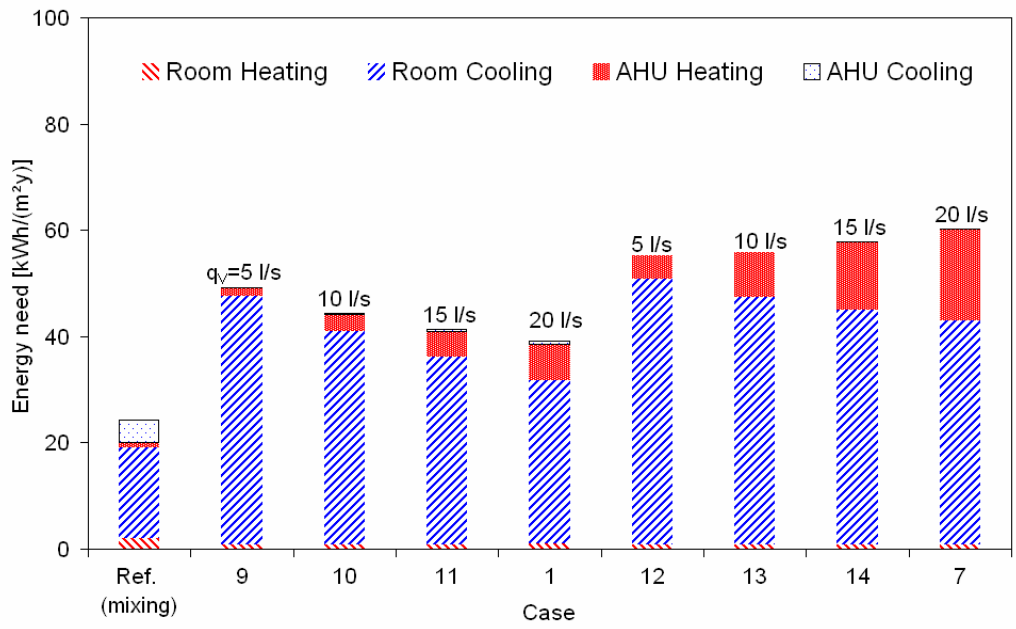

Figure 6 The energy need for the reduced outdoor airflow rates, q_V, for θ_{SUP} constant and equal to 20°C (Cases 9, 10, 11, 1) and for θ_{SUP} following the comfort profile shown in Figure 4 (Cases 12, 13, 14, 7).

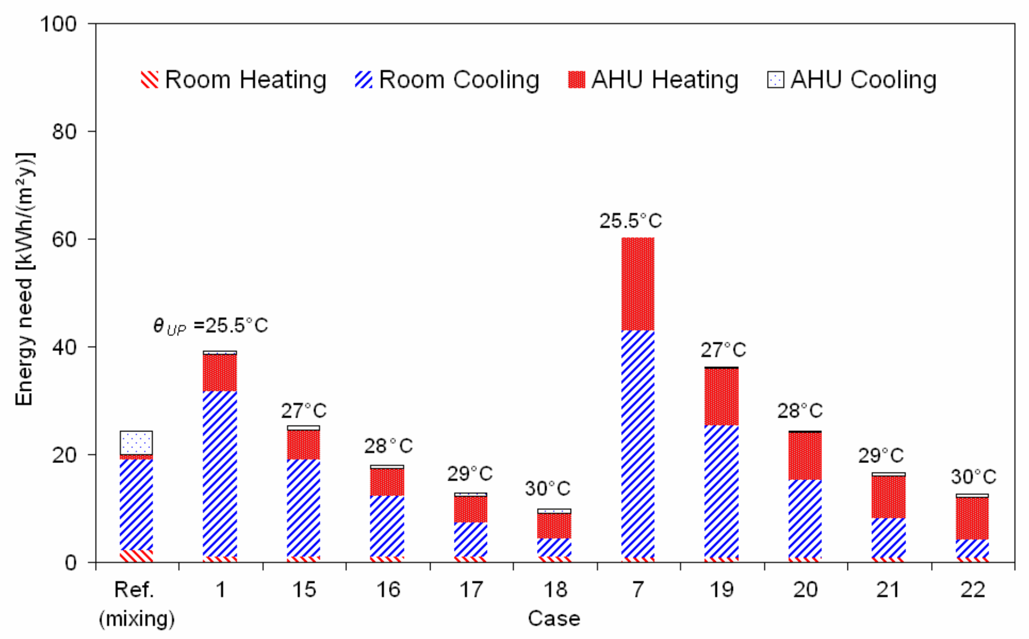

Figure 7 The energy need for the expanded room temperature comfort limits, θ_{UP}, rates for θ_{SUP} constant and equal to 20°C (Cases 1, 15, 16, 17, 18) and for θ_{SUP} following the comfort profile shown in Figure 4 (Cases 7, 19, 20, 21, 22).

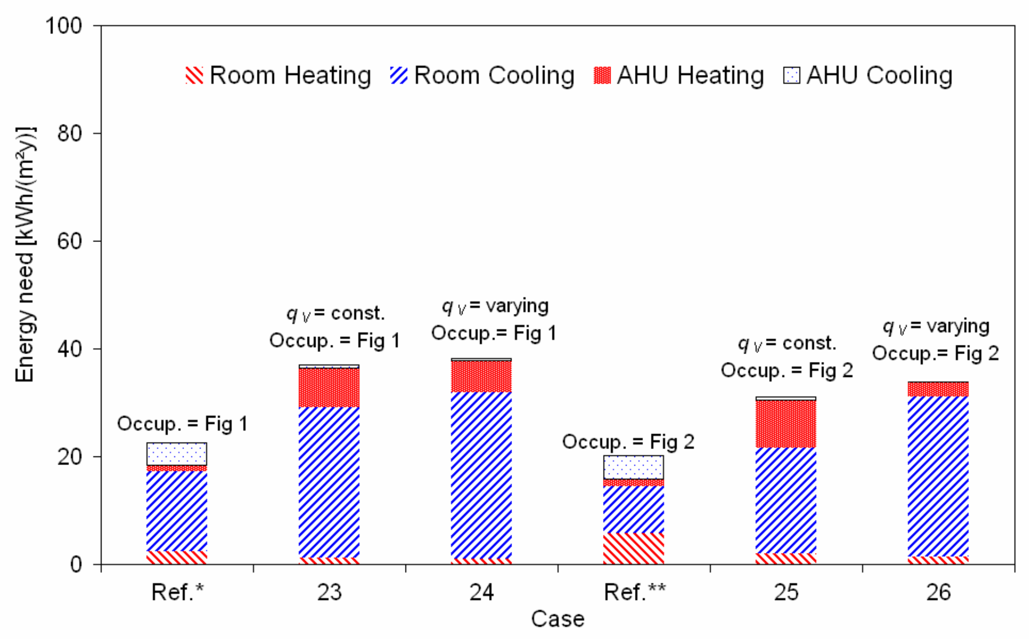

Figure 8 The energy need for personalized air supplied only when the occupants are present at the desk for the occupancy profile shown in Figure 1 and for the one shown in Figure 2. The airflow rate is constant in Case 23 and 25, and it varies according to the occupancy profile in Cases 24 and 26. Ref* and Ref ** are respectively the energy need for the reference case (mixing ventilation) and when the occupancy and the relative heat loads are varied according to the profiles shown in Figure 1 and Figure 2.

DISCUSSION

Influence of the supply air temperature on energy need (Case 1 - Case 8)

The results shown in Figure 5 reveal that the simulated building needs mainly cooling. Room Heating is needed only for the reference case (mixing ventilation supplying air at 16°C). The building has a good insulation and air tightness and the internal heat gains are sufficient to maintain the required operative temperature. The supplied personalized air needs to be cooled only sporadically; in fact AHU Cooling is equal to zero except for the reference case. The supply temperature and its control strategy have a marked influence on energy consumption. The energy need for the simulated cases is in the range 39.0-89.2 kWh/(m^2y). The energy need for the reference case is 24.3 kWh/(m^2y); it means that by using PV the energy need increases from 61% to 268%. This is mainly due to the fact that the lowest supply air temperature for the PV system was limited to 20°C for comfort reasons. In the reference case the air is supplied at 16°C. The building needs mainly cooling and the need for warming the personalized supplied air up to 20°C is a heat load (AHU Heating) that later has to be removed by the cooling system (Room Cooling). This phenomenon can be seen in Figure 5 by subtracting the AHU Heating from the Room Cooling; the remaining Room Cooling is almost constant in the range between 23.2 and 25.2 kWh/(m^2y). To supply the air at an elevated temperature of 23°C or 26°C (Cases 2 and 3) required a greater amount of energy than to supply at 20°C (Case 1). The energy needs for Cases 1, 4, 5, and 6 are almost equal, i.e. the different supply air temperature control strategies do not differ with regard to the energy need. The reason can be understood by analysing the outdoor air temperature cumulative profile. In Copenhagen the outdoor air temperature is higher than 20°C only 3.2% of the time in one year, higher than 22°C only 1.3%, higher than 24°C only 0.5%, and higher than 26°C only 0.1% of the year. Therefore, controlling the supply air temperature, θ_{SUP}, based on the outdoor temperature, θ_{ODA}, using profiles that differ only for θ_{ODA}>20°C, does not make any significant difference with regard to energy need. Controlling the θ_{SUP} by the indoor air temperature, θ_{IDA} (Case 7 and Case 8) implies high energy consumption. Case 7 has an energy need almost equal to Case 2, where θ_{SUP} = 23°C, but from a thermal comfort point of view, it would perform better. For the simulated building and for the assumptions made in this paper, the best supply air temperature control strategy is to provide air constantly at 20°C, the minimum permissible supply temperature.

The supply air temperature of a personal ventilation system has a marked influence on the energy consumption because it may become a significant heat load that needs to be removed. In a mixing ventilation system the outdoor air, after been conditioned, can be mixed with the recirculated air to reach the desired supply air temperature. This cannot be done with a PV system if its main aim is to improve significantly the inhaled air quality and to reduce the risk of spread of diseases.

Analysis of the energy-saving strategies (Case 9 - Case 26)

The energy-saving strategies with personalized ventilation were studied with Cases 9 – 26, as defined in the Method section, sub-section "Energy-saving strategies". The results are shown in Figure 6, 7 and 8. The influence of reducing the personalized flow rate, q_V, thanks to the higher ventilation effectiveness on the energy need, is shown in Figure 6. In Cases 9, 10, 11 and 1 q_V is equal to 5, 10, 15 and 20 l/s per person respectively, and the θ_{SUP} is in all cases constant and equal to 20°C. In Cases 12, 13, 14 and 7 q_V is equal to 5, 10, 15 and 20 l/s per person respectively, and the θ_{SUP} is a function of the θ_{IDA} and varies according to the "comfort" profile (see Figure 4, Case 7). In all cases the energy need is determined mainly by the AHU Heating and the Room Cooling. From Figure 6 it can be deduced that reducing q_V implies: a reduction of AHU Heating because the amount of outdoor air that needs to be heated is reduced and an increase of the Room Cooling because the outdoor air has a free cooling effect. Therefore, reducing q_V is beneficial only when the decrement in AHU Heating is higher than the increment in the Room Cooling. This is valid for the Cases 12, 13, 14, and 7 but not for the Cases 9, 10, 11, and 1 because the supply air does not need to be warmed up more than 20°C. When the θ_{SUP} is kept constant and equal to 20°C (Cases 1, 11, 10, 9) the energy need increases from 39.2 kWh/(m^2y) to 49.3 kWh/(m^2y) with the decrease of q_V from 20 to 5 l/s per person which corresponds to 26% of energy penalty. In this case it is not an advantage to reduce the airflow because the supplied air has a free cooling effect. When θ_{SUP} follows the "comfort" profile the energy need slightly decreases from 60.2 kWh/(m^2y) to 55.2 kWh/(m^2y) with the decrease of q_V from 20 to 5 l/s per person. In this case energy is reduced by 8% and it is an advantage to reduce the airflow. In conclusion, in a cold climate, reducing the personalized airflow rate does not always lead to a reduction of energy need because the outdoor air may have a free cooling effect. PV requires more energy than the reference case (mixing ventilation) even if the temperature of the supplied personalized air follows the applied "comfort" profile. However, it is believed that reducing q_V would always lead to energy-saving in hot and humid climates.

The influence of extending the upper room operative temperature, θ_{UP}, on energy need is shown in Figure 7. In Cases 1, 15, 16, 17 and 18 θ_{UP} is equal to 25.5, 27, 28, 29, and 30°C respectively, and the personalized supply air temperature, θ_{SUP}, is constant and equal to 20°C. In Cases 7, 19, 20, 21 and 22 θ_{UP} is equal to 25.5, 27, 28, 29, and 30°C respectively, and the θ_{SUP} follows the "comfort" profile. Also in these cases the energy need is determined mainly by the AHU Heating and the Room Cooling. From Figure 7 it can be deduced that increasing θ_{UP} implies a significant decrease of the Room Cooling, but it does not affect substantially the AHU Heating. Therefore, extending the upper room operative temperature limit is always beneficial. Independently of the θ_{SUP} strategies, the extension of θ_{UP} leads to energy need reduction, and when θ_{UP} is equal or higher than 28°C, using the personal ventilation system implies less energy need than the reference case of mixing ventilation.

The results in Figure 7 show that when the θ_{SUP} is kept constant and equal to 20°C and θ_{UP} is increased from 25.5 to 30°C, the energy need decreases from 39.2 kWh/(m^{2}y) to 9.9 kWh/(m^{2}y), corresponding to 75% of energy-saving (Cases 1, 15, 16, 17, 18). When θ_{SUP} follows the "comfort" profile (Figure 4) and θ_{UP} is increased from 25.5 to 30°C the energy need decreases from 60.2 kWh/(m^{2}y) to 12.7 kWh/(m^{2}y), corresponding to 79% of energy-saving (Cases 7, 19, 20, 21, 22). This energy-saving strategy is an effective way of reducing the energy need. However, it can be recommended only in the working environment where the occupants spend most of their time at their workstation in a comfortable thermal environment achieved by personalized ventilation.

The influence of supplying the personalized air only when the occupant is at the desk is shown in Figure 8.

Ref.* and Ref.** are the energy needs for the reference case (mixing ventilation) when the internal heat load generated by occupants and equipment follows the occupancy profiles reported in Figure 1 and Figure 2 and the ventilation airflow is constant. This leads to an energy decrease from 24.3 kWh/(m^{2}y) to 22.6 kWh/(m^{2}y) for the Ref.* case and to 20.2 kWh/(m^{2}y) for the Ref.** case. This means that the reduction of the internal heat load generated by occupants and equipment implies a reduction of 7% and 17% respectively for the occupancy profiles shown in Figure 1 and Figure 2. The energy need for the reference case was recalculated in order to be comparable (same internal heat load) with the energy need with the PV.

Supplying the personalized air only when the occupant is at the desk implies lower airflow rates. As in the previous cases (Cases 9-14) the reduction of the airflow rate causes two effects, a reduction of the AHU Heating (less outdoor air needs to be warmed up) and an increase of the Room Cooling (reduced free cooling). From Figure 8 it can be seen that for both occupancy profiles it is not effective to supply the airflow rate only when people are at the desk. When the airflow rate is adjusted according to the occupancy profile shown in Figure 1, the energy need slightly increases from 37 kWh/(m^{2}y) to 38.3 kWh/(m^{2}y), corresponding to 3% of energy penalty (Cases 23 and 24). When the airflow rate is adjusted according to the occupancy profile shown in Figure 2, the energy need increases slightly from 31.1 kWh/(m^{2}y) to 33.9 kWh/(m^{2}y), corresponding to 9% of energy penalty (Cases 25 and 26). This energy-saving strategy is not effective for reducing the energy need.

It has been documented that personalized ventilation may provide better inhaled air quality, thermal comfort and protection from cross-infection compared to mixing ventilation [1, 41, 42]. The results of this study reveal that in a cold climate, depending on the θ_{SUP} control strategy and on the energy-saving strategies applied, this can be achieved with higher (up to almost 4 times), equal or lower (up to 60% of energy-saving) energy consumption compared to traditional systems. In hot and humid climates where the outdoor air cannot be used for free cooling, the energy-saving strategies described may provide higher energy-saving than the one reported in this paper for a cold climate.

In this paper, only the energy-saving potential has been studied. The most important benefit in the use of personalized ventilation for the improvement of occupants' health, comfort and performance, as well as protection against cross-infection, has not been considered. It is important to note that the advantages and savings due to improvement of these factors may be much higher than the energy consumption of personalized ventilation.

CONCLUSIONS

The main conclusions of this study on energy-saving potential of personalized ventilation when used in cold climates are:

- The control strategy of the personalized air temperature supplied has a significant influence on energy consumption. In cold climates the energy consumption with personalized ventilation may increase substantially (between 61% and 268%) compared to mixing ventilation alone if energy-saving strategies are not applied.
- The best supply air temperature control strategy is to provide air constantly at 20°C, the minimum permissible supply temperature.
- The most effective way of saving energy with personalized ventilation is to increase the maximum permissible room temperature (saving up to 60% compared to the mixing ventilation may be achieved) but it can be applied only in offices where occupants spend most of their time at the desk.

- Reducing the airflow rate does not always imply a reduction of energy consumption because the outdoor air may have a free cooling effect. Supplying the personalized air only when the occupant is at the desk is not an effective energy-saving strategy.

NOMENCLATURE

ACE	air change effectiveness
AHU	air handling unit
AHU Cooling	Energy that is extract by the AHU from the outdoor airflow rate in one year (kWh/(m^2y))
AHU Heating	Energy that is supplied by AHU to the outdoor airflow rate in one year (kWh/(m^2y))
PV	personalized ventilation
q_V	personalized volume airflow rate per person (l/(s person))
Room Cooling	Energy that is extracted by the fan coil units from the room in one year (kWh/(m^2y))
Room Heating	Energy that is supplied by the fan coil units to the room in one year (kWh/(m^2y))
SBS	sick building syndrome
Greek symbols	
θ_{IDA}	indoor air temperature (°C)
θ_{ODA}	outdoor air temperature (°C)
θ_{SUP}	supply air temperature (°C)
θ_{UP}	upper room operative temperature limit (°C)

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PAPER C

Schiavon, S. and Melikov, A. 2008. Introduction to the Cooling Fan Efficiency Index. (Submitted to *HVAC&R Research*)

Introduction to the Cooling Fan Efficiency Index

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ABSTRACT

In a warm environment air movement with elevated velocity is a well-known cooling strategy. The air movement can be generated by cooling fans (e.g. ceiling fan, table fans, etc.). Appearance, power input and price are the main parameters considered today when purchasing cooling fans, while their cooling capacity and efficiency of energy use are unknown. A new index is introduced, named "cooling fan efficiency index" defined as the ratio between the cooling effect (measured with a thermal manikin) generated by the device and its power consumption.

The index was determined for a ceiling fan, a desk fan, a standing fan and a tower fan in a real office at three room air temperatures and at different fan velocity levels. The results revealed that the index is sensitive enough to identify differences in the performance of the cooling devices. The desk fan had a significantly ($p<0.01$) higher efficiency than the other three fans tested. The cooling fans generate a non-uniform velocity field around occupants which cannot be described with a single value. This makes the recommendation in the actual standards for elevated velocity in a warm environment difficult to apply in practice.

INTRODUCTION

In a warm environment, elevated air movement is a widely used strategy for the cooling of occupants. The air movement increase can be produced by several devices such as cooling fans (ceiling, floor standing, tower and table fans), furniture-installed personalized ventilation, body-attached ventilation devices and, under certain conditions, with operable windows. The underfloor air distribution system, which is one of the total volume ventilation principles used in practice also allows for increase or decrease of the velocity close to workplaces. The cooling capacity of cooling fans is limited because they operate under isothermal conditions, i.e. the cooling of the body is a result of increased velocity only. The use of cooling fans in practice is easy and does not require special installations. The personalized ventilation systems (Melikov 2004) and the task-ambient conditioning systems (Arens et al. 1991) perform better with regard to thermal comfort since they may operate under non-isothermal conditions, i.e. the supplied air can be cooled below the room air temperature in addition to elevated velocity. Appearance, power consumption and price are the main parameters considered when purchasing cooling fans, while their cooling capacity and efficiency of energy use are unknown. Other factors such as ergonomics, control options, etc. are also important. Comparison of the performance of cooling fans from the point of view of cooling capacity and energy consumption is important for their application in practice.

According to the international standards on thermal comfort (ASHRAE 2004; ISO 2005; CEN 2007) elevated air speed can offset the indoor temperature rise and provide occupants with thermal comfort. This can be achieved by providing occupants with the opportunity to individually control locally applied air movement, i.e. air speed. A relationship between the air speed and the upper operative temperature limits is included in the present standards in graphical form (ASHRAE 2004; ISO 2005; CEN 2007). The body surface area exposed to the air movement is also important for the heat exchange between the body and the environment. This, however, is not discussed in the standards.

It has been suggested (Sekhar 1995; Olesen and Brager 2004; Aynsley 2005) that setting a high room temperature and cooling of the body by elevated air movement lead to a substantial energy saving. Schiavon and Melikov (2008), by means of energy simulations, found that the required power input of the fan is a critical factor for energy saving. The results obtained for the boundary conditions of their study reveal that traditional cooling devices, such as ceiling, standing, tower and desk fans, may consume more electrical energy than is saved by not using a traditional HVAC system. Thus, knowledge as to how efficiently fans of different types use the electrical energy for cooling occupants is needed in order to justify the use of the strategy of elevating the room temperature at increased air movement.

In this paper an index is introduced that relates the cooling effect of fans generating local air movement in the vicinity of occupants with their energy consumption. Experiments with different cooling fans are performed to validate the usefulness of the index.

COOLING FAN EFFICIENCY INDEX

The efficiency is the ratio of the output to the input. It can be improved by reducing input and/or improving output. In the case of fans, used for cooling people in warm environments by increasing the air velocity around the human body, the input is the electrical energy needed for running the fan (the power requirement of a fan is almost constant and it can be used instead of energy in order to make the input variable time-independent) and the output is the body cooling effect.

The body cooling effect produced by a fan depends on generated air velocity and turbulence field, body area exposed to moving air, body posture, air and mean radiant temperature, air humidity, clothing insulation, metabolic rate, humidity, and skin wettedness. Sophisticated thermal manikins with full body size and a complex shape have been developed and used for determination of the heat loss from the human body under different environmental conditions (Tanabe et al. 1994; Tsuzuki et al. 1999; Melikov et al. 2002). A manikin's body is typically divided into several segments. They can be operated to maintain constant heat flux from the body, constant body surface temperature, or to have surface temperature equal to the skin temperature of an average person in a state of thermal comfort under the particular environmental condition of the exposure. Thermal manikins can be used to measure the fan cooling effect and thus to determine the cooling fan efficiency index. Thermal manikins that can measure dry heat loss from the human body are most commonly used today though sweating thermal manikins are under development as well (Psikuta et al. 2008). Therefore at this stage, dry heat loss from the human body can be used for determining the cooling fan efficiency. Clothing thermal insulation and metabolic rate (personal factors that may vary substantially in real life) can be assumed to be constant, while air humidity and skin wettedness are not taken into account. The equivalent temperature (t_{eq}) is a well-known parameter that can be used for determining the cooling fan efficiency index. In the SAE (1993), equivalent temperature (former Equivalent Homogenous Temperature) is defined as: "The uniform temperature of the imaginary enclosure with air velocity equal to zero in which a person will exchange the same dry heat by radiation and convection as in the actual non-uniform environment". The same definition was used by Nilsson et al. (1999). In the definition it is assumed that the body posture, the activity level and the clothing design and thermal insulation is the same in both environments. The equivalent temperature is a pure physical quantity that in a physically sound way integrates the independent effects of convection and radiation on human body heat loss. t_{eq} does not take into account human perception and sensation or other subjective aspects, but may correlate with them. It is important to notice that t_{eq} is not a temperature that can be measured by a thermometer and that t_{eq} cannot be translated to an air temperature in a complex climate (Bohm et al. 1999). The body cooling effect achieved by air movement can be quantified by the change in whole-body manikin-based equivalent temperature, t_{eq}, from the reference condition, t_{eq}* (similar indoor environmental conditions but without air movement), i.e. $\Delta t_{eq} = t_{eq} - t_{eq}^*$. The concept of Δt_{eq} has been already used by several authors to quantify the whole-body cooling effect of air movement (Tanabe et al. 1994; Tsuzuki et al. 1999; Melikov et al. 2002; Watanabe et al. 2005; Sun et al. 2007). Thus, the Cooling Fan Efficiency (*CFE*) is defined by Equation 1.

$$CFE = (-1)\frac{\Delta t_{eq}}{P_f} \qquad (1)$$

where
P_f = fan power. It is the input power of the fan (define according to CEN 2003).
$\Delta t_{eq=}$ whole-body cooling effect.

The measuring unit of *CFE* is °C/W (°F/W). Δt_{eq} would be usually negative (the equivalent temperature of the body cooled by a fan would be lower that the temperature without the fan). In order to have an index that is easy to interpret, the ratio between the cooling effect and the fan power has been multiplied by -1 (Equation 1). The higher the *CFE* index, the better the fan performance.

Figure 1 shows the cooling fan efficiency as a function of the fan power calculated at cooling effect Δt_{eq} of -0.5, -1, -2, -3 and -4°C (-0.9, -1.8, -3.6, and -7.2°F). It has been reported that a cooling effect of -4°C (-7.2°F) obtained by local body cooling can be acceptable for people (Watanabe et al. 2005 and 2008). An internet survey showed that the typical power consumption of cooling fans is lower than 90 W. The results in the figure show that at constant cooling effect the *CFE* increases with the decrease of the fan power, i.e. fans with different power may have the same cooling effect. The results also show that fans with the same air power may have a different cooling effect due to differences in the generated flow, e.g. different target area, velocity and turbulence field, etc.

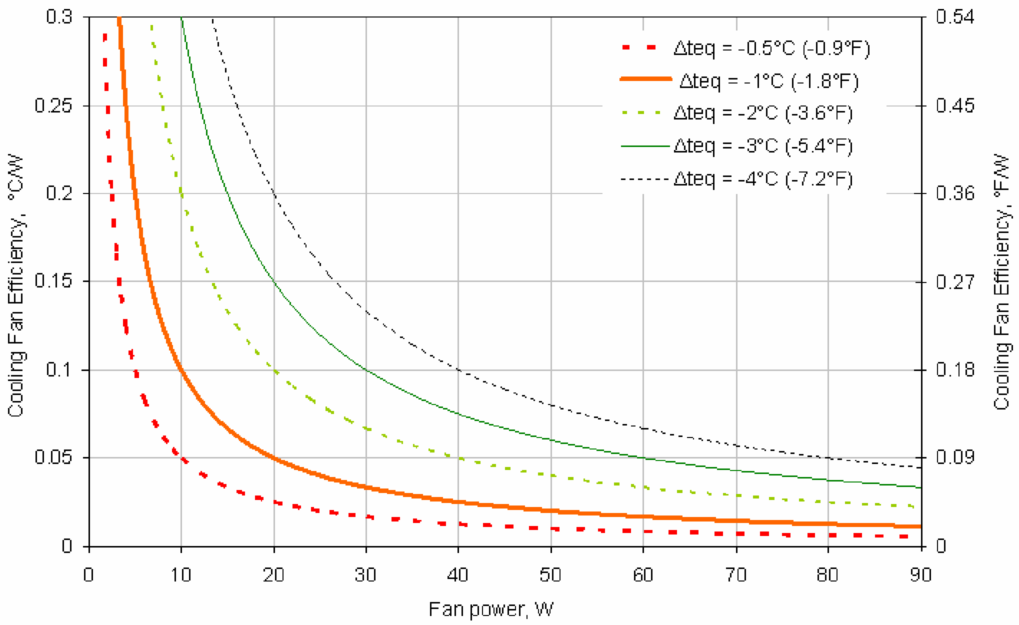

Figure 1. Cooling fan efficiency versus fan power for five cooling effect levels.

Knowing the cooling fan efficiency index (*CFE)* and its cooling effect *(*Δt_{eq}*)* will help customers to purchase a better fan, fan designers/manufacturers to assess and develop better products, and policymakers to fix minimum values or classes of fan efficiency as is usually done with other electrical appliances (e.g. air-conditioner, refrigerators, boilers, etc.). HVAC designers may choose the summer maximum allowed room temperature, depending on the cooling capacity of the fan, as well as evaluate the possibility for energy saving based on the strategy of increased air movement at elevated room air temperature.

EVALUATION OF THE *CFE* INDEX OF COOLING FANS

The usefulness of the introduced cooling fan efficiency index, *CFE*, for comparison of cooling fans was demonstrated. Experiments were performed with four fans available on the market including a ceiling fan (CF), a desk fan (DF), a standing fan (SF) and a tower fan (TF). The index of the cooling fans was determined and compared.

Method

Experimental facilities

The fans used are described in Table 1. They were purchased for the purpose of these experiments. The rotation speed of the fans (velocity of the generated flow is expected to increase with the rotation speed), is defined by the manufacturers and can be varied in steps. The desk and the standing fans have two velocity levels and the ceiling and tower fans have three velocity levels. Experiments were performed in a real office room (5.8 m x 4.42 m x 3.5 m (19 x 14.5 x 11.5 ft)) with a suspended ceiling (0.5 m (1.6 ft) from the top). A double pane strip window (5.80 m (19.0 ft) width and 1.85 m (6.1 ft) height) is located in one of the walls. The lower edge of the window is 1.15 m (3.8 ft) above the floor. The window faces north-west. Solar radiation was shielded with internal blinds. During the experiments, the outdoor temperature was always lower than 22°C (71.6°F). The room temperature was controlled with an electrical heater managed by a PID controller. The room was not equipped with ventilation and air-conditioning systems. A workplace was arranged in the room (see Figure 2), and a desk was placed in the centre of the office.

Table 1. Main Characteristics of the Fans Used

Type	Velocity levels	Dimension, m (in.)	Power[a], W
Ceiling fan (CF) - axial fan	3	$Ø_{CF}$[b]= 1.20 (47)	65
Desk Fan (DF) - axial fan	2	$Ø_{DF}$[c]= 0.22 (9); h_{DF}[c]= 0.22 (9)	30
Standing Fan (SF) – axial fan	2	$Ø_{SF}$[e]= 0.39 (15); h_{SF}[e]= 1.10 (43)	50
Tower Fan (TF) - centrifugal fan	3	$Ø_{TG}$[f]= 0.15 (6); h_{TF}[f]= 0.45 (18); w_{TF}[f]= 0.08 (3); d_{TF}[f]= 0.46 (18)	50

[a] Nameplate fan power declared by the company.
[b] $Ø_{CF}$ = external diameter of the blades of the CF;
[c] $Ø_{DF}$ = external diameter of the blades of the DF; h_{DF} = height over the floor of the rotation axis of the DF.
[e] $Ø_{SF}$ = external diameter of the blades of the SF; h_{SF} = height over the floor of the rotation axis of the SF.
[f] $Ø_{TF}$ = external diameter of the blades of the TF; h_{TF} = height of the inlet opening of the TF; w_{TF} = width of the inlet opening of the TF; d_{TF}= diagonal of the opening of the TF.

Measuring instruments

A thermal manikin was used to simulate an occupant and to evaluate the cooling effect of the fans. The thermal manikin is 1.68 m (5.51 ft) tall and shaped as an average size Scandinavian woman. The total area of the manikin is 1.48 m^2 (15.93 ft^2). The body of the manikin consists of 23 independently controlled segments (see Appendix A), manufactured as polystyrene shells wound with embedded nickel wire, which serves to heat the body parts and monitor the ‘‘skin temperature’’. Low-voltage power is pulsed to each segment at a rate needed to keep the surface temperature of the manikin equal to the skin temperature of an average person in a state of thermal comfort. The power consumption under steady-state conditions is then a measure of the convection, radiation and conduction heat losses (dry heat loss). For each body segment the segmental equivalent temperature, $t_{eq,i}$, can be calculated using the following equation:

$$t_{eq,i} = t_{sk,i} - \frac{Q_{t,i}}{h_{cal,i}} \qquad (2)$$

where

$t_{sk,i}$ =surface temperature measured for the *i*-th segment;

$Q_{t,i}$ =sensible heat loss (power consumption) of the *i*-th segment;

$h_{cal,i}$=dry heat transfer coefficient, determined during calibration of the manikin in a standard environment.

The t_{eq} for the whole-body is obtained by computing the area-weighted average over all the body segments (see Appendix A). A multichannel low velocity thermal anemometer with omnidirectional velocity transducers was used to perform mean velocity, turbulence intensity and air temperature measurements at several points in the room. The characteristics of the anemometer comply with the requirements for such instruments specified in the standards (ISO 1998; ASHRAE 2005). The room air temperature was measured also with a mercury thermometer. The relative humidity was monitored but not controlled. The resolution of the used hygrometer was 0.1% RH. The fan power input was measured with a power-meter.

Experimental conditions

The performance of the four fans was studied at three room air temperature levels, namely t_a= 25, 27, 30°C (77, 80.6, 86°F). Two velocity levels for the desk fan and the standing fan and three velocity levels for the ceiling fan and the tower fan were explored. Measurements were also performed in a still environment without fan. The experiments were performed in randomized. Throughout the experiments the room temperature and the relative humidity vary within the ranges reported in Table 2.

Table 2. Room Air Temperature and Relative Humidity During the Experiments

Temperature set point		Measured room air temperature		Relative humidity, %
°C	°F	°C	°F	
25	77	24.9 - 25.2	76.82-77.36	22.7 – 44
27	80.6	26.8 - 27.2	80.24-80.96	23.5 - 34.5
30	86	29.9 - 30.2	85.82-86.36	21.5 - 31.7

The experimental set-up and the location of the fans is shown in Figure 2. The CF was installed in the centre of the room. The distance between the suspended ceiling and the blades was 0.25 m (0.82 ft) and between the blades and the floor it was 2.75 m (9.02 ft). The DF was located in front of the manikin on the table on the left side of the laptop at a distance of 0.66 m (26 in.) (three times its diameter) from the manikin. The SF was located on the left side of the manikin at a distance of 1.17 m (46 in.) (three times its diameter) distance. The TF was located on the left side of the manikin at a distance of 1.35 m (53 in.) (three times the diagonal of its opening). The thermal manikin was dressed with a long-sleeved shirt, thin long trousers, panties, ankle socks and shoes. This typical summer office clothing was equal to 0.47 clo. The manikin was seated upright on an office chair (0.15 clo).

Experimental procedure

The surface temperature, $t_{sk,i}$, and the power consumption, $Q_{t,i}$, were recorded for 10 min after steady-state conditions were obtained, i.e. when the difference in the average surface temperature of the manikin during the last 10 minutes was less than 0.05°C (0.09°F). The fan power was manually recorded while logging the manikin data. The manikin was then moved from the desk and the mean air velocity and the turbulence were measured at its location at four heights (0.2, 0.6, 1.1 and 1.7 m) (8, 24, 43 and 67 in.). Three-minute velocity measurements were performed as recommended in the indoor climate standards.

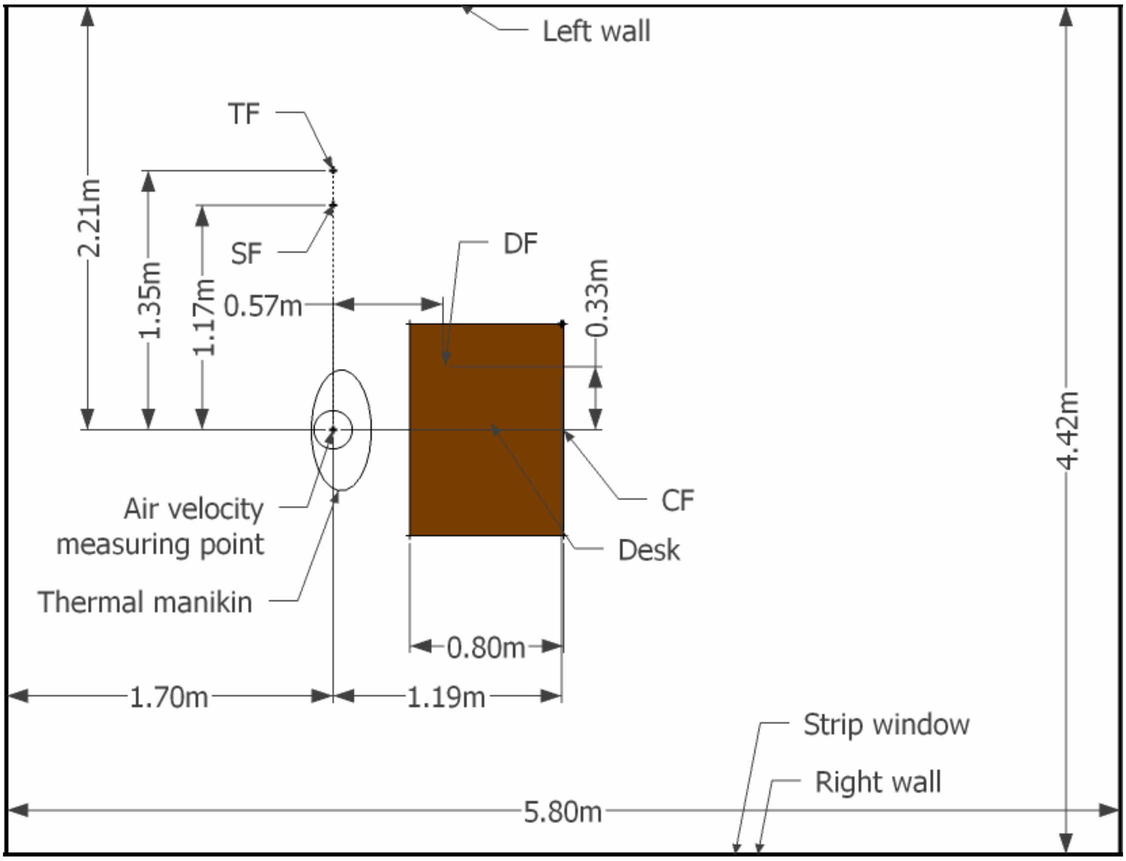

Figure 2. Room plan (5.8 x 4.42 x 3.5 m (19 x 14.5 x 11.5 ft)). Location of the office desk, thermal manikin, ceiling fan (CF), desk fan (DF), standing fan (SF) and tower fan (TF).

Measurements uncertainty

The description of the uncertainties of the measured and derived quantities is reported in Appendix B.

Statistical analysis

t-tests were performed to determine whether the type of fan or the room air temperature has a significant influence on the cooling effect and the cooling fan efficiency index. The tests were performed using S-Plus (Insightful 2007). P-values less than 5 % ($p<0.05$) were considered to be statistically significant.

Results

The cooling effect, Δt_{eq}, the fan power, P_f, and the cooling fan efficiency index, *CFE*, were obtained for each of the four fans under the experimental conditions studied. These are listed in Table C-1, Appendix C. The results identify a large variation in the whole-body cooling effect (between -3.2 and -0.4°C (-5.76 and -0.72°F)), in the fan power (between 15.6 and 49.3 W), and in the *CFE* index (between 0.009 and 0.177°C/W (0.016 and 0.319°F/W)).

Cooling fan efficiency index

The results obtained with the four fans at the room air temperatures and velocity levels studied are compared in Figure 3. The desk fan has the highest *CFE* index (*CFE* varies between 0.095 and 0.177°C/W (0.171 and 0.319°F/W)) and the smallest power consumption (P_f varies between 16 and 20 W). The ceiling, the standing and the tower fans have similar results: *CFE* and P_f for the ceiling fan, the standing fan and the tower fan varied respectively in the ranges 0.018 - 0.079°C/W (0.032 and 0.142°F/W) and 37 - 48 W, 0.038 - 0.058°C/W (0.068 and 0.104°F/W) and 33 - 40 W and 0.009 - 0.066°C/W (0.016 and 0.119°F/W) and 37

- 49 W. The results also indicate that the *CFE* of the desk fan is substantially more sensitive to the changes in the room air temperature and velocity level than the *CFE* of the other three fans. The *CFE* of the standing fan is least affected by the change of the room air temperature and fan velocity.

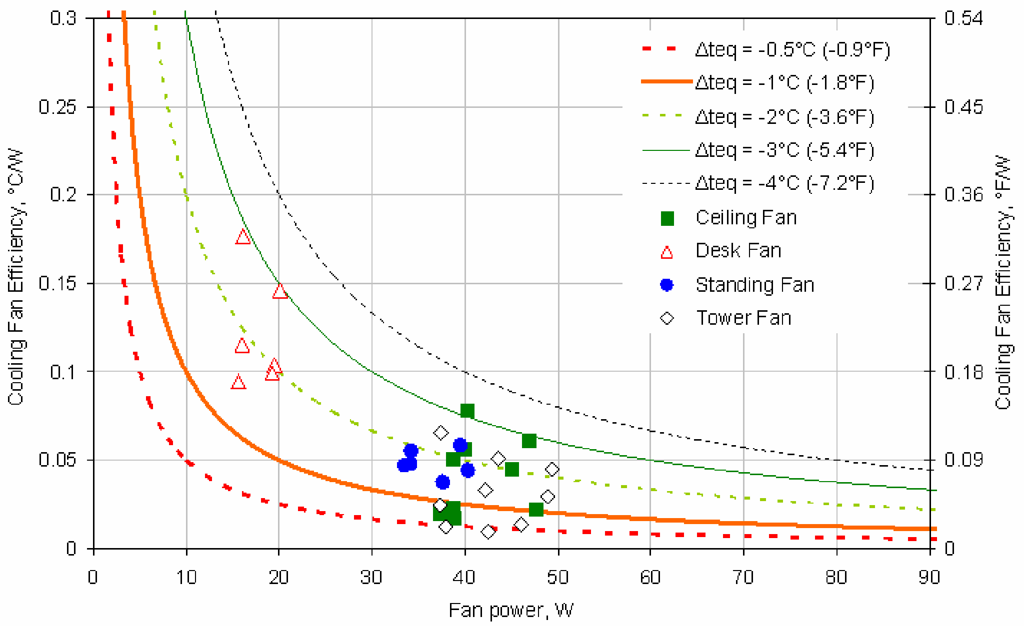

Figure 3. Fan power versus cooling fan efficiency index for the ceiling fan (CF) the desk fan (DF), the standing fan (SF), and the tower fan (TF). Lines with constant cooling effect (Δt_{eq}) are plotted.

The average of the cooling fan efficiency obtained for different room air temperatures and velocity levels with each of the fans was calculated. It is compared in Figure 4. The sample standard uncertainty of the index is equal to ±0.009°C/W (±0.016°F/W) (see Appendix B). The desk fan is the most effective cooling device; its cooling fan efficiency (*CFE*=0.123°C/W (0.221°F/W)) being more then double the index of the other fans (between *CFE*=0.032-0.048°C/W (0.058-0.086°F/W)). The tower fan is the least efficient cooling device. The efficiency of the DF is significantly ($p<0.01$) higher than the *CFE* of the remaining three fans. No significant difference in efficiency of these three fans was found (except that the *CFE* of the SF is higher than the *CFE* of the TF).

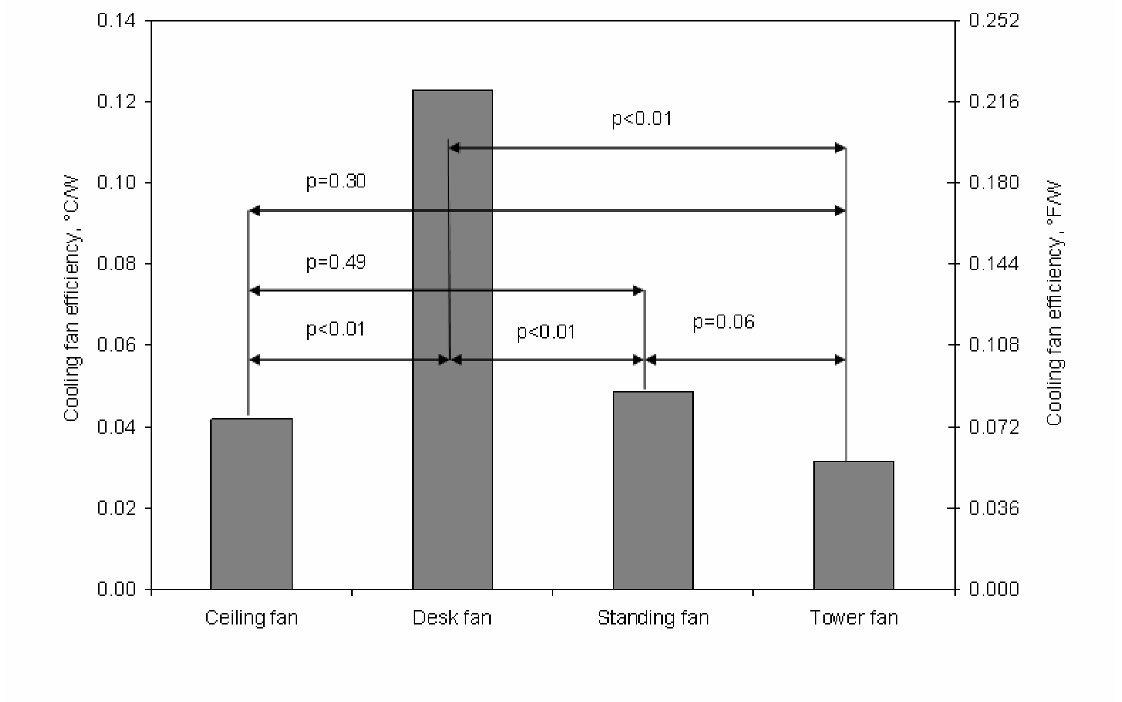

Figure 4. Averaged (over the velocity levels and room air temperature) cooling fan efficiency index for the ceiling, desk, standing and tower fan.

The influence of the room air temperature on the *CFE* was analysed. The average of the *CFE* index obtained with the four fans at the tested velocities was calculated for each of the room air temperatures. The results are compared in Figure 5. From a heat transfer point of view the room air temperature has an influence on the cooling effect, and thus should have an influence on the cooling fan efficiency index. A significant ($p<0.01$) difference was found between the *CFE* determined at 25°C (77°F) and the *CFE* at 30°C (86°F). The results (Table C-1, Appendix C) also reveal that the room air temperature has no effect on the power consumption of the fan.

Cooling effect

As expected, the cooling effect of the fans varied when the room air temperature and the velocity changed (Table C-1, Appendix C). For the tested conditions, the cooling effect of the Ceiling Fan (CF) varied between -3 and -0.5°C (-5.4 and -0.9°F), of the DF between -3 and -1.5°C (-5.4 and -2.7°F), of the SF between -2.5 and -1.5°C (-4.5 and -2.7°F) and of the TF between -2.5 and 0.5°C (-4.5 and -0.9°F). The cooling effect of the SF was least affected by the change in the experimental conditions.

The influence of the room air temperature on the cooling effect was analysed. The average of the cooling effect obtained with the four fans at the tested velocities was calculated for each of the room air temperatures. The results are compared in Figure 6. As expected, the cooling effect decreased significantly ($p<0.01$) with the increase of the room air temperature.

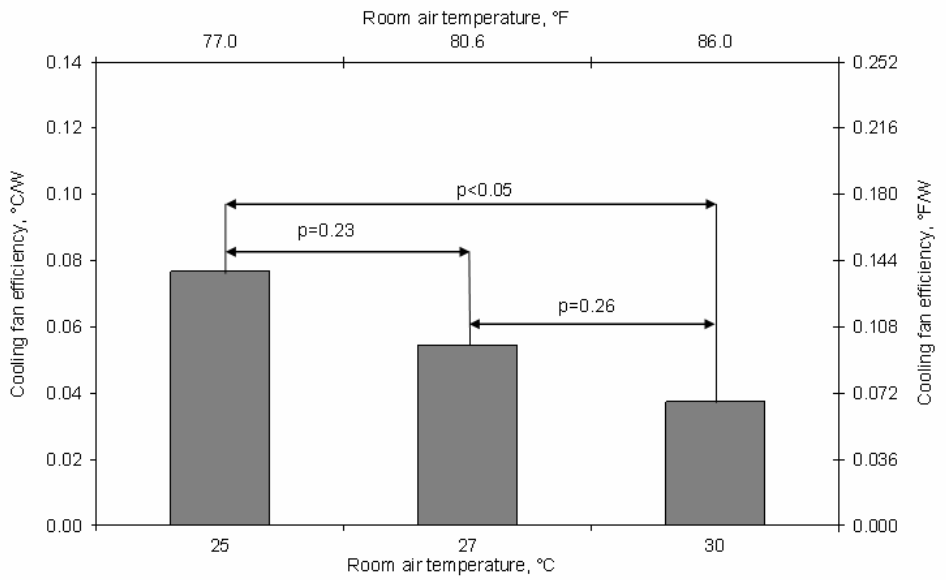

Figure 5. Averaged (over the type of cooling fan and velocity level) cooling fan efficiency for three room air temperatures.

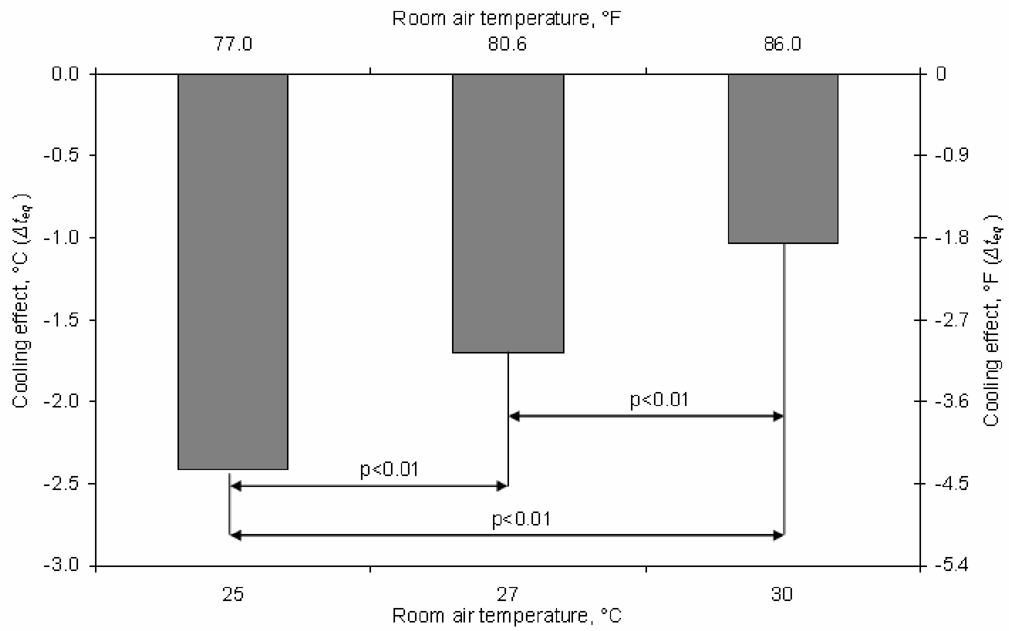

Figure 6. Averaged (over the type of cooling fan and velocity level) cooling effect for three room air temperatures.

The whole-body cooling effect (Δt_{eq}) discussed so far and reported in Appendix C is the weighted average of the cooling effect of each body segment. The cooling of the body segments depends on the local flow field generated by the fans. Analyses of the local cooling effect obtained by the tested fans for each body segment were performed. In the following, the results obtained at a room air temperature of 25°C (77°F) are shown and discussed because the conclusions were rather similar for the results obtained at 27°C (80.6°F) and 30°C (86°F). The local cooling effect of the four fans on each of the 22 body segments of the manikin is shown in Figure 7. The desk and standing fan had two velocity levels, while the ceiling and the tower fan had three levels. The cooling effect increases with the increase of the velocity level. However, the exposure to the airflow has much a stronger effect. The body segments exposed directly to the flow are cooled much more than for those in “shadow”. The impact of the velocity level on the cooling effect is greater for the exposed body segments than the segments in “shadow”. The cooling effect of the CF is quite symmetrical. The body segments that are exposed to the air movement generated by the fan (left and right front thigh, left and right face, back of the neck, right hand, left and right forearm, left and right chest) are cooler than the rest of the body. The air movement generated by the ceiling fan runs over the manikin from top-front. The desk fan provides a non-uniform cooling effect of the body. The airflow generated by the fan attacks the manikin’s body from the left (the fan is located only 0.66 m (26 in.) from the manikin). The coolest segments are those exposed to the flow generated by the fan, i.e. scull, left and right face, back of neck, left chest, left upper arm and left forearm. The local cooling provided by the SF has a pattern rather similar to the cooling provided by the DF. Similarly, the standing fan generates a non-uniform cooling effect. The rotation axis of the standing fan is located at 1.1 m (43 in.) above the floor, i.e. the highest velocities are generated at the manikin’s head level. Therefore, the coolest parts are the left and right face, the scull and the back of the neck. The lower segments of the manikin are slightly warmer with the SF in operation than in the reference case without fan (up to 1°C (1.8°F) warmer). The uncertainty in determination of the cooling effect (estimated to be 0.3°C (0.54°F), see Appendix B) alone cannot explain the difference. Complex airflow interaction in the vicinity of the body may be the reason. This needs to be further studied. The tower fan generated the most uniform cooling effect. The coolest parts are those on the lower-left, i.e. the left front thigh, the pelvis and the lower back. The cooling effect of the head is almost negligible.

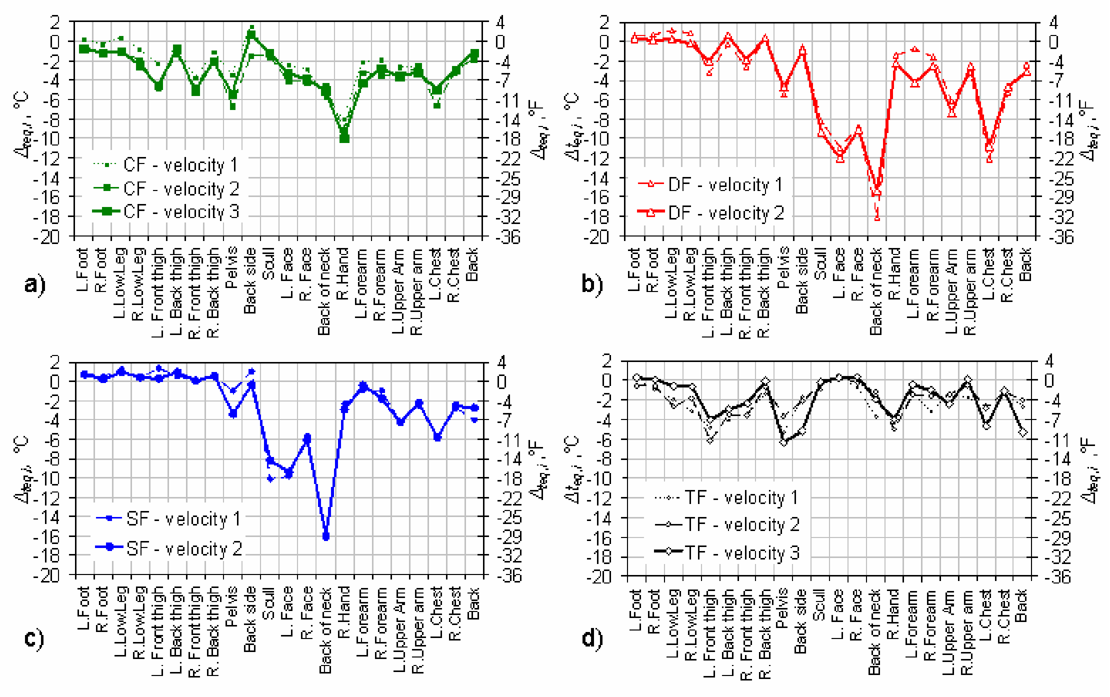

Figure 7. Change in manikin-based equivalent temperature ($\Delta_{teq,i}$) on each body part from the reference condition (room temperature equal to 25°C and no devices used for air movement) for

the a) ceiling fan (CF); b) desk fan (DF); c) standing fan (SF); and d) tower fan (TF). Step-change control of the fan velocity is possible. The $\Delta_{teq,i}$ has been calculated for the different velocity levels reported in Table 1.

The flow field generated by the fans was non-uniform and therefore caused non-uniform local cooling of the manikin's body. The asymmetric cooling on the body areas was investigated further. The average cooling effect for the upper body segments (right hand (left hand was broken), forearm (right and left), upper arm (right and left), chest (right and left), and back) and for the head (skull, face (right and left), back of neck) was determined. The total area of the upper body segments was 0.68 m^2 (7.32 ft^2), of the head it was 0.13 m^2 (1.40 ft^2) and of the whole-body it was 1.48 m^2 (15.93 ft^2). The results are compared in Figure 8.

The cooling effect of the ceiling fan was the most uniform. The difference in the whole-body cooling effect for the four types of fans is less than 2°C (3.6°F). The cooling effect of the upper body parts is always higher than the cooling effect of the head and the whole-body. The desk and the standing fan generate the largest non-uniformity in the local cooling effect. The head and the upper body parts are substantially cooler than the whole-body. The head is much cooler than the reference condition (11°C (19.8°F) for the DF and between 9°C (16.2°F) and 10°C (18°F) for the SF) and it is cooler than the whole-body (8°C (14.4°F) for the DF and between 7.5°C (13.5°F) and 8.5°C (15.3°F) for the SF). The tower fan causes a quite uniform but weak cooling of the body. The whole-body and the upper parts are cooler than the head (between 1 and 2°C (1.8 and 3.6°F) cooler). The velocity level does not affect significantly the whole-body cooling effect except for the ceiling fan. The impact of the velocity level on the cooling of the upper body part and the head is also smaller in comparison with the effect of exposure to the flow.

Figure 8. Cooling effect for the whole-body (22 body segments), the upper body part (12 body segments), and the head (4 body segments) for the ceiling, desk, standing and tower fan when the room temperature was set to 25°C.

In Figure 9 the whole-body cooling effect determined is plotted versus the fan power measured. The relative uncertainties are shown. The whole-body cooling effect of the desk fan and the ceiling fan is almost the same (around -2.5°C (-4.5°F)). However, the desk fan needs less than half of the electrical power used by the ceiling fan (around 20 W compared to 40 W). The DF and CF have a higher cooling effect than the TF and the SF. The SF has the lowest cooling effect, lower than -2°C (-3.6°F), with a fan power that varies in the range 35-40 W. For the TF an increase of the velocity level implies a slight reduction of the cooling effect with an increase of the needed power. Increasing the velocity level always implies an increase of the power requirement but this does not always cause a higher cooling effect. From the results shown in Figure 9 it can be concluded that changing the velocity level is not an effective way of controlling the cooling effect.

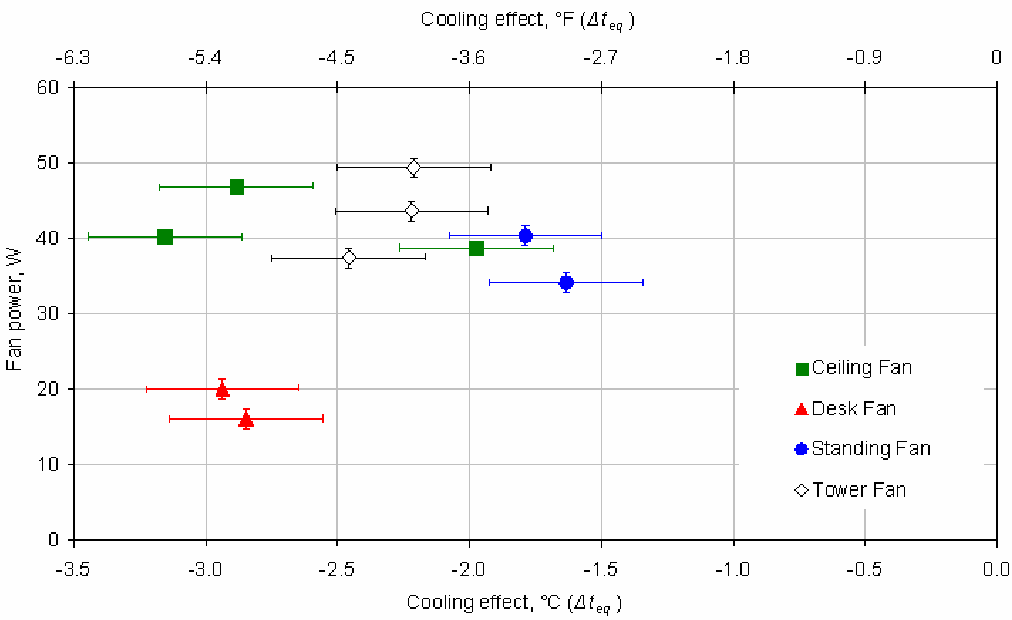

Figure 9. Cooling effect versus fan power for the ceiling, desk, standing and tower fan for the tested velocity levels when the room temperature was set to 25°C.

The air velocities measured at 0.2, 0.6, 1.1 and 1.7 m (8, 24, 43 and 67 in.) height above the floor with the CF and the TF are shown in Figure 10-a and with the DF and the SF in Figure 10-b. The air velocity field generated by the four fans is different. The CF generates downward airflow from the ceiling to the floor. The highest velocity (2.2 m/s (433 fpm)) is measured at the floor level. Therefore it may be expected that the generated flow will cool mostly the lower part of the manikin (legs and feet). This, however, is not seen from the results of the segmental cooling effect because the air velocity was measured while the manikin was moved away from the desk. Furthermore, the manikin was seated in front of the desk with its legs under the table far from the location of the velocity measurement. The blocking effect of the manikin's body and the interaction between the fan flow and the thermal plume generated by the thermal manikin may have had an impact on the cooling of the body segments. The TF also causes air movement mainly in the lower part of the room. The highest velocity (3.2 m/s (630 fpm)) was measured at the flow level. The desk and the standing fans generated similar air velocity profiles. In both cases the maximum air velocity (2.4 m/s (472 fpm) for the DF and 1.8 m/s (354 fpm) for the SF) was recorded at 1.1 m (43 in.) above the floor, i.e. the height of the manikin's head. The high velocity at the head level caused the strong non-uniform cooling of the body segment (Figure 8) as already discussed.

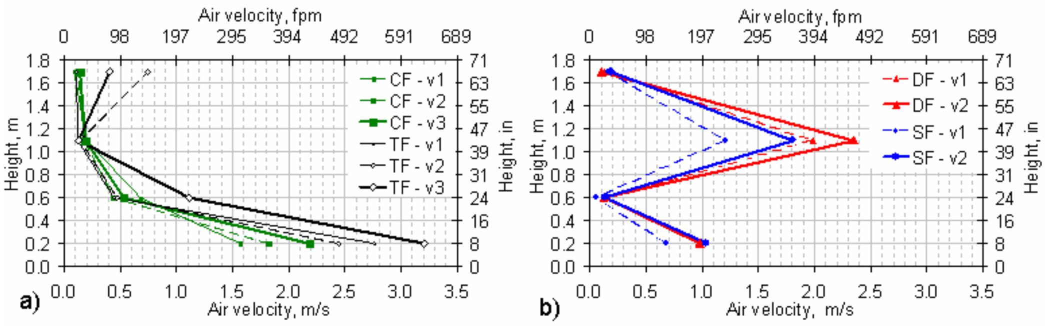

Figure 10. Air velocity measured at 0.2, 0.6, 1.1 and 1.7 m (8, 24, 43 and 67 in.) height above the floor where the manikin was located during the experiments for the ceiling and the tower fan (Figure 10-a) and for the desk and the standing fan (Figure 10-b) when the room temperature was set to 25°C.

DISCUSSION

Elevated air speed is widely used to provide comfort for occupants in warm environments. Cooling fans, e.g. ceiling fans, desk fans, etc., are used to generate air movement. It is accepted that energy saving can be achieved with this strategy, as opposed to air-conditioning of the whole building. Due to different design, installation and use the performance of cooling fans with regard to their cooling effect can be quite different. As the results of the present study show, at the same cooling effect the power consumption of different fans can be different as well. The cooling fan efficiency index introduced in the present study makes it possible for the first time to evaluate and compare cooling fans. This index combines in a single value the fan performance with regard to its cooling effect and its energy use. The experiments performed with four cooling fans of different design available on the market, i.e. ceiling, desk, tower and floor standing fans, document that the cooling fan efficiency index is sensitive in identifying differences in the performance of the cooling devices. The body cooling effect caused by the fans was different. The ceiling fan and the desk fan had a rather similar cooling effect which was substantially higher than the cooling effect of the floor standing fan and the tower fan. However, the electrical power used by the desk fan was twice as low as that used by the ceiling fan, and the desk fan therefore had a significantly higher cooling fan efficiency index than the remaining three fans. The index can be used by HVAC engineers and policymakers as well as for classifying fans according to their performance.

As already discussed, elevated air speed under individual control is recommended in the present indoor climate standards (ASHRAE 2004; ISO 2005; CEN 2007) for providing occupants with thermal comfort in warm environments. A relationship between the air speed and the upper operative temperature limits is provided in the standards. The recommended speed increase depends not only on the air temperature but also on the difference between mean radiant temperature and air temperature (t_a). When the mean radiant temperature is lower than the air temperature, elevated air speed is less effective for increasing the heat loss from the body. Conversely, elevated air velocity is more effective for increasing the body heat loss when the mean radiant temperature is higher than the air temperature. The relationship included in the standards is based on a theoretical calculation of the body cooling when it is exposed to uniform airflow. The relationship has been verified in human subject experiments performed under laboratory conditions when the air temperature is equal to the mean radiant temperature (Toftum et al. 2003). However, the validity of the relationship is not easily usable in practice when cooling fans are applied because, as the present results reveal, the body cooling by such fans is non-uniform due to large non-uniformity in the generated velocity field. The velocity field and its direction cannot be described with a single value. Therefore, it is not clear how to apply in practice the recommendations in the standards. Other methods for quantification of the cooling effect of air movement have been suggested as well (Szokolay 1998; Aynsley 2007). Aynsley (2007) proposed to use the SET* index (Gagge et al. 1971) since it includes the impact of humidity and the thermal insulation of clothing which are not considered in the relationship for elevated velocity included in the present standards. However, this approach has the same limitation, namely that there is no unique velocity which can describe the complex air velocity field generated by cooling fans. This is demonstrated with the following example, based on the data collected in the present study. The air velocity values of the four tested fans, measured when the mean radiant and air temperatures were equal to 27°C (80.6°F) and the velocity level was one, were used to calculate the SET* index. The measured relative humidity was equal to 26%, the clothing thermal insulation of the manikin was 0.62 clo (including the thermal insulation of the chair) and the activity level was 1.1 met. The results of the calculations are listed in Table 3. The SET* calculations were performed with ASHRAE's thermal comfort program (Fountain and Huizenga 1994). The SET* calculated with the velocities measured at different heights with each of the fans is substantially different (up to 4.1°C (7.38°F) in the case of the tower fan). The indoor climate standards specify using measurements at 0.6 m (24 in.) height (sedentary person) in order to predict occupants' thermal comfort (PMV-PPD index, etc.). At this height the SET* values for the desk and standing fans are almost null (because the desk shades the occupant's body at that height), but their whole-body cooling effects measured with the thermal manikin (Table C-1, Appendix C) are strong (varied between -1.4 and -2.9°C (-2.5 and -5.2°F)). It is clear that the approach recommended in the present standards, as well as the SET*, cannot be used directly in practice. This issue needs to be carefully considered and addressed in the standards.

Table 3. SET* Calculated with the Results Obtained in the Present Study

Measuring height		Air velocity		Type of Fan	SET*		Cooling effect	
m	in.	m/s	fpm		°C	°F	°C	°F
0.2	8	1.35	266	Ceiling fan	23.4	74.1	3.1	5.6
0.6	24	0.32	63	Ceiling fan	25.5	77.9	1	1.8
1.1	43	0.14	28	Ceiling fan	26.5	79.7	0	0
1.7	67	0.13	26	Ceiling fan	26.5	79.7	0	0
0.2	8	0.74	146	Desk fan	24.3	75.7	2.2	4.0
0.6	24	0.1	20	Desk fan	26.5	79.7	0	0
1.1	43	1.76	346	Desk fan	23	73.4	3.5	6.3
1.7	67	0.11	22	Desk fan	26.5	79.7	0	0
0.2	8	1.27	250	Standing fan	23.5	74.3	3	5.4
0.6	24	0.18	35	Standing fan	26.4	79.5	0.1	0.2
1.1	43	1.77	348	Standing fan	23	73.4	3.5	6.3
1.7	67	0.12	24	Standing fan	26.5	79.7	0	0
0.2	8	3.27	644	Tower fan	22.4	72.3	4.1	7.4
0.6	24	0.77	152	Tower fan	24.2	75.6	2.3	4.1
1.1	43	0.27	53	Tower fan	25.7	78.3	0.8	1.4
1.7	67	0.12	24	Tower fan	26.5	79.7	0	0
		0.15		**none**	**26.5**	**79.7**		

The desk fan was found to have the highest efficiency index of the four tested fans (Figures 3 and 4, Table C-1, Appendix C). The whole-body cooling effect of this fan was largest. The non-uniformity of the local cooling effect of this fan was also greatest, with the head region being mostly cooled. It may be suggested to use the head cooling effect together with the cooling fan efficiency index when assessing the performance of cooling fans because the head is an active heat dissipater and in warm environments the whole-body thermal sensation follows the head region thermal sensation closely (Melikov et al. 2004 a and b; Arens et al. 2006; Watanabe et al. 2008). Thus, at the same efficiency, the performance of the fan that provides greater cooling of the head may be considered to be better. However, these selection criteria may fail to be correct in practice because human response to airflow from the front and from the back is different.

In this study, the cooling effect of air movement has been quantified by measuring the dry heat loss. The evaporative heat loss has not been taken into account because the thermal manikin used cannot sweat.

Several studies have used dry heat loss measured by a thermal manikin to quantify the cooling effects of air movement on the human body. Tsuzuki et al. (1999) studied the performance of three designs of task ambient air-conditioning systems and found that the cooling effect of the combined evaporative and sensible cooling may double the total whole-body cooling rate due to dry heat loss alone when 20% of the surface was wet. The cooling effect of the evaporative heat loss will increase with the increase of the room temperature. In the future, the determination of fan efficiency can be made more accurately by sweating thermal manikins. The sweat glands are not uniformly distributed over the human body. Therefore, use of the thermal manikins available today with simulated sweat glands on the surface areas corresponding to the site of the human skin where they are most dense can be considered.

A considerable number of studies focused on the use of fans to cool people in a warm environment (McIntyre 1978; Rohles et al. 1983; Jones et al. 1986; Tanabe and Kimura 1987; Scheatzle et al. 1989; Bauman et al. 1993; Melikov et al. 1994 a and b; Fountain et al. 1994; Arens et al. 1998; Szokolay 1998; Tsuzuki et al. 1999; Khedari et al. 2000; Hayashi et al. 2004; Sekhar et al. 2005; Aynsley 2005 and 2007; Atthajariyakul and Lertsatittanakorn 2008; Sun et al. 2007 and 2008; Watanabe et al. 2008). Only in one study was the fan power reported (Sun et al 2008). The power consumption of cooling fans is considered negligible (usually less then 90 W) and therefore it is not reported. However, as already discussed, it has been demonstrated that the required power input of cooling fans is a critical factor for an energy-saving strategy used in warm environment (Schiavon and Melikov 2008). Based on comprehensive simulations as well as on defined outdoor conditions and building characteristics, it has been shown that in some buildings the use of cooling fans with power input of more than 20 W will actually increase the energy consumption compared to the energy consumption needed to cool the whole building. For the same cooling effect the power input of the desk fan tested in the present study was 16-20 W, i.e. twice as low as the power input of the ceiling fan (approx. 40 W) and therefore its cooling fan efficiency index was twice as high. Nevertheless, one should be cautious when recommending the use of the desk fan instead of the ceiling fan. The ceiling fan may provide cooling to several occupants while the desk fan provides cooling to only one occupant. Individual control with a ceiling fan is difficult to achieve in practice when it aims to provide cooling to several occupants who may have different preferences with regard to the air movement. The development of desk fans with a strong cooling capacity and low energy consumption of a few watts, as for example the fans used by Watanabe et al. (2008) and Sun et al. (2008), is recommended.

The convection heat loss from the body with cooling fans is mainly based on the velocity and the turbulence intensity of the generated flow. As discussed, the fans tested in the present study generated a non-uniform flow. The velocity distribution at the location of the thermal manikin was rather different as well. The CF and the TF generated flow with the highest velocity near the floor, up to 0.6 m (24 in.) above the floor, while the highest velocity generated by the DF and the SF was measured at the head region. The indoor climate standards recommend individual control of the airflow at elevated velocity. Velocity control at two or three levels was provided for the fans tested. The control, however, affected the flow mostly in the high velocity region, i.e. near the floor for the CF and the TF and at the head region for the DF and the SF, and therefore resulted mainly in an increase of the local cooling of the body segments exposed to the flow and affected only slightly the whole-body cooling (Figure 8). In this respect the layout, furniture arrangement, etc. were also factors affecting the local air distribution around the manikin’s body.

The cooling fan efficiency index of the four cooling fans was determined under well defined conditions, based on assumptions of their use in practice. The clothing insulation and its distribution over the manikin’s body (naked/covered body area ratio), the relative distance and direction between the fan and the manikin, metabolic rate, mean radiant and air temperatures were not changed and the latent heat loss was not simulated. Different results would be obtained if one or more of these parameters were changed. For example, the cooling of the ceiling fan would be different when the layout in the room was changed, e.g. the desk with the manikin was moved to another location in the room. The importance of these parameters for the cooling fan efficiency should be studied. A standard procedure for testing the index should be developed considering also other factors, such as number of occupants who can benefit from one cooling fan, maximum non-uniformity of body cooling which will be acceptable for the occupants, maximum velocity limitations to avoid blowing of paper, non-thermal discomfort such as eye blinking, etc. Fan cooling effect and efficiency determined with standard methods and used as product characteristics will allow designers to make the optimal selection for each practical application.

CONCLUSIONS

A new index, named "cooling fan efficiency index" defined as the ratio between the cooling effect of the used device and its power consumption has been introduced for evaluation of the performance of cooling fans.

The measurements performed with ceiling fan, desk fan, standing fan and tower fan in a real office at three room air temperatures and different fan velocity levels revealed that the index is sensitive enough to identify differences in the performance of the cooling devices. The results identify a large variation in the whole-body cooling effect (between -3.2 and -0.4°C (-5.76 and -0.72°F)), in the fan power (between 15.6 and 49.3 W), and in the cooling fan efficiency index (between 0.009 and 0.177°C/W (0.016 and 0.319°F/W)). The local cooling effect for body segments caused by the fans was strongly non-uniform. The desk fan had a significantly ($p<0.01$) higher efficiency than the other three fans tested.

The cooling fans generate a non-uniform velocity field around occupants which cannot be described with a single value. This makes the recommendation in the standards for elevated velocity in warm environments difficult to use in practice. The present thermal comfort standards need to be revised to better address the issue of thermal comfort in warm environments.

A standard method for testing the performance of cooling fans with regard to their cooling effect and power input needs to be developed.

Symbols and units	
CF	ceiling fan
CFE	cooling fan efficiency index, °C/W
DF	desk fan
$h_{cal,i}$	dry heat transfer coefficient of *i*-th segment of the manikin, determined during calibration, W/m^2°C
Pf	fan power, W
$Q_{t,i}$	sensible heat loss of *i*-th segment, W/m^2
SF	standing fan
TF	tower fan
Δt_{eq}	whole-body cooling effect, °C
$\Delta t_{eq,i}$	segmental cooling effect, °C
t_a	ambient air temperature or room air temperature, °C
t_{eq}	whole-body manikin based equivalent temperature, °C
$t_{eq,i}$	segmental equivalent temperature, °C
$t_{sk,i}$	skin temperature of *i*-th segment of the manikin, °C

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APPENDIX A

The body of the thermal manikin consists of 23 independently controlled segments. The thermal manikin measures the power consumption or heat loss, Q_t (in W/m^2), and the surface temperature, t_{sk} (in°C). The dry heat transfer coefficient, $h_{cal,i}$ for each body segment and for the whole-body were obtained from calibration of the manikin in an indoor environmental chamber with a uniform thermal environment, i.e. air temperature equal to the mean radiant temperature and air velocity lower than 0.06 m/s (12 fpm). The calibration was performed at three air temperatures, 24, 27 and 31°C (75.2, 80.6 and 87.8°F). During these experiments the left hand of the manikin was broken; therefore it is not included in the measurements and the calculations. The name and the body surface area of the manikin's body segment are listed in Table A-1.

Table A-1. Surface Area of Manikin's Body Segments

Number	Body part	Area of the body part m^2	Area of the body part ft^2
1	Left Foot	0.043	0.463
2	Right Foot	0.043	0.463
3	Left Lower Leg	0.09	0.969
4	Left Lower Leg	0.09	0.969
5	Left Front Thigh	0.08	0.861
6	Left Back Thigh	0.08	0.861
7	Right Front Thigh	0.083	0.893
8	Right Back Thigh	0.083	0.893
9	Pelvis	0.055	0.592
10	Back side	0.11	1.184
11	Scull	0.05	0.538
12	Left Face	0.0258	0.2777
13	Right Face	0.0258	0.2777
14	Back of Neck	0.0248	0.2670
15	Left Hand	0.038	0.409
16	Right Hand	0.037	0.398
17	Left Forearm	0.05	0.538
18	Right Forearm	0.05	0.538
19	Left Upper Arm	0.073	0.786
20	Right Upper arm	0.078	0.840
21	Left Chest	0.07	0.753
22	Right Chest	0.07	0.753
23	Back	0.13	1.399
Tot Area [m^2]		**1.48**	**15.92**

APPENDIX B
Uncertainty of the measurement

The data were analyzed in accordance with the ISO guideline (1993) for the expression of uncertainty. The sample standard uncertainty, U, was calculated as the combination of the maximum uncertainty of measurement (random error), U_{meas}, and the uncertainty of the instrument (calibration), U_{inst}. Table B-1 summarizes the typical values of absolute uncertainty based on the analyses of measurements. The values are given for each uncertainty component together with the sample uncertainty U and the uncertainty of a derived quantity U_c. The instrument uncertainty U_{instr} was the strongest component in the case of manikin-based equivalent temperature and air temperature. The uncertainty of process stability was the strongest component in the case of fan power. When presented, the uncertainty is indicated by means of error bars. The level of confidence is 95% (coverage factor of 2).

Table B-1. The Sample Uncertainty, *U*, and the Uncertainty of a Derived Quantity, U_c, are Reported with a Level of Confidence of 95%

Quantity	U_{meas}	U_{instr}	U	U_c
Manikin- based equivalent temperature, t_{eq}	< 0.05°C, 60 readings	0.2°C	0.21°C	Δt_{eq}: 0.3°C *CFE*: 0.009°C/W
Air temperature, t_a	-	0.1°C	0.1°C	
Fan power, P_f	1 W	0.5 W[a]	1.3 W	*CFE*: 0.009°C/W
Air velocity	-	See note[b]	See note[b]	
Relative humidity		See note[c]	See note[c]	

[a] The fan power (P_f) was measured with an accuracy of ±0.5% of the full scale (100W). The instrument was in class 0.5 according to IEC (1980).
[b] 0.02±1% of the readings for velocity range between 0.05 and 1 m/s, and accuracy of ±3% of the readings for velocity range between 1 and 5 m/s.
[c] ±2% of the readings for the relative humidity range from 0% to 90%, and ±3% for the range from 90 to 100%.

APPENDIX C

The measured cooling effect and fan power and the determined cooling fan efficiency index for the experimental conditions tested.

Table C-1. Whole-body Cooling Effect, Fan Power and Cooling Fan Efficiency Index for the Four Cooling Fans (Ceiling Fan, Desk Fan, Standing Fan and Tower Fan) and for the Three Room Temperature Levels (t_a = 25, 27, 30°C (77, 80.6 and 86°F).

Type of fan	Velocity level	Room air temperature		Cooling effect (Δt_{eq})		Fan power, W (P_f)	Cooling fan efficiency (*CFE*)	
		°C	°F	°C	°F		°C/W	°C/W
CF	1	25	77	-2	-3.6	38.7	0.051	0.092
CF	2	25	77	-3.2	-5.8	40.2	0.079	0.142
CF	3	25	77	-2.9	-5.2	46.8	0.062	0.112
CF	1	27	80.6	-0.9	-1.6	38.7	0.023	0.041
CF	2	27	80.6	-2.3	-4.1	39.8	0.057	0.103
CF	3	27	80.6	-2	-3.6	44.9	0.045	0.081
CF	1	30	86	-0.7	-1.3	37.2	0.02	0.036
CF	2	30	86	-0.7	-1.3	38.8	0.018	0.032
CF	3	30	86	-1.1	-2.0	47.6	0.022	0.040
DF	1	25	77	-2.8	-5.0	16.1	0.177	0.319
DF	2	25	77	-2.9	-5.2	20.1	0.146	0.263
DF	1	27	80.6	-1.8	-3.2	16	0.115	0.207
DF	2	27	80.6	-2	-3.6	19.5	0.104	0.187
DF	1	30	86	-1.5	-2.7	15.6	0.095	0.171
DF	2	30	86	-1.9	-3.4	19.3	0.099	0.178
SF	1	25	77	-1.6	-2.9	34.1	0.048	0.086
SF	2	25	77	-1.8	-3.2	40.3	0.044	0.079
SF	1	27	80.6	-1.9	-3.4	34.2	0.055	0.099
SF	2	27	80.6	-2.3	-4.1	39.5	0.058	0.104
SF	1	30	86	-1.6	-2.9	33.5	0.047	0.085
SF	2	30	86	-1.4	-2.5	37.6	0.038	0.068
TF	1	25	77	-2.5	-4.5	37.4	0.066	0.119
TF	2	25	77	-2.2	-4.0	43.6	0.051	0.092
TF	3	25	77	-2.2	-4.0	49.3	0.045	0.081
TF	1	27	80.6	-0.9	-1.6	37.3	0.024	0.043
TF	2	27	80.6	-1.4	-2.5	42.2	0.033	0.059
TF	3	27	80.6	-1.4	-2.5	48.9	0.029	0.052
TF	1	30	86	-0.5	-0.9	38	0.012	0.022
TF	2	30	86	-0.4	-0.7	42.5	0.009	0.016
TF	3	30	86	-0.6	-1.1	46	0.014	0.025

PAPER D

Schiavon, S., and Melikov, A. 2008. Energy saving and improved comfort by increasing air movement. *Proceedings of International Conference Indoor Air 2008*. Copenhagen, Denmark.

Energy saving and improved comfort by increased air movement

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SUMMARY

In this study the potential saving of cooling energy by elevated air speed which can offset the impact of increased room air temperature on occupants' comfort as recommended in the present standards (ASHRAE 55 2004, ISO 7730 2005 and EN 15251 2007) was quantified, by means of simulations with EnergyPlus software. Fifty-four cases comprising six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem, Athens), three indoor environment categories - I, II and III (according to standard EN 15251 2007) and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated. The required cooling/heating energy was calculated assuming a perfectly efficient HVAC system. A cooling energy saving between 17 and 48% and a reduction of the maximum cooling power in the range of 10-28% has been obtained. The results reveal that the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature. Energy saving may not be achieved with the widely used today methods for air speed increase, such as ceiling, standing, tower and desk fans.

KEYWORDS

Energy savings, Indoor air movement, Personalized ventilation, EN 15251 and ASHRAE 55, Fan

INTRODUCTION

Air speed increase at room temperature elevated above the range of comfortable room temperature is recommended in the standards (ASHRAE 55-2004, ISO 7730- 2005, EN 15251-2007) in order to maintain the heat exchange between human body and environment needed for occupants' thermal comfort. The relationship between the upper operative temperature limits and the air velocity is shown in Figure 1. The figure is based on a theoretical calculation when the whole human body is exposed to air movement. However the neutral curve (air temperature, t_a, equals to mean radiant temperature, t_r) has been verified in human subject experiments (Toftum et al., 2003). It was also shown that the requirement of personal control over the air speed is essential for its acceptance. Therefore, it may not be appropriate to offset a temperature increase by increasing the air speed within a centrally-controlled air system (Olesen and Parsons, 2002).

The possibility of increasing the upper operative temperature limit may reduce the energy consumption without significantly affecting occupants' thermal comfort. The individual control of air movement can be achieved with personalized ventilation systems, task/ambient systems, desk, standing, tower or ceiling fans, and under some conditions with operable windows.

The energy consumption of these methods for air movement generation is different. The purpose of this study is to quantify, by means of simulations, the potential savings of energy need for cooling achieved by elevated air without reducing the comfort conditions is presented. The required energy for the control of humidity and the efficiency of the HVAC system is not taken into account.

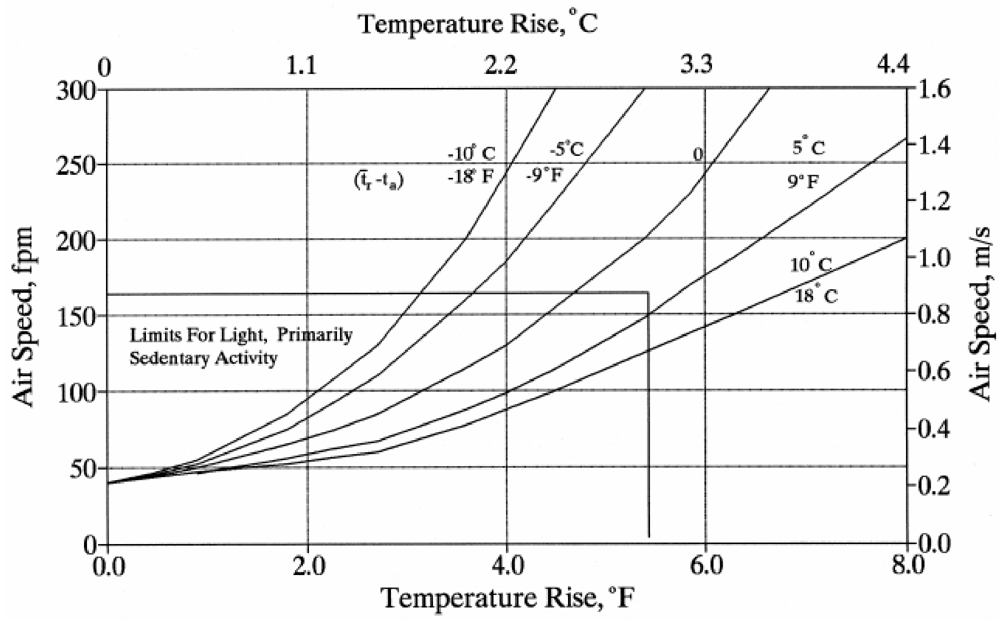

Figure 1. Air speed required to offset increased temperature. (Figure 5.2.3 from ASHRAE 55 2004).

METHODS

Building locations and weather data

The energy simulations were performed for an identical single office room sited in six European and Mediterranean cities. The Cooling Degree Days, CDD, (ASHRAE 2005) with a base temperature of 18°C were used as an indicator of the intensity of the summer period. The cities are: Helsinki (Finland, CDD=33), Berlin (Germany, CDD = 170), Bordeaux (France, CDD =263), Rome (Italy, CDD = 508), Jerusalem (Israel, CDD = 647), Athens (Greece, CDD = 1076). Thus it was possible to describe in a homogeneous way different climate conditions. The focus was on summer conditions. The ASHRAE IWEC Weather Files are used as input data in the simulation model.

Description of the office room

The single office room has floor surface area of 4 m by 2.5 m. The room height is 3 m. The external walls are constructed with 20 mm of plaster, 100 mm of glasswool, 240 mm of brick and 10 mm of internal plaster. The window is composed by an external low-emissivity glass pane (thickness of 6 mm), 13 mm of air and an internal glass pane (thickness of 6 mm). It has a U-value equal to 1.72 $WK^{-1}m^{-2}$ and g-factor or Solar Heat Gain Coefficient equal to 0.56. It has a total area of 2.4 m^2 (24% of the floor area, height of 1.2 m and width of 2 m). The window is facing south. An external shading device is present. It has a shading coefficient of 0.48 (g-factor equal to 0.43). It is activated when the total irradiance on the windows is higher than 400 W/m^2. The internal walls, floor and ceiling are adiabatic. The effect of thermal mass is taken into account.

Internal temperature and ventilation and infiltration rate

The thermal comfort conditions and ventilation specifications were chosen in order to guarantee the values defined in EN 15251 (2007) for the category I, II and III of indoor environment in the room during occupation time. From 7:00 am till 6 pm the heating and cooling system keeps the internal operative temperature inside a range limited between the minimum operative temperature bellow which heating is required (Min t_{op} for heating) and the maximum operative temperature above which cooling is required (Max t_{op} for cooling). The minimum and maximum operative temperatures are shown in Table 1. During weekends and nighttime the temperature set-back is 12°C in winter and 40°C in summer. The design ventilation rates are shown in Table 1. The design airflow rate is supplied during occupation hours. The air flow rates during un-occupied time are 7% of the design values, i.e. from 0.06 to 0.14 l/ sm^2 (the standard suggests a minimum airflow rate for unoccupied hours in the range 0.1 to 0.2 l/sm^2). The infiltration is considered null.

Internal heat gains, occupancy and description of HVAC system

One occupant is present in the room (10 m^2 per person). She/he contributes to both sensible and latent heat loads. The activity level of the occupant is 1.2 met (1 met = 58.15 W/m^2), therefore the total heat produced per occupant is around 125 W. The balance between sensible and latent heat is calculated by the used software. The occupant is present in the room from Monday to Friday, from 9:00 to 18:00 with an hour break at noon. Saturday and Sunday are holidays. No public holidays are assumed. The heat load due to office equipment is 5.4 W/m^2. According to ASHRAE (2005), this value corresponds to a “light load office”. The loads follow the schedules of the occupant. The lighting load is 6 W/m^2, common value used in practice for an office. The lighting load is at 90% of its capacity from 9:00 to 10:59, at 70% from 11:00 to 12:59 and from 14:00 to 15:59, at 100% from 16:00 to 17:59. In the other hours the light is switched off. The needed energy is calculated assuming a perfectly efficient HVAC system. The airflow network and the heating and cooling plants were not modelled, therefore the needed airflow is supplied at outdoor conditions. The humidity level is monitored but not controlled.

Simulated cases

From Figure 1, assuming that air temperature is equal to mean radiant temperature, it is obtained that the allowed increase in operative temperature is equal to 1.7°C for an air flow of 0.5 m/s and 2.5°C for an air flow of 0.8 m/s. Those values are added to the maximum summer operative temperatures for the three categories as specified in EN 15251 (2007). The values shown in Figure 1 were obtained for a comfort limit of 26°C, which is the comfortable temperature limit for category II in EN 15251 (2007). It is reasonable to assume that the same increments in operative temperature can be applied for the comfortable temperature limits for categories I and III, i.e. 25.5°C and 27°C. In total fifty-four cases, comprising six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem, Athens), three indoor environment categories - I, II and III and three air velocities (<0.2, 0.5 and 0.8 m/s) as listed in Table 1, are simulated The summer design day simulation was performed for each city and each indoor environment category in order to calculate the maximum power needed for providing the comfort conditions. The maximum power is used to size the chiller. The summer design day conditions were taken from ASHRAE (2005).

Table 1. Simulated cases: category of indoor environment, air flow rates, minimum and maximum operative temperatures. The maximum operative temperatures for cooling are increased according to the local air velocity.

Category according EN 15251 2007	Air flow per pers. [$ls^{-1}pers^{-1}$]	Air flow per floor area[§] [$ls^{-1}m^{-2}$]	Min t_p for heating [°C]	Velocity [m/s]	Temperat. increase [K]	Max t_{op} for cooling [°C]
I	10	1	21	<0.2	0	25.5
				0.5	1.7	27.2
				0.8	2.5	28
II	7	0.7	20	<0.2	0	26
				0.5	1.7	27.7
				0.8	2.5	28.5
III	4	0.4	19	<0.2	0	27
				0.5	1.7	28.7
				0.8	2.5	29.5

§ Recommended values from EN 15251 2007 for low polluting buildings (see Annex C).

Simulation software

A robust building energy simulation program, EnergyPlus, was used for the simulations. This software allows for performing simulations of the building and the HVAC system as a whole. It calculates the thermal loads to be satisfied and defines the system strategy needed to fulfil the required comfort conditions. In the present research, EnergyPlus is mainly used in order to predict energy consumption needed for keeping the room operative temperature within the comfort limits (specified in Table 1).

RESULTS

The energy need for cooling of the room (EN 15615, 2007) when located in each of the selected six cities in order to comply with the three categories (Table 1) at the three levels of velocity (0.2 m/s, 0.5 m/s, and 0.8 m/s) and the corresponding operative temperatures (Table 1) are listed in Table . The energy need for cooling is the annual amount of cooling energy needed to keep the operative temperature below the maximum summer operative temperature limit. The cooling energy for the control of humidity and the energy losses in the system are not included.

The fan operation total hours are shown in Table 2 as well. It is supposed that, when the indoor operative temperature is higher than the maximum operative temperature limit the occupant switches on the fan. Thus the fan operation hours were calculated as the sum of hours during which operative temperature was higher than the maximum operative temperature limit and the occupant was in the room, e.g. an hour is counted if the occupant is in the room and the room operative temperature is above 25.5°C for category I, or it is above 26°C for category II, or above 27°C for category III. The total number of hours that the fan is operating is proportional to the energy consumption of the fan. In Table 2 the ratio between the fan operation hours and the total yearly occupant working hours is reported. The total occupant working hours per year (260 working days) are 2080 h.

The maximum cooling power per unit of floor area has been calculated but not reported due to space limitation. The results show that a reduction of the maximum cooling power due to the increase of air movement is in the range of 8-22%. It is higher for the air velocity equal to 0.8 m/s, for the cold climates and for higher quality of indoor environment. As a consequence, smaller chillers may be installed; this will lead to a reduction of the initial (investment) costs.

Table 2. Energy need for cooling per unit of floor and fan operating hours at the three velocity levels for the three categories of indoor environment when the room is located in the six cities with different outdoor climatic conditions. The energy saved due to increase of air velocity (or relative increase of upper operative temperature limits) is listed.

		Velocity < 0.2 m/s Reference case	Velocity = 0.5 m/s				Velocity = 0.8 m/s			
			Energy		Fan		Energy		Fan	
City	C. #	Energy Need§	Energy Need§	Saved°	Hours*	Perc+	Energy Need§	Saved°	Hours*	Perc+
Hel-sinki	I	18	12	34%	636	31%	9	48%	645	31%
	II	21	15	29%	765	37%	12	41%	788	38%
	III	24	18	24%	859	41%	16	35%	867	42%
Berlin	I	24	16	32%	814	31%	13	45%	826	31%
	II	26	19	28%	848	37%	16	40%	864	38%
	III	27	21	23%	907	41%	18	34%	916	42%
Bor-deaux	I	39	28	27%	1080	52%	24	38%	1091	52%
	II	41	31	24%	1184	57%	27	34%	1204	58%
	III	42	33	21%	1345	65%	29	31%	1368	66%
Rome	I	52	40	23%	1300	63%	35	33%	1308	63%
	II	53	42	21%	1406	68%	37	30%	1420	68%
	III	53	43	19%	1499	72%	38	27%	1509	73%
Jeru-salem	I	65	51	21%	1483	71%	45	30%	1491	72%
	II	66	52	20%	1722	83%	47	29%	1746	84%
	III	66	54	19%	1909	92%	48	27%	1928	93%
Athen s	I	75	61	18%	1419	68%	56	25%	1439	69%
	II	74	61	17%	1555	75%	56	25%	1579	76%
	III	73	61	17%	1888	91%	55	24%	1921	92%

C. = Category according EN 15251 2007.
§ Energy Need = Energy need for cooling [$kWhm^{-2}y^{-1}$].
° Saved. = Percentage of the saved energy need for cooling compared to the reference case.
* Fan hours = Annual number of hours that the fan for increasing the air velocity is operating.
+ Perc. = Annual number of hours that the fan is operating over yearly occupant working hours.

DISCUSSION

The energy need for cooling for the fifty-four simulated cases is summarized in Table 2. In all simulated cases increasing the air velocity implied a reduction of the energy consumption. Saving of energy need for cooling between 17 and 48% is obtained. The highest percentage of energy savings has been obtained in Helsinki for category I of the indoor environment. The lowest percentage of energy savings has been obtained in Athens for category III of the indoor environment. The percentage of savings decreases when the quality of the indoor environment category decrease, e.g. in Bordeaux for category I the saving is 27% and for category III it is 21%. The percentage of savings decreases with the increase of the cooling degree days (defined in section “Building location and weather data”). The percentage of savings increases when the air velocity increase, in fact, the higher savings have been obtained for the air velocity equal to 0.8 m/s. Those considerations can be drawn from Figure 2. In Helsinki, Berlin and Bordeaux, the energy needs for cooling increase with the reduction of the quality of indoor environment due to the free cooling effect of the outdoor air.

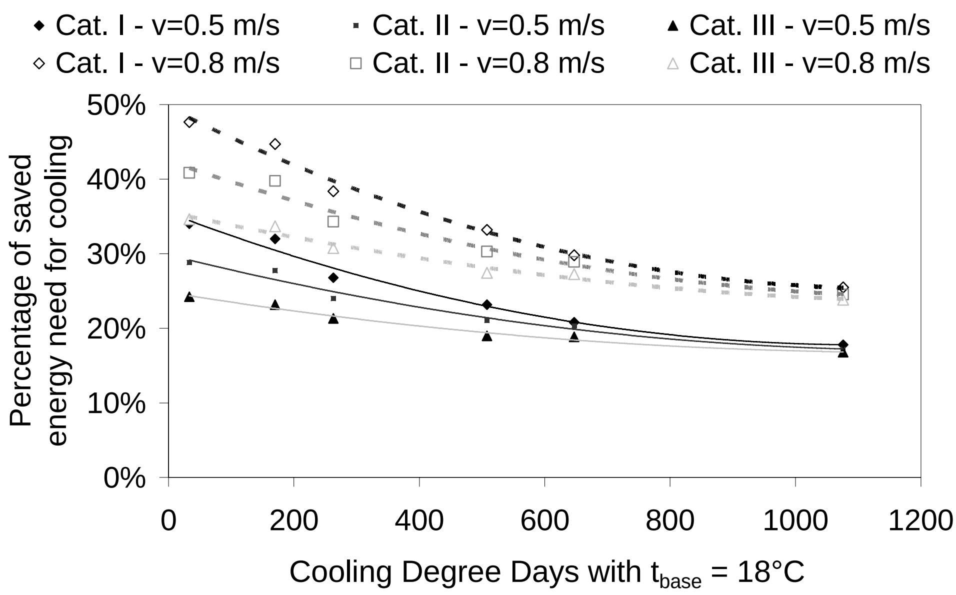

Figure 2. Percentage of saved energy need for cooling vs. cooling degree days. The points are the values obtained from the simulations. The lines are second order polynomial interpolations of the calculated data. The reference case for each category is the cooling energy need to obtain the thermal environment at each category without any increase in air velocity.

The fan operation hours are listed in Table 2. The fan operation hours increase with the increase of the number of cooling degree days and with the reduction of the indoor environment category. The fan operation hours are almost independent of the increase of air velocity. In Table 2 the ratio between the fan operating hours and yearly occupant working hours is shown. The ratio varies between 31 to 93%. High values of the ratio means that the fan would work also during winter time, when it is supposed that people would be dressed with a clothing insulation equal to 1 clo. In this case the graph, as shown in Figure 1, can not be applied. However the fan is working during winter-time in warm climates, Jerusalem and Athens, where probably the occupant would have lighter clothing. During these calculations the relative humidity was not controlled by the HVAC system but it was monitored. The relative humidity requirements were almost always fulfilled during occupation hours.

Energy consumption of the fan

The air movement increase can be produced by ceiling fans (common nameplate power consumptions around 70W), standing fans (50W), tower fans (40W), desk fans (30W), personal ventilation systems and under some conditions with operable windows. In order to check whether the electrical consumption of the fan is a critical factor for the performed energy saving calculations, the differences between the saved electrical energy for running the chiller and the electrical energy consumed by the fan have to be calculated. This difference is hereafter named “net electrical energy saved”. The saved electrical energy can be obtained by the saved energy need for cooling (Table 2) considering the energy losses from emission, distribution and storage and the Coefficient Of Performance (COP) of the chiller. Those values depend on the type of cooling system used and on the building characteristics. In this analysis has been assumed that the COP is equals to 3.5 and the cooling energy increase due to losses is 15%.

The net electrical energy saved has been calculated for several fan input powers (in the range of 2-70 W). The net electrical energy saved depends on the simulated case (location, indoor environment category, air velocity increase), on the COP and percentage of losses chosen, and on the fan power input. It was calculated using the data shown in Table 2. These results are not reported in this paper. However an easy-to-use graph for checking, as a rule of thumb, how much energy can be saved as a function of the fan input power is shown in Figure 3. For the cases where the velocity was equal to 0.5 m/s the net electrical energy saved has been calculated as function of the fan input power. The maximum and minimum values for each fan input power has been plotted in Figure 3. The use of the graph is explained in the following example. If the input power of the fan is 10 W, then the expected net electrical energy saved is at minimum 1.1 $kWhm^{-2}y^{-1}$ and at maximum 3 $kWhm^{-2}y^{-1}$. The main advantage is that the graph is independent of the location and the indoor environment category and so it can give a first estimation of the saving. For example if the fan power input is 60 W, then it can be easily seen that there is not energy savings. From the figure it can be deduced that traditional systems, as ceiling fans and standing fans can not be used to save energy when the assumptions made in this study are fulfilled. It is needed to use a system that would require a power input lower than 20W. This can be done using a desk fan or a personal ventilation system.

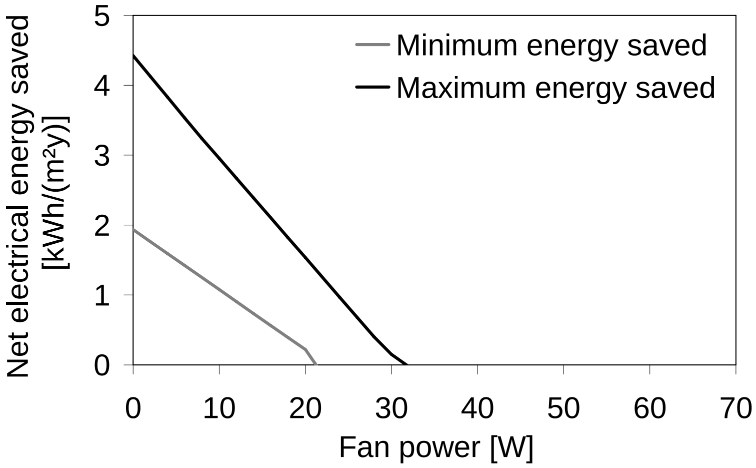

Figure 3. Total electrical energy that can be saved versus fan input power when the COP is equal to 3.5 and the energy losses are equal to 15%.

The results in Figure 1 were obtained and verified with an air flow over the whole body. Personal ventilation systems or desk fans are able to cool only the upper part of the body. Nevertheless the authors believe that the difference would not be significant, because most of the heat loss occur in the higher part of the body (the head is strong dissipater of heat). Another advantage of the personal ventilation system is that it will increase the inhaled air quality and this will improve occupants' health and productivity (Melikov, 2004).

Limitations of the study

The HVAC system has not been modelled, therefore the interaction between the building and the system can not be predicted. The moisture control has not been modelled. Those simplifications may change the range of saved energy need for cooling. Sensitivity analyses for internal and external heat loads and behaviour of the occupant have not been performed. The values of saved net electrical energy were obtained under the assumptions that COP = 3.5 and 15% increment of cooling energy due to losses (emission, distribution and storage). A higher COP would lead to lower total electrical energy saved. If the system is properly design the energy loss will be lower and this will lead to lower total electrical energy saving.

CONCLUSIONS

The main conclusions of this study are:

- Cooling energy savings in the range of 17-48% have been obtained. The percentage of savings increase when: the air velocity increases, the indoor environment category increase, and the number of cooling degree days decreases.
- A reduction of the maximum cooling power in the range 8-22% has been obtained.
- The required power input of the fan is a critical factor. Energy saving may not be achieved with the widely used today methods for air speed increase, such as ceiling, standing, tower and desk fans when the assumptions of this study are fulfilled.

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PAPER E

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ENERGY ANALYSIS OF A PERSONALIZED VENTILATION SYSTEM IN A COLD CLIMATE: INFLUENCE OF THE SUPPLIED AIR TEMPERATURE

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ABSTRACT

In this study the influence of temperature of the supplied air of a personalized ventilation system on energy need has been investigated by means of simulations with IDA-ICE software. GenOpt software was used to determine the optimal supply air temperature. The simulated office room was located in a cold climate. The results reveal that the temperature of air supplied by personalized ventilation and its control strategy have a marked influence on energy consumption. The energy consumption with personalized ventilation may increase substantially (in the range: 61-268%) compared to mixing ventilation alone if energy saving strategies are not applied. The results show that the best supply strategy is to provide air constantly at 20°C, the minimum allowed supply temperature. Energy savings (in the range: 32-47%) may be achieved with personalized ventilation in comparison with mixing ventilation when the room temperature is controlled between 18°C and 29°C.

1. INTRODUCTION

Personalized Ventilation (PV) aims to supply clean and cool air at low velocity and turbulence directly at workplaces. Each occupant may be provided with control of the supplied flow rate and/or supplied air temperature. PV beside its ability to decrease the level of pollution in inhaled air, improves occupants' thermal comfort (Melikov, 2004). Large differences exist between people with regard to preferred temperature (Melikov, 2004). When the occupants are not provided with control over the temperature of the supplied personalized air, the building manager has to define the air supply temperature (θ_{SUP}) needed for providing the occupants with thermal comfort at a minimal level of energy consumption. In a single duct constant air volume system, the θ_{SUP} set-point may be constant, or it may be reset based on the outdoor (θ_{ODA}) or indoor (θ_{IND}) air temperature. The purpose of this study is to investigate the influence of the temperature of the supplied personalized air on energy need, by means of simulations with IDA-ICE software.

2. METHODS

2.1 Input data for the energy simulation

The input data are presented according to the European Standard EN 15265 (2006) which defines the data needed for reporting the hourly energy calculations.

2.1.1 Building location and weather data

An office in a building located in Copenhagen (Denmark) was simulated. The weather is characterized by a cold climate. The ASHRAE IWEC Weather File for Copenhagen is used as input data in the simulation model.

2.1.2 Description of the room

The open-space office has a floor surface area of 6 x 20 m. The room height is 3 m. The external walls are constructed with 20 mm of plaster, 150 mm of glasswool, 240 mm of clay brick and 10 mm of internal plaster; the overall U-value of the external wall is 0.2 $WK^{-1}m^{-2}$. The double panes window with internal low-emissivity glass pane has an U-value of 1.2 $WK^{-1}m^{-2}$, a g-factor or Solar Heat Gain Coefficient equal to 0.61, and a light transmittance equal to 0.77. The window has a total area of 36 m^2 (20% of the floor area, height = 1.8 m and width = 20 m). The window faces south. There is a shading device composed by blinds between the window panes. It has a multiplier for a total shading coefficient equal to 0.39. It is activated when the incident light on the windows is higher than 200 W/m^2. The internal walls, floor and ceiling are adiabatic. The effect of thermal mass is taken into account.

2.1.3 Internal temperature, ventilation and infiltration rate

The thermal comfort conditions and ventilation specifications were chosen in order to comply with the values defined in EN 15251 (2007) for the category I for indoor environment in the room during occupation. From 6:00 till 17:00 the heating and cooling systems kept the internal operative temperature within a range between 21 and 25.5°C. During weekends and night-time the temperature set-back was 12°C in winter and 40°C in summer. Only in Case 10 and Case 11 (Table 1) was the room temperature kept within a range between 18 and 29°C. The design airflow rate was supplied during occupation hours. The airflow rate is calculated according to the European standard EN 15251 (2007). The total air flow rate is the sum of the required ventilation rate per person (10 l/s person for the indoor environmental category I) and per floor area (the building is considered to be a low-polluting, therefore the air flow rate per floor area is 1 $l/(sm^2)$). The floor area per occupant is 10 m^2. Therefore the total airflow rate is equal to 20 l/s per person during occupation hours. The total airflow rate is more than double of the one required in the ASHRAE standard 62.1 (2004). The European standard requires higher ventilation rate than the ASHRAE standard. Twelve occupants were present in the room, thus the total outdoor airflow rate is 240 l/s. The infiltration is taken into account by using an Equivalent Leakage Area (Sherman and Grimsrud, 1980) equal to 0.0093 m^2.

2.1.4 Internal heat gains, occupancy and description of the HVAC system

The twelve occupants contribute to both sensible and latent heat load in the room. The activity level of the occupants was 1.2 met (1 met = 58.15 W/m^2). The balance between sensible and latent heats is calculated by the program. The occupants were present in the room from Monday to Friday, from 8:00 to 17:00 with an hour as break at noon. Saturday and Sunday were free days and no public holidays were involved. The heat load due to office equipment was 6 W/m^2. According to ASHRAE (2005), this value corresponds to a "light load office". The loads follow the schedules of the occupants. The lighting load was 10 W/m^2 during working hours (8:00-17:00). Outside these hours the light was switched off. Two independent systems are used to control the indoor air quality and the thermal comfort in the room. The operative room temperature was controlled by four-pipe fan coil units. An air handling unit with a heat recovery exchanger (efficiency of 0.7) was used to provid the needed outdoor air. The humidity was not controlled during the simulations since this is not common practice in Denmark. A free-cooling strategy during night-time (from 18:00-6:00) from 1 May to 30 September was used. The supplied airflow was 3 l/(sm^2). The free-cooling starts when the outdoor air temperature is at least 5°C cooler then indoor air and the indoor air temperature is at least 25°C. It stops if the indoor air temperature is lower than 21°C or the difference between indoor and outdoor is less than 3°C.

2.1.5 The simulation software

IDA Indoor Climate and Energy (ICE) is a tool for simulation of thermal comfort, indoor air quality and energy consumption in buildings. The mathematical models are described in terms of equations in a formal language, NMF. This makes it easy to replace and upgrade program modules (Vuolle and Sahlin, 2000). GenOpt is an optimization program designed for finding the values of user-selected design parameters that minimize a so-called objective function (or cost function), such as annual energy use, leading to optimal operation of a given system (Wetter, 2001).

2.2 Simulated cases

The temperature of the supplied personalized air (θ_{SUP}) is the parameter investigated in this study. The supply air temperature may be constant, or may vary as a function of the outdoor or indoor air temperature. The simulated cases are listed in Table 1. A mixing ventilation system supplying the air at a constant temperature (16°C) throughout the year is the reference case.

2.2.1 Constant supply air temperature

PV supplies the air close to occupants. Therefore the lowest and highest allowed supply air temperatures are limited by comfort issues. In this study it has been chosen that θ_{SUP} may vary in the range 20-26°C. All the personal supply air temperature profiles presented in the following are restricted within this range. Three cases with constant supply air temperature were investigated (Case 1, 2, 3).

2.2.2 Supply air temperature set-point controlled by outdoor air temperature

Four profiles in which θ_{SUP} is reset based on θ_{ODA} were investigated (shown in Figure 1 A). Three of them were defined by authors (Cases 4, 5 and 6) and the last one, "Case 7", was obtained using GenOpt. GenOpt software was used to find the optimal supply air temperature profile (Case 7) within the boundaries of the room air temperature given by En 15251 (2007) for category I of the indoor environment. GenOpt was set to minimize the sum of energy needed for heating and cooling of the supplied personalized air and the room (mathematically named cost function). In order to minimize the cost function, GenOpt changes the supply air temperatures corresponding to the following fixed outdoor temperatures (-20, 10, 15, 18, 20, 21, 23, 25, 26, 27, 30, 40°C) by choosing an integer value within the range 20-26°C.

Table 1. Simulated cases with PV.

Case	Control strategy of the air supply temperature	Air supply temperature profile	Room temper.[a] [°C]
1	Constant	20°C	21 - 25.5
2	Constant	23°C	21 - 25.5
3	Constant	26°C	21 - 25.5
4	Outdoor	Figure 1A	21 - 25.5
5	Outdoor	Figure 1A	21 - 25.5
6	Outdoor	Figure 1A	21 - 25.5
7	Outdoor	Figure 1 A	21 - 25.5
8	Indoor	Figure 1 B	21 - 25.5
9	Indoor	Figure 1 B	21 - 25.5
10	Constant	20°C	18 – 29
11	Indoor	Figure 1 B	18 – 29

[a] The heating and cooling systems keep the internal operative temperature within the reported range.

2.2.3 Supply air temperature set-point controlled by indoor air temperature

In a constant air volume system the θ_{SUP} set-point can be controlled by the indoor air temperature (θ_{IND}), which in a mixing ventilation principle is also equal to the return air temperature. Two temperature profiles (see Figure 1 B) were analysed. The “Case 8” profile aims to optimize occupants’ thermal comfort. In “Case 11” the air is supplied as in “Case 8” within an expanded room air temperature range 18-29°C. In “Case 9” the air is supplied isothermally within the rage 20-26°C, based on recent findings that indicate that elevated velocity at the breathing zone improves inhaled air quality and compensates for the negative impact of increased temperature on perceived air quality (Melikov et al. 2008).

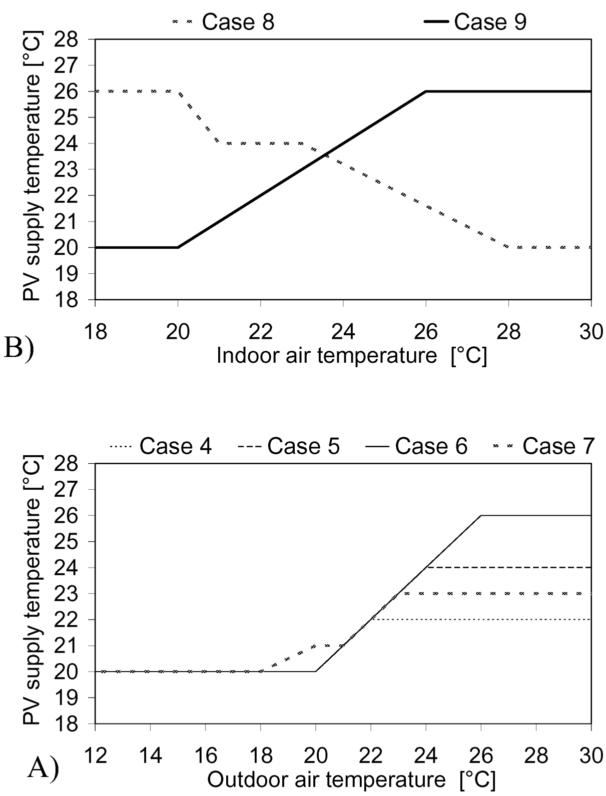

Figure 1. A) PV air supplied temperature profiles as a function of the outdoor air temperature for cases 4, 5, 6, 7. B) PV air supply temperature profiles as a function of the indoor air temperature for “Case 8” and “Case 9”.

3. RESULTS

The “energy need” is the sum of energies for heating (AHU Heating) and cooling (AHU Cooling) of the supplied air in order to obtain the needed θ_{SUP} and for heating (Room Heating) and cooling (Room Cooling) of the conditioned space in order to maintain the intended temperature conditions during a given period of time. The energy need obtained for the simulated cases is shown in Figure 2.

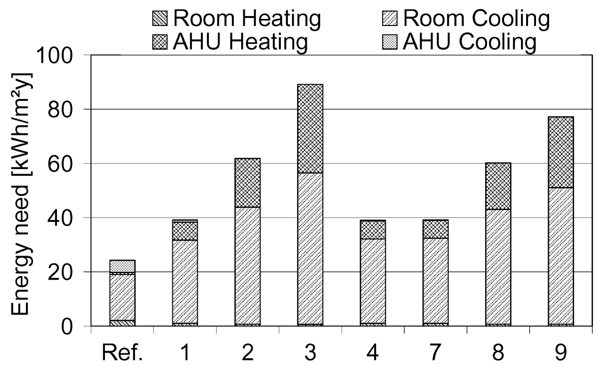

Figure 2. Energy need for the simulated cases (Table 1).

4. DISCUSSION

4.1Influence of the temperature of the supplied personalized air on energy need

The results shown in Figure 2 reveal that the simulated building does not need Room Heating. The building has a good insulation and air tightness and the internal heat gains are sufficient to maintain the required operative temperature. The supplied personalized air needs to be cooled only sporadically; in fact AHU Cooling is equal to zero except for the reference case (Figure 2). The supply temperature and its control strategy have a marked influence on energy consumption (Figure 2). The energy need for the simulated cases is in the range 39.0-89.2 kWh/(m^2y). The energy need for the reference case is 24.3 kWh/(m^2y); it means that by using PV the energy need increases from 61% to 268%. This is mainly due to the fact that the lowest supply air temperature for the PV system was set equal to 20°C. In the reference case the air is supplied at 16°C and it has a free cooling effect. If, for thermal comfort reasons, the personalized supplied air has to be warmed up at least up to 20°C, then the free cooling effect is reduced and the heat added to the air (AHU Heating) has to be compensated by the room cooling system. This phenomenon can be seen in Figure 2: by subtracting the AHU Heating to the Room Cooling, the remaining Room Cooling is constant (in the range between 23.2 and 25.2 kWh/(m^{2}y)).To supply the air at an elevate temperature (23 or 26°C) required a greater amount of energy than to supply at 20°C (see Figure 2). The energy needs for cases 1, 4, 5, 6, and 7 are almost equal. This means that the different supply air control strategies do not differ between them. The reason can be understood by analyzing the outdoor air temperature cumulative profile. In Copenhagen the outdoor air temperature is higher than 20°C only 3.2% of the time in one year, therefore, controlling the θ_{SUP} by the θ_{ODA} using profiles that differ only for θ_{ODA}>20°C does not make any significant difference. Controlling the θ_{SUP} by the θ_{IND} (Case 8 and Case 9) implies high energy consumption. "Case 8" has an energy need almost equal to "Case 2", where the air is supplied constantly at 23°C, but from a comfort point of view, it will perform better because it supplies hot air when it is cold in the room and cool air when it is warm. For the simulated building and for the assumptions made in this paper, the best supply strategy is to provide air constantly at 20°C, the minimum allowed supply temperature.

4.2 Decreased energy need by personalized ventilation

The results presented so far reveal the importance of the control strategy for the energy need. Personalized ventilation may save energy by using the following strategies:

1. Reducing the outdoor airflow rate due to higher ventilation effectiveness (Faulkner et al, 2004; Sekhar et al, 2005).
2. Supplying the personalized air only when occupants are present at the desk (similar to demand ventilation).
3. Expanding the room temperature comfort limits, taking advantage of the ability to create a controlled microenvironment (Bauman et al, 1993; Sekhar, et al, 2003, 2005; Niu et al, 2007).

The energy-saving potential of one of these strategies (no. 3) is demonstrated with Cases 10 and 11, which repeat the simulated Cases 1 and 8 but at expanded room temperature limits between 18°C and 29°C. The energy need for "Case 1", "Case 8", "Case 10" and "Case 11" is shown in Figure 3.

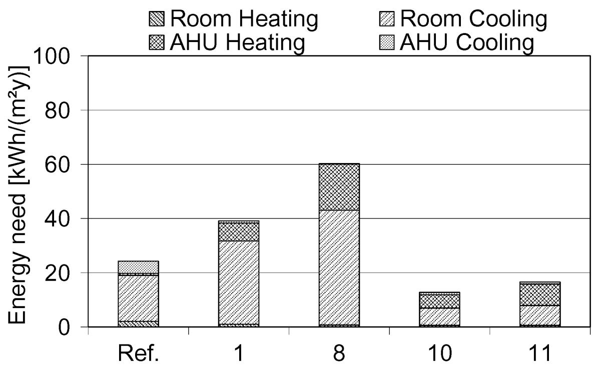

Figure 3. Energy need for the cases 1, 8, 10 and 11.

The energy need for the two cases is strongly reduced, for "Case 10" from 39.2 (Case 1) to 12.8 kWh/(m^{2}y), for "Case 11" from 60.2 (Case 8) to 16.6 kWh/(m^{2}y), corresponding to a reduction of 67% and 72% respectively. From Figure 3 it can be seen that the energy need for "Case 10" and "Case 11" is lower than for the reference case; an energy reduction of 47% and 32% has been obtained. It has been documented that personalized ventilation may provide better inhaled air quality, thermal comfort and protection from cross-infection compared to mixing ventilation (Kaczmarczyk et al. 2004, 2006, Cermak and Melikov 2007). The results of this study reveal that in a cold climate, depending on the control strategy this can be achieved with higher, equal or lower energy consumption compared to traditional system.

5. CONCLUSIONS

The main conclusions of this study are:

- The temperature of air supplied by personalized ventilation and its control strategy have a significant influence on energy consumption. The energy consumption with personalized ventilation may increase substantially (between 61% and 268%) compared to mixing ventilation alone if energy saving strategies are not applied.
- For the simulated building and for the assumptions made in this paper, the best supply strategy is to provide air constantly at 20°C, the minimum allowed supply temperature.
- Energy savings (between 32% and 47%) may be achieved with personalized ventilation compared to mixing ventilation when the room temperature is controlled between 18 °C and 29°C.

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PAPER F

Schiavon, S., Melikov, A.K., Cermak, C., De Carli, M., and Li X. 2007. An Index for Evaluation of Air Quality Improvement in Rooms with Personalized Ventilation Based on Occupied Density and Normalized Concentration. *Proceedings of International Conference Roomvent 2007*. Helsinki, Finland.

An Index for Evaluation of Air Quality Improvement in Rooms with Personalized Ventilation Based on Occupied Density and Normalized Concentration

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SUMMARY

The personal ventilation (PV) system decreases the pollutant concentration mostly in the microenvironment at the workstation, but it can also increase the contaminant in other zone of the room. Therefore, occupant's exposure to pollutant depends on the ratio of time occupant stays at the workstation over total time he/she stays in the room. This ratio is named occupied density (OD).

An index, using a modified definition of OD, is developed to compare and quantify the variation in terms of inhaled pollution by occupant in a room with PV in conjunction with a total-volume ventilation system. The index is applied to data collected during full-scale room measurements.

The results show that the index can be used at the design stage for assessment the benefit of PV when applied in practice for office buildings with different OD. It is for example demonstrated that if the occupied density is lower than 0.5 the use of displacement ventilation alone will be advantageous with regard to human-produced contaminates in comparison when it is combined with PV system.

INTRODUCTION

The Personalized Ventilation (PV) system aims for supplying clean and cool air at low velocity and turbulence directly at workplaces. PV provides user with control of his/her personal microenvironment. Several studies had shown the capacity of a PV system to decrease the pollution in inhaled air [1] and to reduce the transport of contaminants between occupants [2], to improve the perceived air quality and thermal comfort [3]. PV system has the potential to save energy due to the possibility to reduce the ventilation airflow thanks to its high ventilation efficiency and to the possibility of raising the ambient air temperature [4, 5].

Occupants, depending on their activities during working time, may spend only a part of time in the office and even a smaller time at the desk [6, 7, 8]. Most of the studies focused on the measurement of the time an occupant stay in a room over the working time. To the knowledge of the authors only one study reported on the time occupants in office buildings spend at the workstation over the time they stay in the office [9].

To describe the probability distribution of occupants in the room Zhao et al [10] develop the concept of occupied density. The occupied density for the *i*th occupant is the ratio of time that occupant stays in a certain region over the time that occupant stays in the room, e.g. if the occupant stays at the desk for 3 hours and the total time he stays in the room is 4 hours, then the occupied density of the desk of that occupant is 0.75. Computational Fluid Dynamic (CFD) was used to apply this concept for studying the contaminant exposure of occupant when PV is used in combination with a total-volume ventilation system [11]. The results showed that the effect of desk mounted personalized ventilation depends significantly on the type of occupant activity patterns, and so on occupied density, therefore the application of PV should be restricted to certain types of space and human activities. The capacity of PV to decrease the pollutant intake depends on, among other parameters, the time the occupant stays at the desk. The longer the occupant stays at the workstation, the higher he/she will benefit the advantages of PV. In order to apply the occupied density index to full-scale measurement of PV is needed to discretized it and to clearly define which are the zones that influences the human contaminant exposure.

In this paper, a new index combining a normalized concentration and a tailored definition of occupied density is proposed for assessment of benefit in regard to inhaled air quality from use of PV in practice is presented. Data from full-scale measurements are used to demonstrate the applicability of the index. The benefit of this new index is that it can be applied to real measurement and not only to Computational Fluid Dynamic, as the one proposed by Yang at al [11]. It can help to evaluate and quantify the contaminant occupant exposure, therefore the applicability of a PV system in practice.

METHOD

Occupant normalized concentration index

In order to describe the different location an occupant can stays in a room ventilated with PV and, at the same time, do not increase too much the number of measurements needed to quantify the assumed locations a modified definition of the occupied density index suggested by Zhao et al [10] is developed. The occupied zone of the room is divided in two regions:

1. Workstation region, e.g. occupant working at the desk, characterized by the average values of physical parameters measured at the workstation at the height of 1.1 m above the floor.
2. Background region, characterized by the average values of physical parameters measured at the height of 1.7 m above the floor. It is supposed that the occupant is standing in the office when he/she is not at the workstation.

Thus the ratio of time the occupant is at the workstation over the total time he/she stays in the ventilated room, defines the workstation occupied density index ODW:

$$ODW = \frac{\tau_W}{\tau_{TOT}} \qquad (1)$$

Where τ_{TOT} is the total time the occupant stays in the ventilated room, τ_W is the time the occupant spends at the workstation and τ_S is the time the occupant spends standing in the remaining (background) area of the room, e.g. $\tau_{TOT}= \tau_W+ \tau_S$. Similarly, the ratio of time that the occupant spends in the background area of the room over the total time he/she stays in the

ventilated room is defined as, the background occupied density index, ODB. It is clear that the sum of ODB and ODW will be equal to 1.
The normalized concentration of contaminant c is defined by the following equation:

$$c = \frac{\overline{c} - \overline{c}_S}{\overline{c}_E - \overline{c}_S} \qquad (2)$$

where $\overline{c}$ is the contaminant concentration in a point, $\overline{c}_S$ is the contaminant concentration in the supply air, $\overline{c}_E$ is the contaminant concentration in the exhaust air.

The normalized concentration is equal to 1 if there is complete mixing of air and contaminants. If the air quality is better than in the exhaust, the normalized concentration is lower than 1 and vice versa. The supply air has a normalized concentration of 0. The reciprocal value of the normalized concentration is known as ventilation effectiveness [12] or as pollutant removal efficiency [13].

The occupant normalized concentration (C) is the normalized concentration weighed by the workstation occupied density, ODW. i.e. it is the weighed normalized concentration to which the occupant is exposed in average if he/she stays for τ_W at the workstation and for τ_S in the background area. This index is mathematically described by the following equation:

$$C = c_W \cdot ODW + c_S \cdot (1 - ODW) \qquad (3)$$

c_W is the normalized concentration of pollution inhaled by the occupant at the workstation; c_S is the normalized concentration inhaled by the occupant standing in the background area of the room. The occupant normalized concentration (C) is a linear function of ODW. The occupant normalized concentration is an index which determines the quantity of pollutant in air inhaled by the occupant. The occupant normalized concentration can be used to calculate the average pollutant exposure as function of the pollutant distribution in a space and of the occupant activity. It can be applied to total-ventilation system and to personal ventilation system. The lower the normalized concentration is, the better the inhaled air quality is.

The index can be used for comparison of different air distribution systems in regard to quality of the air inhaled by occupants performing office work with different type of occupancy. In the following the index is applied in the case of PV in conjunction with total volume ventilation. Three scenarios are considered: first, the performance of only the total-volume ventilation system in operation is characterized by the normalized concentration defined at the workstation (c_{TVW}) and in the background of the room (c_{TVS}); second, the performance of the total-volume ventilation operating in conjunction with PV which efficiently protects the occupant and provides clean air in inhalation is characterized by the normalized concentration at the workstation (c_{PVpW}), and by the normalized concentration in the background (c_{PVS}); third, the performance of the total-volume ventilation operating in conjunction with PV which does not provide clean air to inhalation (or may be turned off) and does not protect the occupant from air pollution present in the room air is characterized by the normalized concentration at the workstation (c_{PVnpW}),, and by the normalized concentration in the background (c_{PVS}). The defined normalized concentrations are used to calculate the occupant normalized concentration, in the case of total volume ventilation alone (C_{TV}), total volume ventilation in conjunction with personalized ventilation protecting the occupant (C_{PVp}), and total volume ventilation in conjunction with PV which does not protect the occupant efficiently or is turned off (C_{PVnp}). The normalized concentrations, c_{TVW}, c_{TVS} c_{PVpW}, c_{PVnpW} and c_{PVS} are function of the type of the total-volume and the personalized ventilation systems adopted and of the pollution source considered; the occupant normalized concentrations C_{TV}, C_{PVp} and C_{PVnp} are also function of the ODW. The lower the occupant normalized concentration is the better the inhaled air quality will be because the amount of inhaled pollution will be lower.

In order to quantify the difference in performance of two air distribution solutions the Variation of Occupant Normalized Concentration is define by Equation 4:

$$VONC_j = \left(\frac{C_{TV}}{C_{PVj}} - 1 \right) \cdot 100 \qquad (4)$$
$$j = p, np$$

The evaluation is made in case of occupant protected by PV (p) and unprotected occupant (np). A positive value for $VONC_j$ means that the PV system decreases the pollution concentration in inhalation, e.g. improves the quality of the inhaled air, while negative values mean that the total-ventilation system alone can provide occupant with better inhaled air quality. The index $VONC_j$ can be used by designers for justification of the use of a PV system in practice from inhaled air quality point of view.

Validation of the index

The usefulness of the developed index is demonstrated with data collected during full-scale measurements of personalised ventilation in conjunction with total volume ventilation system (mixing and displacement) and total volume ventilation performing alone as reported in [2,14].

A typical two-person office arrangement was simulated in a full-scale test room (4.8 x 5.4 x 2.6 m^3) as shown in Figure 1. Each workstation consisted of a desk with a personalized air terminal device, a breathing thermal manikin simulating a seated occupant, typical office furniture, a PC, and a desk lamp. The total heat load in the office, including six fluorescent light fixtures evenly distributed over the ceiling, was 22.5 W/m^2. A PV system with round movable panel as air terminal devices was used. This air terminal device is designed to supply airflow at low turbulence intensity. Detail description of the device is given in [15].
Two types of total-volume ventilation system were used: mixing and displacement.
A swirl diffuser situated in the centre of the ceiling was used for the mixing ventilation and a semicircular unit placed on the floor in the middle of the longer wall was used for the displacement ventilation. Air was exhausted at the ceiling level. Clean air at 20°C with a total flow rate of 80 l/s (= 4.3 air changes per hour) was supplied to the room, ensuring a maximum room air temperature of 26°C. The 80 l/s was supplied either entirely through the total volume ventilation system or partly through the PV system. When combined, the PV of the front manikin (position 1, Figure 1) was used at 0 or 15 l/s and the PV of the back manikin (position 2) at 15 or 0 l/s.

The breathing thermal manikins' surface temperature was controlled so as to correspond to the skin temperature of an "average" person in thermal comfort. An artificial lung placed outside the manikins simulated the human breathing during light physical work. It consisted of 2.5 s inhalation, 2.5 s exhalation, and pause; exhalation through the nose/inhalation through the mouth; pulmonary ventilation 6 l/min. The exhaled air was heated at 36°C to achieve density similar to the density of air exhaled by people (1.144 kg/m^3: 3.6% CO_2, 95% RH, 34

°C at room temperature 20-26°C). The pause was set at 0.9 and 1.1 s respectively for the two manikins to prevent synchronization. Airborne pollution was simulated by means of tracer-gas. A concentrate and active pollution source was simulated. A constant dose of sulphur hexafluoride (SF_6) was used to mark the air exhaled from the front manikin (here named polluting manikin), representing virulent agents or tobacco smoke.

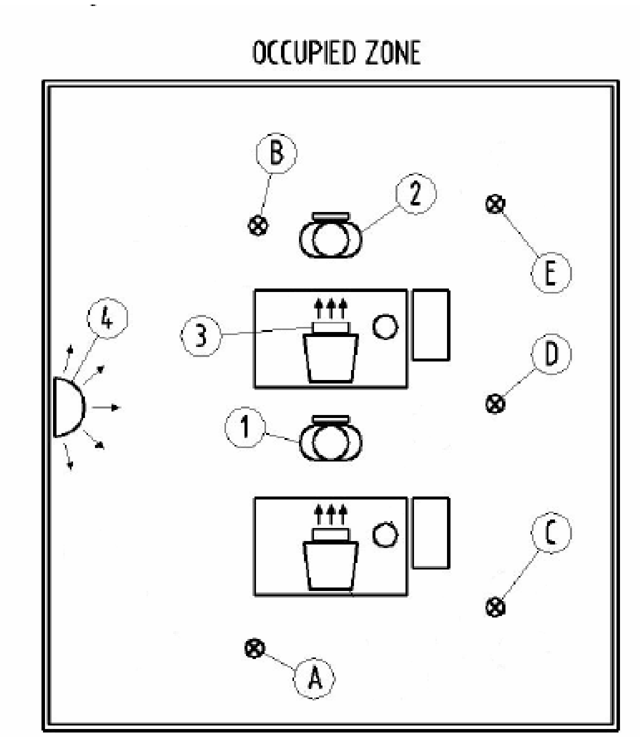

Figure 1 Office plan: (1) Front or polluting thermal breathing manikin, (2) Back or exposed thermal breathing manikin, (3) Personalized ventilation –round movable panel, (4) Displacement ventilation supply, (A)-(D) measuring points at 1.7m above the floor. At ceiling are placed the ceiling light fixture, in the centre of the ceiling is placed the mixing ventilation supply. The total heat of the room is 22W/m^2 (Computers, desk lamps, thermal manikins, ceiling light fixtures)

The concentration of the tracer gas was measured at several points and in the air inhaled by the thermal manikins. A tracer-gas monitor based on a photo-acoustic principle of measurement was used. The characteristics of the instruments and the analysis of uncertainty are detailed presented by Cermak [2]. The conditions and the locations of the measurements of normalized concentrations (c_{TVW}, c_{TVS} c_{PVpW}, c_{PVnpW}, c_{PVS}) are listed in Table 1.

Table 1 Locations and conditions of the normalized concentration measurements for the human-produced contaminant

Normalized Concentration	TV air flow*	PV Front air flow*	PV Back air flow*	Pollution source	Where is measured
c_{TVW}	80	0	0	Front	Inhaled by Back**
c_{TVS}	80	0	0	Front	Average of A B C D E***
c_{PVWp}	65	0	15	Front	Inhaled by Back
c_{PVWnp}	65	15	0	Front	Inhaled by Back
c_{PVS}	65	15	0	Front	Average of A B C D E

* The air flow is expressed in l/s

** The concentration was measure in the air inhaled by the back manikin.

*** The average value measured at 1.7 m above the floor at points A, B, C, D, E (see Figure 1).

RESULTS

Data from two types of total-volume systems (mixing and displacement), an active and concentrate pollution sources, and a PV system using round movable panel as air terminal device were taken from an higher number of experiments in order to show the potential of the new index. The measured normalized concentrations, listed in Table 2, were used in Equation 3 to calculate the occupant normalized concentrations C_{TV}, C_{PVp}, C_{PVnp} as function of ODW.

Table 2 Normalized concentration of human-produced contaminant (SF_6) for mixing ventilation and displacement ventilation. Round movable panel was used as air terminal device

Normalized Concentration	Mixing	Displacement
c_{TVW}	0.93	0.15
c_{TVS}	1.06	0.76
c_{PVpW}	0.13	0.03
c_{PVnpW}	0.98	0.85
c_{PVS}	1.07	0.9

An example is shown in Figure 2, when the total-volume system used was mixing ventilation. Previous analyses of this experimental data compared the normalized concentration for ODW=1, i.e. when occupants are steady exposed to the personal ventilation flow[16]. With the occupant normalized concentration is possible to quantify the occupant exposure for the whole range of ODW values, from 0 till 1. In Figure 2, can be seen that the introduction of PV does not influence significantly the contaminant distribution in the room and the inhaled air quality of the un protected occupant does not change appreciably. The PV is able to reduce the contaminant concentration of the occupants protected by PV. Thanks to the occupant normalized concentration index is possible to show and quantify that, due to the higher concentration of pollutant outside the personal airflow, the occupant exposure to contaminant increase with the reduction of ODW.

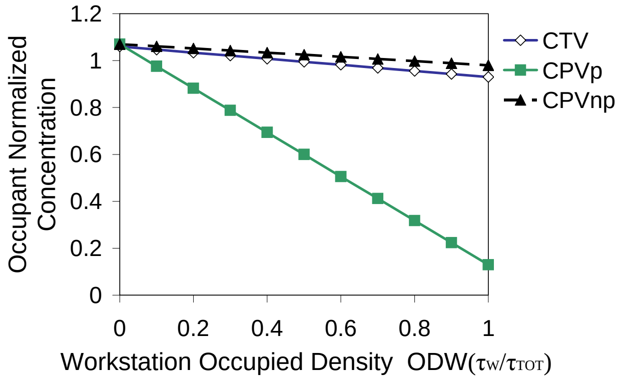

Figure 2 Occupant normalized concentration (C_{TV}, C_{PVp}, C_{PVnp}) versus workstation occupied density (ODW) when the total-volume system used was mixing ventilation.

In Figure 3 is shown the occupant normalized concentrations versus the ODW when total-volume system used was displacement ventilation. The comparison of the results in the figure show that the occupant normalized concentration for displacement ventilation alone at ODW=0.5 is three times higher than at ODW=1, and four time higher than at ODW=0.3. This means that the benefits of a displacement ventilation will be lower for minor values of ODW.

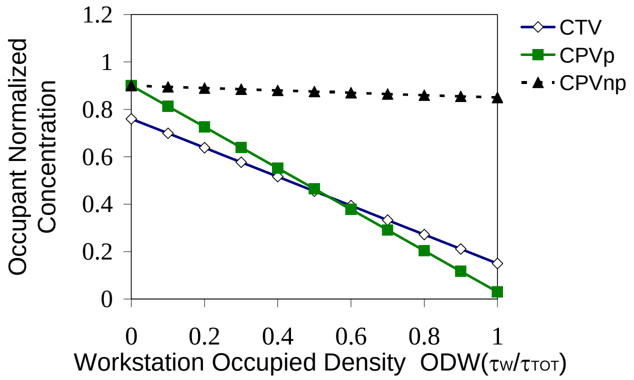

Figure 3 Occupant normalized concentration (C_{TV}, C_{PVp}, C_{PVnp}) versus workstation occupied density (ODW) when the total-volume system used was displacement ventilation.

When ODW=1, the normalized concentration (c_{TVW}) to which a sitting occupant is exposed if only displacement ventilation is used is 0.15 and in the case of combined PV and displacement systems the normalized concentration (c_{PVpW}) of a protected occupant is 0.03. The PV has a ventilation effectiveness that is 5 times higher than the ventilation effectiveness of displacement ventilation and therefore PV is able to provide a better inhaled air quality than displacement ventilation alone. For ODW=0.5 the occupant normalized concentration is the same for the two systems, but the normalized concentration will be almost 2 times higher if the occupant does not use it PV system, i.e. unprotected occupant. For lower values of ODW, displacement ventilation appears to be more effective in providing the occupant better inhaled air quality.

Using the normalized concentrations measured in the experiments, and ODW=0.3 and ODW=0.5 $VONC_j$ was calculated for j=p and np, i.e. for the protected and unprotected occupant. The results are summarized in Table 3.

Table 3 Variation of Occupant Normalized Concentration ($VONC_j$) calculated for protected and unprotected occupant, when the ODW=0.3 or 0.5, e.g. occupied density as identified in office buildings [9]. The results listed in the table are expressed in percent.

TV system	ODW=0.3*		ODW=0.5*	
	p**	np	p	np
Mixing	30	-2	66	-3
Displacement	-10	-35	-2	-48

* to ODW=0.3 correspond clerical work and to ODW=0.5 correspond business work [9]
** p is the protected occupant, np is the unprotected occupant

DISCUSSION

Nobe et al [9] have measured the average seat occupancy rate in a large scale office in Japan. 240 workstations were monitored, during weekday office hours for the attendant occupants only (the outing persons were removed). The results were classified in a function of the type of occupants' activity. It was obtained that for clerical work the average value of ODW was equal to 0.47, for technical work ODW was equal to 0.37, for business work ODW was equal to 0.31. This indicates that occupants stay at the workstation less often than away from it.
Moreover the time an occupant spent at the desk was found to depend on the type of job, e.g. the ODW could be related to the type of human activity.

Figure 2 and Figure 3 show that the occupant exposure to pollutant depends also on the occupied density. Comparing only the performance of a total-volume and PV for ODW=1 is not enough. In order to accurately assess the performance of PV the concentration of pollution at the workstation (typically in inhaled air) as well as in the rest of the room should be reported, This will make it possible to accurately assess the occupant's exposure to contaminants considering also ODW.

Values of ODW lower than 0.5 indicate a strong influence of the pollution concentration in the room away from the workstation on the occupant's exposure. Therefore the performance of PV with regard to inhaled air quality should be evaluated based on at least two criteria: first its ability to provide 100% clean air in inhalation (ODW=1) and second, on its ability to avoid an increase of pollutant concentration in the background region, measured at 1.7 m, compared to the total-volume system alone. It means that the occupant normalized concentration have to be evaluated also for ODW<0.5. For example, in the case of Figure 3, Melikov et al [16] underlined that PV generate an higher concentration of pollutant at 1.7 m than displacement ventilation alone because it promotes mixing of contaminants located in its vicinity. When ODW is lower than 0.5, the occupant exposure will be lower for displacement ventilation alone than with the personal ventilation system. For ODW=0.3, corresponding to business work according to Nobe et al [9], the occupant normalized concentration of displacement ventilation alone is 0.54 while for the PV system is 0.64. The introduced in this paper index makes it possible to assess more realistically occupants' exposure in a room based on non-uniformity in pollution distribution in the room and occupant activity.

$VONC_j$ is used to quantify how much the occupant normalized concentration would vary if PV is used in conjunction with total-volume ventilation system, compared to a total-volume system alone. When mixing ventilation is used in conjunction with PV system, as reported in Table 3, $VONC_p$ would be equal to 30% for occupants performing business work (ODW≈0.3). If the occupants perform a clerical work (ODW≈0.5), $VONC_p$ would increase to 66%. The occupant normalized concentration for unprotected occupant will not change (-2%). In rooms with PV in conjunction with displacement ventilation an occupant performing business type of work (ODW≈0.3) will be exposed to a high pollution concentration $VONC_p$ = -10% while protected with PV system and much higher pollution concentration ($VONC_p$ = -35% when he/she is not protected by PV system. In this way is possible to quantify the improvement or worsening in terms of occupant exposure or total intake contaminant by the $VONC_j$ index and thus to estimate applicability of a PV system.

The main limitations of the developed index are: 1) The database providing occupant density as a function of occupant activity is so far limited; 2) The index considers only two possible position of the occupants, standing in the background area of the room or sitting at the desk.

CONCLUSIONS

- An index which makes it possible to assess more realistically occupant's exposure in a room characterized by a non-uniform pollution distribution is introduced.
- The performance and applicability of personalized ventilation in practice should be evaluated on its ability to provide clean air in inhalation and to avoid an increase of pollutant concentration in the background region, measured at 1.7 m, compared to one generated by the total-volume system alone, therefore they depend also on occupied density.
- It is demonstrated that displacement ventilation alone was able to provide to the occupant with better inhaled air quality than displacement ventilation in conjunction

with PV with round movable panel as an air supply device when occupied density is lower than 0.5.

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