Instituto Superior de Engenharia do Porto

Experimental and numerical study on the evaluation of ventilation efficiency

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Thesis submitted for the Master degree in Mechanical Engineering: Energy

Instituto Superior de Engenharia do Porto
Departamento de Engenharia Mecânica

October 2013

Relatório da Unidade Curricular Dissertação/Projecto/Estágio do 2º ano do Mestrado em Engenharia Mecânica – Área de Especialização em Energia.

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Instituto Superior de Engenharia do Porto

Outubro de 2013

"The beginning of all sciences is the surprise of things being what they are."
Aristóteles

To Elsa

Acknowledges

First of all, I would like to thank Professor Olga Castro who gave me the possibility to accomplish my Master's thesis on this subject. She guided and supported me all along the process giving precious comments and advice.

Then I want to express all my gratitude to my wife and my parents. They have been of a great help and support during this years and their contribution has made them become a great and important part of my life.

Finally I am grateful to all my colleagues from the Master is Mechanical Engineering who always helped me to solve the problems making the frequency of this course easier and cheerful.

Abstract

Buildings account for 40% of total energy consumption in the European Union. The reduction of energy consumption in the buildings sector constitute an important measure needed to reduce the Union's energy dependency and greenhouse gas emissions. The Portuguese legislation incorporate this principles in order to regulate the energy performance of buildings. This energy performance should be accompanied by good conditions for the occupants of the buildings. According to EN 15251 (2007) the four factors that affect the occupant comfort in the buildings are: Indoor Air Quality (IAQ), thermal comfort, acoustics and lighting. Ventilation directly affects all except the lighting, so it is crucial to understand the performance of it. The ventilation efficiency concept therefore earn significance, because it is an attempt to quantify a parameter that can easily distinguish the different options for air diffusion in the spaces. The two indicators most internationally accepted are the Air Change Efficiency (ACE) and the Contaminant Removal Effectiveness (CRE). Nowadays with the developed of the Computational Fluid Dynamics (CFD) the behaviour of ventilation can be more easily predicted. Thirteen strategies of air diffusion were measured in a test chamber through the application of the tracer gas method, with the objective to validate the calculation by the *MicroFlo* module of the *IES-VE* software for this two indicators. The main conclusions from this work were: that the values of the numerical simulations are in agreement with experimental measurements; the value of the CRE is more dependent of the position of the contamination source, that the strategy used for the air diffusion; the ACE indicator is more appropriate for quantifying the quality of the air diffusion; the solutions to be adopted, to maximize the ventilation efficiency should be, the schemes that operate with low speeds of supply air and small differences between supply air temperature and the room temperature.

Keywords : Air Change Efficiency, Contaminant Removal Effectiveness, Ventilation, Tracer Gas Method, Indoor Air Quality, CFD, Air Diffusion.

Resumo

Os edifícios são responsáveis por 40% da energia consumida na União Europeia. A redução do consumo de energia no sector dos edifícios constitui uma importante medida necessária para reduzir a dependência energética da União e as emissões de gases de efeito estufa. A legislação Portuguesa incorpora estes princípios, a fim de regular o desempenho energético dos edifícios. Este desempenho energético deve ser acompanhado de boas condições para os ocupantes dos edifícios. De acordo com EN 15251 (2007) os quatro fatores que afetam o conforto dos ocupantes dos edifícios são: Qualidade do Ar Interior (QAI), conforto térmico, conforto acústico e iluminação. A ventilação afeta diretamente todos estes fatores, exceto a iluminação, por esta razão é fundamental compreender o desempenho da mesma. Neste sentido, o conceito de eficiência de ventilação ganha importância, porque é uma tentativa de quantificar um parâmetro que pode distinguir as diferentes estratégias para a difusão do ar nos espaços. Os dois indicadores internacionalmente mais aceites são: *Air Change Efficiency* (ACE) e *Contaminant Removal Effectiveness* (CRE). Hoje em dia, com o desenvolvimento das técnicas de *Computacional Fluid Dynamics* (CFD), o comportamento da ventilação pode ser mais facilmente previsto. Treze estratégias de difusão do ar foram medidas numa câmara de testes através da aplicação do método dos gases traçadores, com o objetivo de validar o cálculo efetuado pelo módulo *MicroFlo* do software *IES-VE* para estes dois indicadores. As principais conclusões deste trabalho foram: os valores das simulações numéricas estão de acordo com as medições experimentais; o valor do CRE é mais dependente da posição da fonte de contaminação, do que da estratégia utilizada para a difusão do ar; o indicador ACE é mais adequado para a quantificação da qualidade da difusão de ar; as soluções a serem adoptadas, para maximizar a eficiência de ventilação, devem ser os sistemas que operam com baixas velocidades de insuflação do ar e pequenas diferenças de temperatura entre o ar insuflado e o espaço.

Palavras-chave: ACE, CRE, Ventilação, Método dos Gases Traçadores, Qualidade do Ar Interior, CFD, Difusão de Ar.

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Nomenclature

Roman Characters

A_i	age of air at location i
A_{avg}	arithmetic average of the ages of air measured at breathing level within the test space
$A_{ex,m}$	age of air in exhaust air-stream m
$C + R$	sensible heat loss from skin
$< C >$	room average contaminant concentration
$C_{i,avg}$	the time-averaged tracer gas concentration at location i
$C_{i(t_{start})}$	the tracer gas concentration at location i at time t_{start}
C_p	concentration of contaminant at a point p
C_{res}	rate of convective heat loss from respiration
c_{SUP}	concentration of CO_2 in the supply air
E	air-change effectiveness
E_{dif}	evaporation of moisture diffused through the skin
E_{res}	rate of evaporative heat loss from respiration
E_{rsw}	latent heat flow from sweat evaporation
E_{sk}	total rate of evaporative heat loss from skin
E_z	zone air distribution effectiveness
H_b	body height
I	air exchange (or change) rate
I_N	nominal air exchange rate
L_{AFmax}	noise maximum level
L_{eq}	noise equivalent level
M	rate of metabolic heat production
m	total inaccuracy of a measuring point
M_{act}	metabolic rate required for the person's activity
m_C	fluctuation of CO_2 concentration
m_G	inaccuracy of gas analyser
M_{shiv}	metabolic level required for shivering
m_Δ	total inaccuracy of concentration differential
n_{AC}	number of air changes
NR	noise rating curves
P, p	pressure
p_a	ambient water vapour pressure
Pe	Peclet number
Q	volumetric flow rate of air into space
$Q_{ex,m}$	rate of airflow in exhaust air-stream m

Q_{oa}	outdoor airflow rate
q	contaminant injection rate
q_H	head heat loss
q_{res}	total rate of heat loss through respiration
q_{sk}	total rate of heat loss from skin
Re	Reynolds number
RQ	respiratory quotient
s	standard deviation
S_{cr}	rate of heat storage in core compartment
s_R	standard deviation of room air concentration
S_{sk}	rate of heat storage in skin compartment
T	temperature
t	time
$\bar{t_r}$	mean radiant temperature
t_{sk}	skin temperature
t_{start}	time when tracer injection is stopped at the beginning of tracer gas decay
t_{stop}	time of the final tracer gas measurement at location i
Δt	time interval
V	interior volume of space
$\dot{V}$	flow rate
v	air velocity
V_c	equivalent volume of contaminant in the room
$\dot{V}_{CO_2}$	CO_2 emission of one person
$\dot{V}_{O_2}$	O_2 consumption of one person
W	rate of mechanical work accomplished
W_b	body mass
w_{sk}	skin wettedness

Greek Characters

ϵ	rate of dissipation of turbulence kinetic energy
ε^a	air change efficiency
ε_p^a	local air change index
ε^c	contaminant removal effectiveness
ε_p^c	local air quality index
η	coefficient of air change performance
θ_a	air temperature in the room
θ_{SUP}	temperature of supply air
θ	temperature
κ	turbulence kinetic energy
τ	time constant
$< \bar{\tau} >$	room mean age of air
τ_n	nominal time constant
τ_n^c	nominal time constant for the contaminant
$\bar{\tau}_r$	air change time
$\bar{\tau}_p$	local mean age of air

Abbreviations

ACE	*Air Change Efficiency*
ACH	*Air Changes per Hour*
ACR	*Air Change Rate*
ADENE	*Agência para a Energia*
AHU	*Air Handling Unit*
ASHRAE	*American Society of Heating, Refrigerating and Air Conditioning Engineers*
BRI	*Building-Related Illness*
CFD	*Computational Fluid Dynamics*
CRE	*Contaminant Removal Effectiveness*
CV	*Constant Volume Ventilation*
DCV	*Demand Controlled Ventilation*
DEM	*Department of Mechanical Engineering*
ECA	*European Concerted Action*
EPBD	*European Directive on the Energy Performance of Buildings*
FTIR	*Fourier Transform Infrared Spectroscopy*
HVAC	*Heating, Ventilation and Air Conditioning*
IAP	*Indoor Air Pollution*
IAQ	*Indoor Air Quality*
IDA	*Indoor Air*
IES	*Integrated Environmental Solutions*
IR	*Infrared*
NDIR	*Non Dispersive Infrared Absorbance*
NTC	*Negative Temperature Coefficient*
ODA	*Outdoor Air*
PAS	*Photoacoustic Spectroscopy*
PFC	*Perfluorocarbons*
PMV	*Predicted mean vote*
PPD	*Predicted percentage dissatisfied*
PPM	*Parts per Million*
RCCTE	*Regulamento das Características de Comportamento Térmico dos Edifícios*
RSECE	*Regulamento dos Sistemas Energéticos de Climatização em Edifícios*
SATEC	*Sala de Testes de Equipamentos de Climatização*
SCE	*Sistema Nacional de Certificação Energética e da QAI dos Edifícios*
SBS	*Sick Building Syndrome*
UN	*United Nations*
VOC	*Volatile Organic Compounds*
WHO	*World Health Organization*

Chapter 1

Introdution

1.1 Motivation and objectives of the present work

According with EPBD - 31/CE (2010) and EED - 27/CE (2012) buildings account for 40% of total energy consumption in the European Union. The sector is expanding, which is bound to increase its energy consumption. In these directives is also emphasized the need to promote energy efficiency strategies focusing on the goal of reducing the consumption of primary energy by the European Union at least 20% by 2020.

Therefore, reduction of energy consumption and the use of energy from renewable sources in the buildings sector constitute important measures needed to reduce the Union's energy dependency and greenhouse gas emissions.

The Portuguese legislation SCE (2006), RSECE (2006) and RCCTE (2006) incorporate this principles in order to regulate the energy performance of buildings. The energy performance of buildings should be accompanied by a good air quality, designated by IAQ, associated to conditions of comfort for the occupants, according to ASHRAE 55 (2004), which establishes the criteria for setting the environmental of thermal comfort in buildings.

While the energy performance (EPBD - 31/CE (2010)) translate the operation costs of the building, the aspects of IAQ directly affect the health of occupants, as well the aspects relating to thermal comfort, acoustics and lighting influence the productivity of the occupants according to EN 15251 (2007). Of this four factors the ventilation affect directly three: IAQ, thermal comfort and acoustics. The aspects related to the adequate ventilation rate for the occupants of buildings and the hygienic of ventilation systems already have been quantified in American standards, ANSI/ASHRAE 62.1 (2007), and European standards through, CR 1752 (1998) and EN 13779 (2007). Therefore it is essential to decrease the amount of energy used to make the proper ventilation of the interior spaces of buildings through the choice of strategies that provide increased the ventilation efficiency. Overlooking the implementation of optimum standards for ventilation rates and ventilation efficiencies, currently are in development numerous studies, either analytical or prescriptive nature, in order to obtain acceptable values for health, that minimize consumption energy and promote the productivity and well being of the occupants.

Currently the parameters used for ventilation are well specified in good design practice. The only parameter which provides a subjective quantification is its efficiency. This parameter is a combination of various technical, economic and aesthetic options the designer has to take.

This thesis aims to measure the efficiency of ventilation in a test room, with the objective of validate the numerically simulations performed with *IES-VE* software. This tests will be performed in several air diffusion strategies, with the objective of determine the best solutions to use. In experimental tests will be followed three different methodologies:

- methodology used in NTvvs 047 (1985) (Nordtest method), which represents the ability to change the supply air in a space and involves the concept of "age of air". For this purpose will be carried concentration decay tests using the tracer gas CO_2;
- methodology used in ANSI/ASHRAE 129 (1997), which is similar to the Nordtest method, differing only the equations used;
- methodology used in CR 1752 (1998), which represents the contaminant removal capacity of the occupied space. This method involves measuring the steady state CO_2 concentrations in zones of air intake and exhaust, as well in the occupant breathing zone.

1.2 Content of the thesis

This thesis is divided into six chapters, including this introductory chapter, and an appendix.

The first chapter seeks to do by way of introduction, a general framework of the theme, which features some of the assumptions taking account when the energy efficient ventilation are analysed. Finally, we present the objectives of this dissertation and motivational aspects.

The second chapter presents the main reasons for the use of ventilation in buildings, the factors that influence this use, the good design practice and the indicators used to determine the efficiency of ventilation. This was followed by a description of methods used for measuring the efficiency of ventilation. Finally was presented the main methodologies used for predicting the behaviour of ventilation in buildings.

The third chapter focuses on the characterization of the laboratory facilities, measuring equipment and procedures used for the experimental determination of the ventilation efficiency indicators.

In chapter four is made a presentation of the software used for the numerical simulations. Also described are the assumptions used for the modelling of the test chamber.

In the fifth chapter is presented the results obtained from the experimental measurements and numerical simulations, as well as their discussion.

Finally, in the sixth chapter, are exposed the main conclusions resulting from this dissertation.

Chapter 2

Ventilation in buildings

2.1 Background of ventilation

Ventilation is essential for the health and comfort of building occupants. It is specifically needed to dilute and remove pollutants emitted from unavoidable sources such as those derived from metabolism and from the essential activities of occupants (Liddament (1996)).

2.1.1 What is ventilation?

Definitions covering ventilation and the flow of air into and out of a space include:

Purpose provided (intentional) ventilation: ventilation is the process by which "clean" air (normally outdoor air) is intentionally provided to a space and stale air is removed. This may be accomplished by either natural or mechanical means.

Air infiltration and exfiltration: in addition to intentional ventilation, air inevitably enters a building by the process of air infiltration. This is the uncontrolled flow of air into a space through adventitious or unintentional gaps and cracks in the building envelope. The corresponding loss of air from an enclosed space is termed 'exfiltration'. The rate of air infiltration is dependent on the porosity of the building shell and the magnitude of the natural driving forces of wind and temperature. Some Countries have introduced airtightness Standards to limit infiltration losses. Limb (1994)

Duct leakage: air leakage from the seams and joints of ventilation, heating and air conditioning circulation ducts can be substantial. When, as is common, such ducting passes through unconditioned spaces, significant energy loss may occur. Modera (1993), for example, estimates that as much as 20% of the heat from typical North American domestic warm air heating systems can be lost through duct leakage. Pollutants may also be drawn into the building through these openings.

Air recirculation: air recirculation is frequently used in commercial buildings to provide for thermal conditioning. Recirculated air is usually filtered for dust removal but, since oxygen is not replenished and metabolic pollutants are not removed, recirculation should not usually be considered as contributing towards ventilation need.

In this study only purpose provided ventilation will be treated.

2.1.2 Reasons for ventilation

Ventilation of buildings in general may be done for several reasons, the most important is to remove or dilute the indoor generated pollutants and supply fresh air for human beings. Pollutants from people who produce human effluents, as well as emissions from smoking, combustion, building materials, furniture, house hold and cleaning products, the ingress of soil gases has to be removed from the building and (particularly in dwellings) to provide oxygen to combustion appliances . Ventilation is also needed to reduce the exposure from air borne microbes causing infectious diseases.

Other reasons for ventilation may be e.g. humidity control to: prevent growth of dust mites; prevent from microbiological growth in the building structures: walls, floors, ceilings; prevent the building constructions from damages; control the pressure levels in building to prevent pollutants from spreading.

Moreover ventilation nowadays can also be used for so called "free cooling". This is a way to control temperature, by ventilate the room with quite high rates to remove heat from the building to outside (ECA 23 (2003)).

2.1.3 Type of ventilation

Natural ventilation

Natural ventilation is one of the possible strategies for controlling the indoor air quality. Traditionally, ventilation needs have been met by 'natural' ventilation in which the flow process is driven by wind and temperature. In mild climates, design has often relied on no more than the natural porosity of the building, combined with window opening. In colder climates, natural ventilation designs tend to be more specific and incorporate carefully sized air inlets combined with passive ventilation stacks. Other climates might take advantage of a prevailing wind to drive the ventilation process. The main drawback of natural ventilation is lack of control, in which unreliable driving forces can result in periods of inadequate ventilation, followed by periods of over ventilation and excessive energy waste. Good design can provide some measure of flow control but normally it is necessary for the occupant to adjust ventilation openings to suit demand. Despite the difficulty of control, natural ventilation is still (particularly in dwellings) relied upon to meet the need for fresh air in many types of building throughout the world (Liddament (1996)).

Mechanical ventilation

In principle, the shortcomings of natural ventilation can be overcome by mechanical ventilation. These systems are capable of providing a controlled rate of air change and respond to the varying needs of occupants and pollutant loads, irrespective of the vagaries of climate. Some systems enable incoming supply air to be filtered while others have provision for heat recovery from the exhaust air stream. In milder climate, however, the potential advantages of mechanical ventilation, especially for smaller buildings, can often be outweighed by installation and operational cost, maintenance needs and inadequate return from heat recovery. Regardless of climate, mechanical ventilation is often essential in large, deep plan office buildings where fresh air must penetrate to the centre of the building and high heat gains can cause over heating. Several configurations of mechanical ventilation are possible with each having a specific range of applications. The basic options are: supply ventilation, extract (or exhaust) ventilation and balanced supply extract systems (Liddament (1996)).

Natural and mechanical combined ventilation

This type of ventilation fills the shortcomings of natural ventilation with mechanical ventilation in a combined system.

2.2 Indoor Air Quality and comfort

2.2.1 Indoor Air Pollution

The impact of IAP on man may consist of undesired health effects of different types, ranging from sensory annoyance or discomfort to severe health injuries. "Health" is defined, for the purpose of this study, according to the well-known WHO definition as "A state of complete physical, mental and social well-being, and not merely the absence of disease or infirmity". The public health relevance of the effects of IAP varies, not only from substance to substance, but also from country to country, depending on the presence of specific local sources and climatic influences.

Concern about the health effects associated with indoor air dates back several hundred years, and has increased dramatically in recent decades. This attention was partially the result of increased reporting by building occupants of complaints about poor health associated with exposure to indoor air. Since then, two types of diseases associated with exposure to indoor air have been identified: sick building syndrome (SBS) and building-related illness (BRI).

SBS describes a number of adverse health symptoms related to occupancy in a "sick" building, including mucosal irritation, fatigue, headache, and, occasionally, lower respiratory symptoms and nausea. There is no widespread agreement on an operational definition of SBS (ASHRAE (2009)). In contrast, the term BRI is used when symptoms of diagnosable illness are identified and can be attributed directly to airborne building contaminants (EPA (1991)).

In an effort to understand the causes of sick buildings, many parameters have been investigated. These have tended to focus on ventilation performance, contaminants and various other miscellaneous parameters. Various studies have included surveys and medical examinations. No single cause has been identified and much of the evidence concerning the causes of sick buildings is inconclusive (Liddament (1990)).

Relationship between the type of ventilation system and symptoms is also inconclusive. Burge (1992) found fewer symptoms in naturally ventilated rather than air conditioned office buildings. Similar results are reported by other researchers, especially in air conditioned buildings in which humidifiers are used. On the other hand Sundell (1994) found, in Sweden, that SBS was more pronounced in office buildings that were either naturally ventilated or ventilated by mechanical extract only, than in offices incorporating balanced extract-supply systems.

Indoor and outdoor pollution

According with Alfano et al. (2010), people spent 90% of their life indoors and 10% outdoors. It means that the indoor pollution levels can substantially influence the total air exposure level.

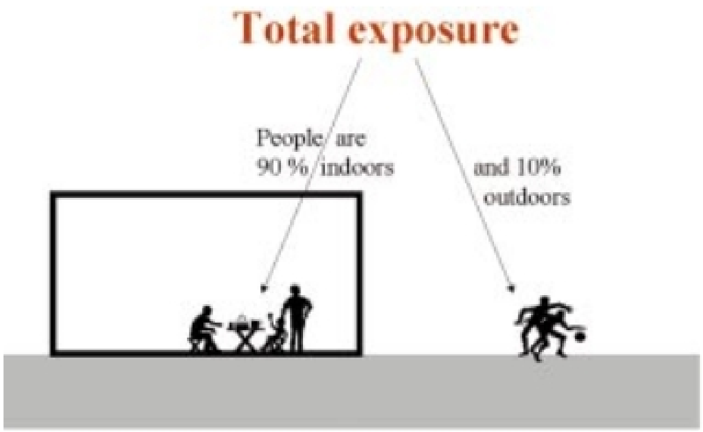

Figure 2.1: Time typically spent indoors and outdoors. ECA 23 (2003)

Indoor pollutants are derived from both outdoor and indoor sources. Each of these sources tend to impose different requirements on the control strategies needed to secure good health and comfort conditions.

Clean outdoor air is essential for achieving good indoor air quality. Although air cleaning is possible, it is costly and not effective in the many offices and dwellings that are either naturally ventilated, leaky or are ventilated by mechanical extract systems. The major sources of outdoor pollutants are:

- **Industrial contaminants**: oxides of nitrogen and sulphur; ozone; lead; volatile organic compounds; smoke, particulates and fibres.
- **Traffic pollution**: carbon monoxide; carbon dust; lead; oxides of nitrogen; fuel additives.
- **Emissions from adjacent exhausts and cooling towers**
- **Rural pollution**

- **Soil borne pollutants**: radon; methane; moisture.

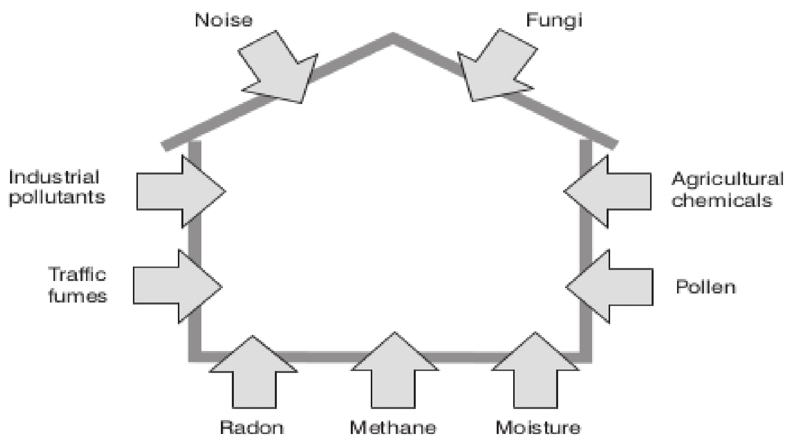

Figure 2.2: Overview of Typical Outdoor Pollutants. Liddament (1996)

Pollutants emitted inside buildings are derived from metabolism, the activities of occupants, and emissions from materials used in construction and furnishing. The major sources of indoor pollutants are:

- **Carbon dioxide**
- **Carbon monoxide**
- **Formaldehyde**
- **Moisture**
- **Odour**
- **Ozone**
- **Particulates**
- **Volatile organic compounds (VOC's)**

Of the pollutants enumerate above we gone analyse deeply the carbon dioxide. Carbon dioxide is a product of metabolism. It is also a product of combustion, in which case it can be found in relatively large concentrations in cooking areas and in areas in which unventilated heating appliances are used. By quantity it is the most important human bioeffluent. At the low concentrations typically occurring indoors CO_2, is harmless and it is not perceived by humans. Still it is a good indicator of the concentration of other human bioeffluents being perceived as a nuisance. As an indicator of human bioeffluents, CO_2, has been applied quite successfully for more than a century (Pettenkofer (1858)). Fig 2.4 shows the percentage of dissatisfied visitors as a function of the CO_2, concentration (above outdoors) for spaces where sedentary occupants are the exclusive pollution sources. Although CO_2, is a good indicator of pollution caused by sedentary human beings, it is often a poor general indicator of perceived air quality. It does not acknowledge the many perceivable pollution sources not producing CO, Berglund and Lindvall (1979), and certainly not the non-perceivable hazardous air pollutants such as carbon monoxide and radon.

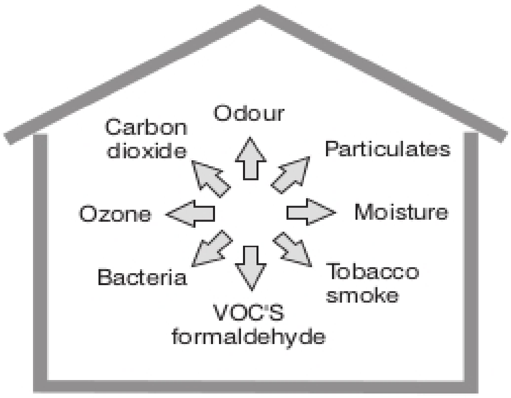

Figure 2.3: Overview of Typical Indoor Pollutants. Liddament (1996)

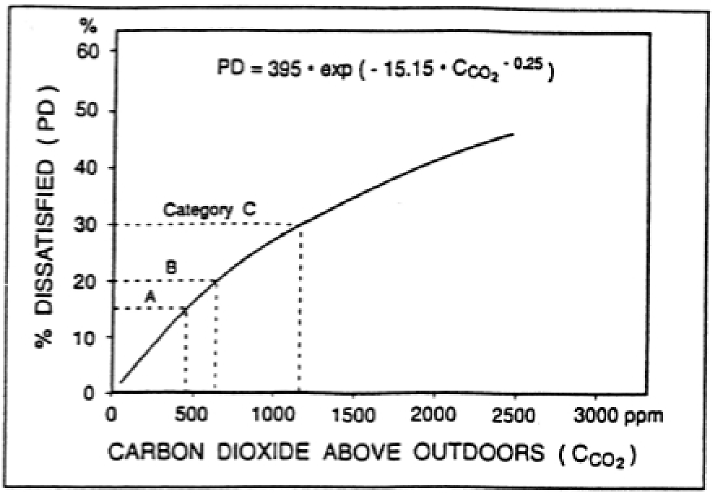

Figure 2.4: Carbon Dioxide as an indicator off human bioeffluents. CR 1752 (1998)

2.2.2 Reducing indoor pollutant concentration

Control strategies for reducing the concentration of pollutants in indoor air depend on the source of contamination. While ventilation or dilution with fresh outdoor air can help to reduce contaminant concentration from emission sources within a space, it cannot eliminate the contamination entirely. Neither is ventilation effective when the incoming air itself is polluted.

Basically we can use the follow strategies to reduce the indoor pollutant concentration:

- **Controlling outdoor air pollutants**: filtration; sitting air intakes; air quality controlled fresh air dampers; building air-tightness.

- **Controlling indoor air pollutants**: source control; enclosing and ventilating at source; general dilution (or displacement) ventilation.

2.2.3 Comfort

Comfort is associated with the physical interaction of the individual with the surrounding environment.

Thermal comfort

Thermal comfort is that condition of mind which expresses satisfaction with the thermal environment. Because there are large variations, both physiologically and psychologically, from person to person, it is difficult to satisfy everyone in a space. The environmental conditions required for comfort are not the same for everyone (ASHRAE 55 (2004)).

A human being's thermal sensation is mainly related to the thermal balance of his or her body as a whole. There are six primary factors that must be addressed when defining conditions for thermal comfort. A number of other, secondary factors affect comfort in some circumstances, such as state of health, level of physical activity, gender, working environment and individual preferences. The six primary factors are listed below (ISO 7730 (2005)):

- Metabolic rate
- Clothing insulation
- Air temperature
- Radiant temperature
- Air speed
- Humidity

When these factors have been estimated or measured, the thermal sensation for the body as a whole can be predicted by calculating the predicted mean vote (PMV). The PMV is an index that predicts the mean value of the votes of a large group of persons on the 7-point thermal sensation scale (see fig. 2.5), based on the heat balance of the human body. Thermal balance is obtained when the internal heat production in the body is equal to the loss of heat to the environment. In a moderate environment, the human thermoregulatory system will automatically attempt to modify skin temperature and sweat secretion to maintain heat balance.

The predicted percentage dissatisfied (PPD) index provides information on thermal discomfort or thermal dissatisfaction by predicting the percentage of people likely to feel too warm or too cool in a given environment. The PPD can be obtained from the PMV (see fig. 2.6). One of the curios conclusions of the application of this standard is that even in a thermoneutral environment, there is likely to have 5% of dissatisfied people. ISO 7730 (2005) defines a thermal environment quality as an environment in which the percentage of people dissatisfied is less than 10%. As also acceptable that thermal environments generate up to 20% dissatisfied people whenever it is not considered essential to have a great accuracy level in monitoring thermal condition.

+ 3	Hot
+ 2	Warm
+ 1	Slightly warm
0	Neutral
– 1	Slightly cool
–2	Cool
– 3	Cold

Figure 2.5: Seven-point thermal sensation scale. ISO 7730 (2005)

Thermal discomfort can also be caused by unwanted local cooling or heating of the body. The most common local discomfort factors are radiant temperature asymmetry (cold or warm surfaces), draught (defined as a local cooling of the body caused by air movement), vertical air temperature difference, and cold or warm floors.

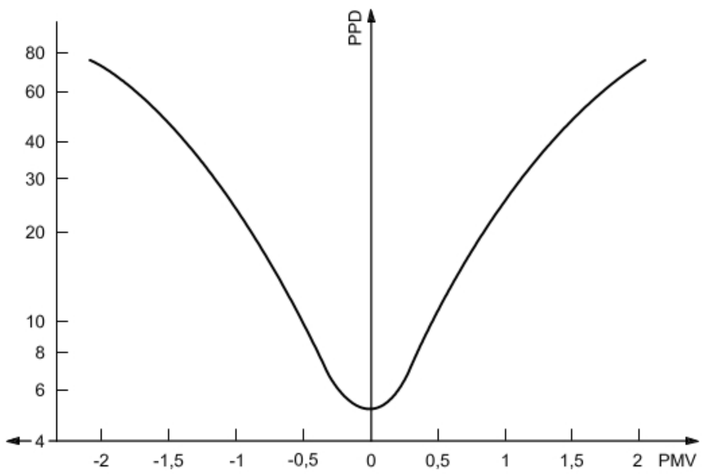

Figure 2.6: PPD as function of PMV. ISO 7730 (2005)

The energy consumption of buildings depends significantly on the criteria for the indoor environment, including environmental and heat indoor air quality, also affects health, productivity and occupant comfort. On the other hand, sustainability and energy efficiency have been assume a greater emphasis. Reducing energy consumption is a priority need of Europe and the world. The European Directive on the Energy Performance of Buildings (EPBD - 91/CE (2002)) came to give greater importance to energy consumption, comfort and indoor air quality by imposing the change and better regulation of EU member states to reduce consumption energy associated with buildings. Comfort standards begin to be rethought and in the present the concept of adaptive comfort begins to gain importance in determining the standard of comfort, as with the EN 15251 (2007) - European standard of comfort. The fundamental assumption of the adaptive approach is expressed by the adaptive principle:

"If a change occurs such as to produce discomfort, people react in ways which tend to restore their comfort" (Humphreys and Nicol (1998)).

This principle codifies the behaviour of building occupants which takes two basic forms:

- Adjustments to the optimal comfort temperature by changes in clothing, activity, posture, etc. so that the occupants are comfortable in prevailing conditions;
- Adjustment of indoor conditions by the use of controls such as windows, blinds, fans and in certain conditions mechanical heating or cooling. Occupants may also migrate around the room to find improved conditions.

Fig. 2.7 and 2.8 shows the EN 15251 (2007) proposes four categories of thermal environment.

Category	Explanation
I	High level of expectation and is recommended for spaces occupied by very sensitive and fragile persons with special requirements like handicapped, sick, very young children and elderly persons
II	Normal level of expectation and should be used for new buildings and renovations
III	An acceptable, moderate level of expectation and may be used for existing buildings
IV	Values outside the criteria for the above categories. This category should only be accepted for a limited part of the year

Figure 2.7: Description of the applicability of the categories used. EN 15251 (2007)

EN 15251 (2007) presents also recommendations for the operative temperatures (room temperatures) for office buildings and other buildings of similar type used mainly for human occupancy with mainly sedentary activities and dwelling, where there is easy access to operable windows and occupants may freely adapt their clothing to the indoor and/or outdoor thermal conditions (see fig. 2.9).

After analyse the factors that influence the thermal comfort in a space, is clear that a appropriate level of ventilation is fundamental to maintain that environment.

Odour

Objective odour creates discomfort and often provides an indication of poor indoor air quality. It is emitted as part of metabolism and can give warning of high levels of formaldehyde and VOC emissions from furnishings and fabrics. It is also emitted by many other compounds that may be found in buildings. Often occupants become acclimatised to odours that are very noticeable to visitors. In general, good indoor air quality

Category	Thermal state of the body as a whole	
	PPD %	Predicted Mean Vote
I	< 6	-0,2 < PMV < + 0,2
II	**< 10**	**-0,5 < PMV < + 0,5**
III	< 15	-0,7 < PMV < + 0,7
IV	> 15	PMV<-0,7; or +0,7<PMV

Figure 2.8: Recommended categories for design of mechanical heated and cooled buildings. EN 15251 (2007)

is equated with an absence of odour.

Humans perceive the air by two senses. The olfactory sense is situated in the nasal cavity and is sensitive to several hundred thousand odorants in the air. The general chemical sense is situated all over the mucous membranes in the nose and the eyes and is sensitive to a similarly large number of irritants in the air. It is the combined response of these two senses that determines whether the air is perceived fresh and pleasant or stale, stuffy and irritating (ECA 11 (1992)).

Since it is not yet possible for odour intensity to be measured with instrumentation, assessment is sometimes based on the judgement of visiting ‘panellists’. Perceived air quality may be expressed as the percentage of dissatisfied, i.e. those persons who perceive the air to be unacceptable just after entering a space. For air polluted by human bioeffluents fig. 2.10 shows the percentage of dissatisfied as a function of the ventilation rate per standard person (average sedentary adult office worker feeling thermally neutral). The pollution generated by such a standard person is called one **olf**. The strength of most pollution sources indoors may be expressed as person equivalents, i.e. the number of standard persons, **olfs**, required to make the air as annoying (causing equally many dissatisfied) as the actual pollution source (Fanger (1988)).

One decipol is the perceived air quality in a space with a pollution source strength of one olf, ventilated by 10 l/s of clean air, i.e. 1 decipol = 0.1 olf/(l/s) (Fanger (1988)). Fig. 2.11 shows the relation between air quality expressed by the percentage of dissatisfied visitors and expressed in decipol.

More recently, extensive studies by Fanger (1993) have concentrated on the emission of odour from other sources. This has highlighted the need to consider the building itself as a polluter in addition to pollutants generated by occupants and occupant activities.

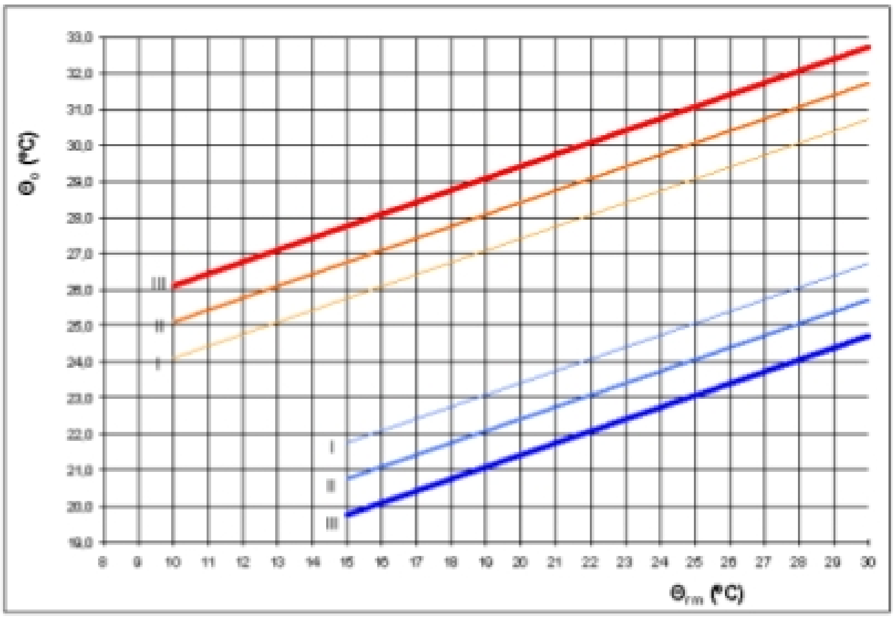

Figure 2.9: Design values for the indoor operative temperature for buildings without mechanical cooling systems as a function of the exponentially-weighted running mean of the outdoor temperature. Θ_{rm} = Outdoor running mean temperature. Θ_o = Operative temperature. EN 15251 (2007)

If the source of odour is from within the building and cannot be eliminated, then control must be by dilution with fresh air, this can result in an additional ventilation load.

Metabolic carbon dioxide

Carbon dioxide is produced as part of the metabolic process. The rate of emission of metabolic carbon dioxide is well defined and is a function of the level of activity. Typical production rates for various activities are summarised in section 4.3. While carbon dioxide, itself, is not harmful, the concentration of metabolically produced CO_2 correlates with metabolic odour intensity. It can thus act as a marker or surrogate to provide an indication of the adequacy of ventilation when occupants themselves represent the dominant source of pollutant.

Following the commencement of occupation in a room or building, the carbon dioxide concentration rises over time to an ‘equilibrium’ or ‘steady state’ value. Provided there are no other sources of CO_2 emission, the ventilation rate per occupant can be estimated from this steady state value. In principle, therefore, ventilation rate can be verified against the measured CO_2 value (Gameiro (2011)).

This characteristic of metabolic CO_2 forms the basis of carbon dioxide demand controlled ventilation systems. It is especially applicable to transiently and densely occupied buildings such as offices, schools and theatres. However, it is not appropriate in buildings in which other sources of pollutant dominate (e.g. tobacco smoke, moisture production,

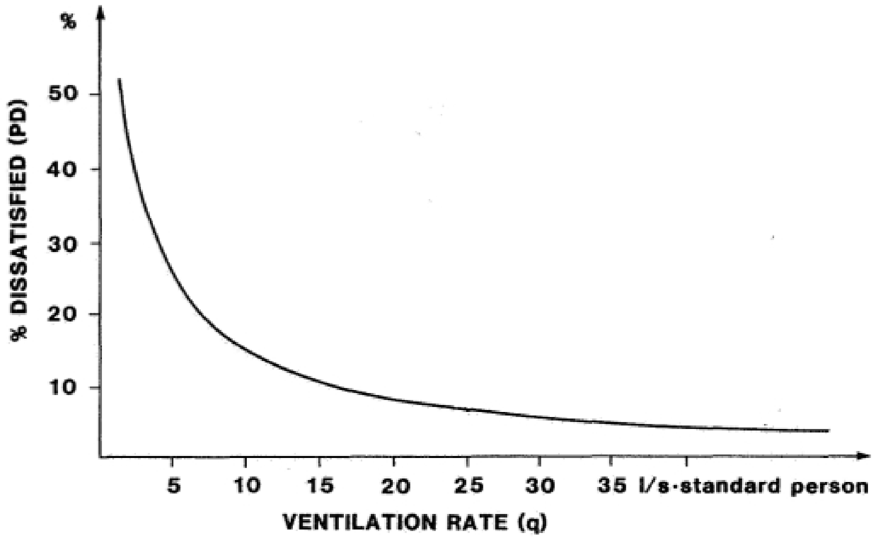

Figure 2.10: Dissatisfaction caused by a standard person (one olf) at different ventilation rates. The curve is based on European studies where 168 subjects judged air polluted by bioeffluents from more than one thousand sedentary men and women. Fanger (1988)

etc.). Neither, in large or sparsely occupied buildings in which the steady state CO_2 concentration may not be reached. As a rule, if the measured CO_2 concentration is found to be above a given target value, corresponding to the desired ventilation rate, it may be concluded that the rate of ventilation is inadequate. On the other hand, if the CO_2 concentration is found to be at or below the target value, the adequacy of ventilation is not necessarily confirmed, since it is possible that the steady state value has yet to be attained (Liddament (1996)). It is the difference between the indoor and outdoor carbon dioxide concentration that provides a measure of metabolic impact. However, threshold or target CO_2 concentrations are frequently based on an assumption that the ambient outside CO_2 value is approximately 350 to 400 ppm (Persily (1993)).

2.2.4 Indoor climate and productivity

There is a general discussion about the relevance and the difficulties for assessing the economic implications of varying the indoor climate specifications. Basically the types of cost impact are divided in:

- Effects at individual level (health, well being);
- Effects for the building owner (losing tenants,...);
- Effects at company level (absence, lower productivity,...);
- Effects for society.

The economic impact of poor IAQ

Many decisions on governmental energy policies are based on economical models, such as cost benefit analysis. However, it is difficult to frame the results of the social health and

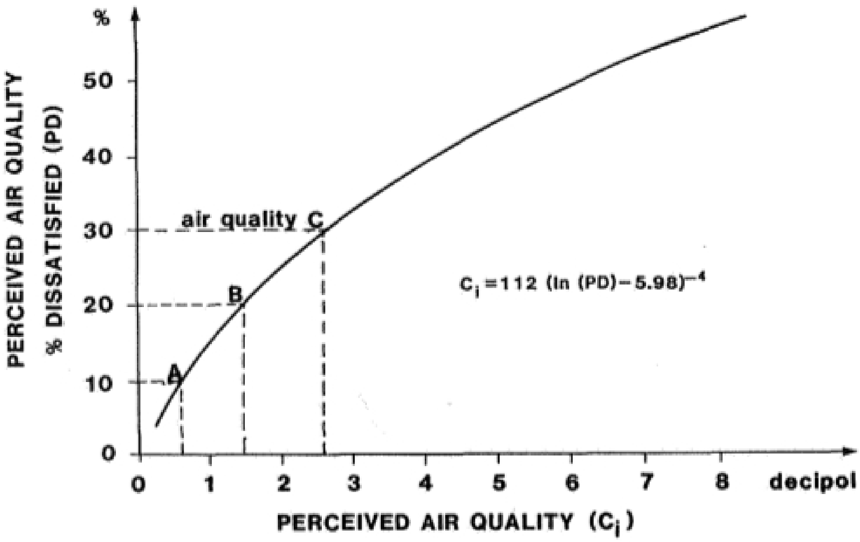

Figure 2.11: The relation between perceived air quality expressed by the percentage of dissatisfied and expressed in decipol. Fanger (1988)

behavioural studies of IAQ in ways that can be easily incorporate into these economical analyses. Such a calculation would need to consider the social costs for IAQ induced illness, direct medical costs and disabilities, as well as the loss of productivity and material and equipment damages. In addition to these, there are secondary costs related to discomfort and annoyance caused by deteriorated IAQ which appears, for example, as lowered real estate market values (WHO (1994)).

This makes it very difficult to carry out economical calculations on the total cost for the society at large exposed to deteriorated IAQ. The "total social cost" is a quantitative expression of the impact of deteriorated IAQ on economic activity, health and well being. Ideally, one would like to express the expenditure, inconvenience and drawbacks of deteriorated IAQ in a single, monetised figure, but as of today, no appropriate estimates have been available for the quantitative aspects of unsatisfactory indoor environment (Hanssen (1997)).

Brooks and Davis (1992) attributed to IAQ problems a productivity loss per-employee in the United States, an estimation of 3 percent (14 minutes/day) and 0.6 added sick day's annually. Thus from a profit and loss standpoint, remedial actions to improve IAQ where productivity is a concern are likely to be cost effective even if they require an expensive retrofit. In Norway, the authorities estimate the societal costs related to deteriorated IAQ are in the order of 1 to 1,5 billion €per year (Larsen (1991)), or about 250 to 350 €per inhabitant. This estimate includes costs related to adverse health effects requiring medical attention and does not include reduced working efficiency or job-related productivity losses. If this is representative of Europe, the total costs related to inadequate IAQ are immense. However, these figures only must be considered as approximations because the calculation methods are rather dubious and not very well developed (ECA 23 (2003)).

The effects of IAQ on employee performance and productivity at work

The relationship between indoor environment and a worker's health has been the subject of research for some years, but there has been comparatively little research which examines the total effect of IAQ on human well-being, employee performance and productivity at work. In order to assess healthy buildings in terms of performance and productivity, it is necessary to consider a variety of measures that might be used as indicators of an effect on productivity (Wargocki et al. (2006)).

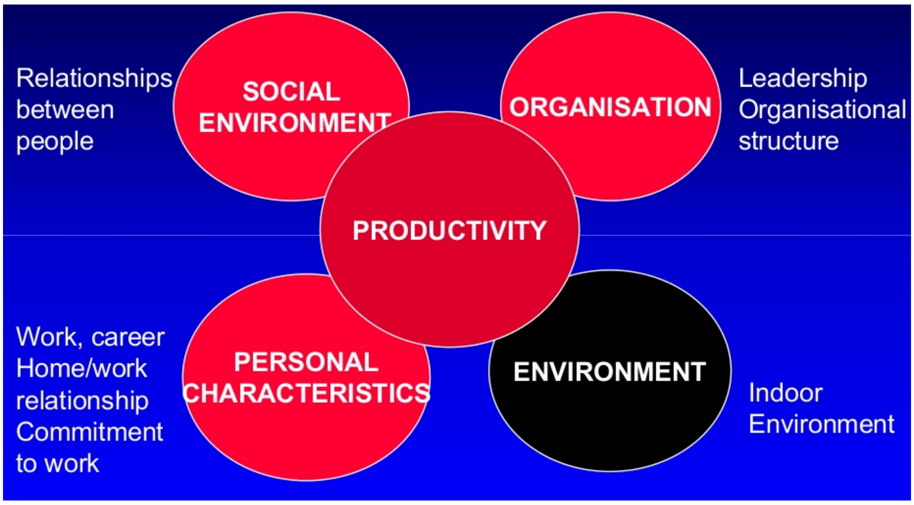

Figure 2.12: Factors affecting productivity. Wargocki et al. (2006)

Obviously, absence from work represents a 100 percent loss of productivity, but also bad or deteriorated IAQ may reduce performance at work in a substantial way. This, in turn, may result in a considerable loss of productivity and business outcome (Fisk and Rosenfeld (1997)), (Wargocki (1998)). Consequently, good indoor air quality is good business. Labour cost is estimated to be 10 to 100 times greater per square meter of office space than energy and other building operational costs (ECA 23 (2003)).

2.3 The energy impact of ventilation

Nowadays when we talk in energy consumption and energy impact, we talk about sustainability. The Brundtland Commission of the United Nations (UN (1987)) stated that development of the built environment is sustainable "...if it meets the needs of the present without compromising the ability of future generations to meet their own needs."

Sustainability is focused on the distant future (e.g., 30 to 50 years). Any actions taken under the name of sustainability must address the impact of present actions on conditions likely to prevail in that future time frame. In designing the built environment, the emphasis has often been on the present or the near future, usually in the form of capital or first-cost impact. As is apparent when life-cycle costing analysis is applied, capital cost assumes less importance the longer the future period under consideration.

This emphasis on the distant future can differentiate sustainable design from green design. Whereas green design addresses many of the same characteristics as sustainable design, it may also emphasize near-term impacts such as indoor environmental quality, operation and maintenance features, and meeting current client needs. Thus, green design may focus more on the immediate future (i.e., starting when the building is first constructed and then occupied). Sustainable design is of paramount importance to the global environment in the long-term while still incorporating features of green design that focus on the present and near future (ASHRAE (2009)).

The urgent need to reduce the energy used in the heating, cooling and lighting of buildings has brought about significant changes in the way buildings are designed and built. Because buildings encompass complex interactions between different requirements and subsystems, care must be taken when changes are made to any one aspect of building performance to ensure that this does not have a detrimental effect on other building functions. This is particularly true when measures are introduced to reduce energy consumption in buildings. It is important to ensure that when energy saving measures are taken, their effects on the indoor environment are identified and quantified, and that action is taken to eliminate or reduce any detrimental effects. The main strategies for improving the use of energy in buildings and/or the indoor environment concern (ECA 17 (1996)):

- building location and orientation
- building design and construction
- building services systems
- control of pollution sources
- building operation and maintenance

ECA 17 (1996) collates the principal ways of implementing strategies to improve energy performance and how they can affect the indoor environment.

The main conclusions we can reach are:

- Both the rational use of energy and the provision of good IAQ are important aspects of building design and refurbishment. There are potential conflicts between these requirements. The impact of possible energy saving measures on IAQ should always be assessed before their eventual adoption, and where unacceptable, the measure should be avoided.
- The first priority should be control de sources of bad IAQ in order to mitigate the ventilation rates.
- Environmental tobacco smoke is a key pollutant, with serious energy penalties because of substantial increases in ventilation requirements. The Lei 37/2007 adopted rules to protect citizens from exposure involuntary tobacco smoke and reduction measures demand associated with dependence and cessation consumption and the subsequent improvement of IAQ in the non-smoking spaces.

- The importance of occupants should be recognised, particularly their role in ensuring that energy and IAQ systems operate correctly, their ability to act appropriately in the event of failure, and their needs for individual control.

- Reducing air exchange rates to save energy may result in poor IAQ.

- Increasing air exchange rates to improve IAQ will increase energy consumption, unless this is partly compensated for by heat recovery (see section 2.6).

- Ventilation requirements should be determined from the total pollutant load in a building resulting from constituents and systems and of occupants and their activities. The goal of ventilation should be to provide good IAQ. A secondary goal is to protect the building, the installations and furnishings.

2.4 Design criteria

Designing for energy efficient and reliable ventilation extends beyond system sizing. System performance is influenced by a vast range of other parameters covering climate, building type, construction, air-tightness and ventilation strategy. Acceptability by occupants, ease of use, reliability and noise performance are also important aspects of the design process.

Good ventilation design is essential to ensure the reliable provision of fresh air to building occupants. In particular, ventilation design should satisfy the following basic requirements (Liddament (1996)):

- comply with relevant building regulations and associated standards and codes of practice

- satisfy minimum ventilation rates for optimum health and comfort

- be capable of removing pollutants at source before they disperse into occupied areas

- be compatible with the building in which the system is installed

- provide high rates of ventilation for cooling purposes or for rapidly purging polluted air from a building

- incorporate occupant or automatic controls to ensure that the ventilation rate can be adjusted to meet changing demand

- be reliable

- be capable of being cleaned and maintained

- comply with smoke and fire control requirements

- be cost and energy efficient

Since such a wide range of parameters is involved, there is rarely a unique solution to a particular ventilation design. Instead judgement must be based on the specific needs of each building.

As we can see in fig. 2.13 there are essentially 3 design parameters: ventilation need, design constraints and design variables.

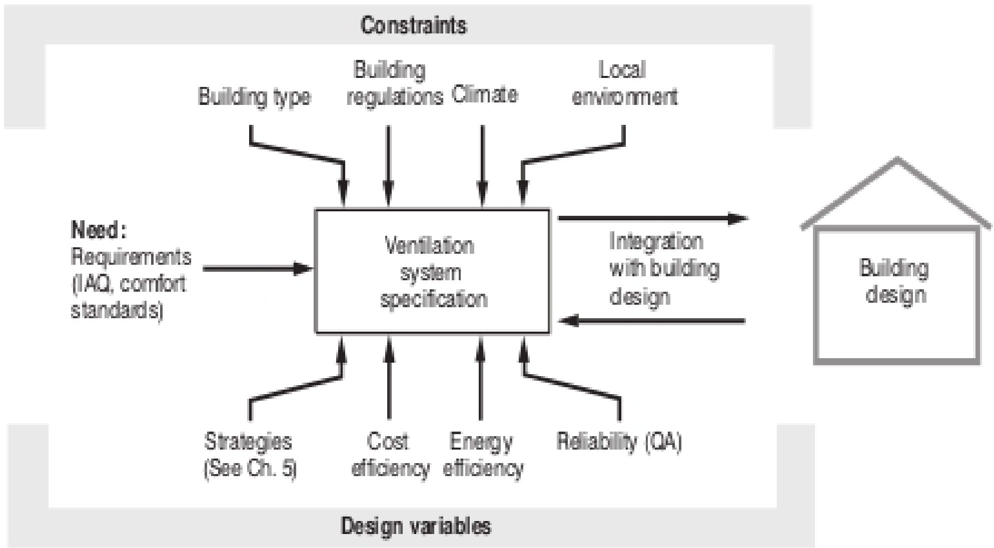

Figure 2.13: Essential Design Parameters. Liddament (1996)

2.4.1 Ventilation need

This parameter is intended to calculate the flow rate necessary to provide a good IAQ. An essential aspect of the design process is to identify how much fresh air is to be provided to a space. Invariably needs will change according to occupancy pattern, pollution emission rates and seasonal changes in climate, therefore provision to control the rate of ventilation to meet prevailing demand is usually necessary. Typically, design may be based on a minimum need, as set by the relevant regulations or code of practice, with additional amounts being based on specific pollutant problems and any requirement for ventilation cooling (Liddament (1996)). Guidelines for calculating the rate of ventilation needed to control pollutant concentration by dilution are presented in section 2.12.

2.4.2 Design Constraints

Compliance with Regulations and Standards

Many countries have introduced ventilation related regulations, standards and codes of practice (Limb (1994) and Limb (2001)). To fulfil the needs of best practice, it is important that these requirements and recommendations are followed. In the United States an guidance is regularly produced and updated as part of ANSI/ASHRAE 62.1 (2007). Within the European Union there are a wide range of standards. In Portugal also the ventilation requirements have regulations and standards. This requirements are 'prescriptive' in the sense that the minimum rate of ventilation (RSECE (2006)) or the minimum size of ventilation openings is specified (NP 1037 (2001)). Air flow rates are indicated for different types of room, occupant density or activity. Additional 'air quality' requirements relate the amount of extra ventilation needed to deal with individual contaminant sources is present. Regulations and standards include the follow topics:

Health: Requirements cover the minimum ventilation needed to avoid injury to health. Values are largely prescribed according to building type, nature of pollutants, emission rates and acceptable exposure levels. The RSECE (2006) requires that the principal pollutants of buildings are maintained under the levels of fig. 2.14. Besides these pollutants is also required to evaluate: microorganisms; bacteria; fungi, legionella and radon. The method of measuring these pollutants is presented on NT-SCE-02 published by ADENE.

Parâmetros	Concentração máxima de referência (mg/m3)
Partículas suspensas no ar (PM10)	0,15
Dióxido de carbono	1800
Monóxido de carbono	12,5
Ozono	0,2
Formaldeído	0,1
Compostos orgânicos voláteis totais	0,6

Figure 2.14: Maximum reference concentrations of pollutants within the existing buildings. RSECE (2006)

Energy efficiency: Standards cover the avoidance of excessive energy waste. In some cases there are a requirement for ventilation heat recovery. According to RSECE (2006) is mandatory the recourse to energy recovery of rejection air, in the heating season, with a minimum efficiency of 50%, where the thermal rejection at design conditions is more than 80 kW.

Comfort: Requirements or recommendations may cover thermal comfort, odour intensity and the presence of draughts (see section 2.2.3).

Ventilation strategies: Standards often cover the type of ventilation appropriate to specific applications (e.g. enclosing polluting processes, extracting from kitchens and bathrooms, provision of fresh air supply to occupied spaces and the sizing of ventilation systems).

Air-tightness: Energy efficient ventilation performance can be destroyed if the air-tightness of the structure is not compatible with ventilation strategy. Several countries have now introduced standards or recommendations covering the air-tightness performance of buildings (Liddament (1996)). Similarly, various standards cover the air-tightness, durability and performance of the various components used in building construction (e.g. the performance windows, doors, sealants and sealing components).

In Portugal up until 2006, there was no check on the quality of the ductwork (most often, building owners did not require the check simply to avoid its cost), and its performance was in general quite poor (high leakage, cheap materials used), resulting in significant losses, with important negative consequences in terms of the energy efficiency of the whole installation (more air had to be circulated and treated to compensate for the leakage). Moreover, it was often impossible to meet the minimum fresh air rates in many spaces, resulting in degraded IAQ levels (Maldonado and Brito (2013)). To comply with the Portuguese regulation (RSECE (2006)), ductwork leakage of air conditioning installations of buildings larger than 1000 m^2 may not exceed 1.5 $l/s.m^2$ under a static pressure of 400 Pa (Class A limit according to EN 12237 (2003) is 1.32 $l/s.m^2$ at 400 Pa).

Building type

Ventilation need varies according to building type. Typical requirements for the main building sectors in Portugal include:

Dwellings and small service buildings: There are no requirements for ventilation.

Large service buildings: The requirements are given by RSECE (2006).

Climate

The amount of energy needed to heat or cool air to comfort levels is dependent on the severity of climate. Thus climate has a significant impact on the choice of strategy, especially in relation to cost and complexity. For ventilation purposes, climate can be classified in terms of mild, moderate and severe (both heating and cooling). Actually in Portugal there are no requirements for ventilation according to the climate.

Local environment

The local outdoor environment influences ventilation planning. Important categories include:

- Heavily industrialised and inner city locations
- Adjacent buildings
- Suburban areas
- Rural areas

2.4.3 Design variables

The remaining parameters are those over which the designer has control. These are the ventilation system itself and the construction characteristics of the building.

Strategy

The main design variable is that of the ventilation strategy itself (see section 2.5). In specifying a strategy, it is important to review the constraints imposed by building type, climate and location, the level of air-tightness to be accomplished, as well as cost performance, energy performance, reliability and ease of maintenance.

Cost efficiency

System costs ultimately fall on the building owner or occupier. Therefore, to be widely accepted, the cost performance of the system must be competitive. Where alternative strategies are feasible, a comparative payback period may be defined such that, over a given period of time.

Energy efficiency

Optimum design must ensure that any unnecessary loss of conditioned air is minimised. This is achieved by good building air-tightness and the elimination of avoidable pollutants to reduce the need for ventilation. Distribution energy is controlled by careful fan selection and good design and routing of ductwork. Design should satisfy ventilation need with minimum use of energy.

Reliability and ease of maintenance

The system should provide the desired air flow rate with comfort and be acceptable to occupants (e.g. easy to use and difficult to misuse). Practice shows that the performances of a ventilation system can substantially change during its lifetime (ECA 23 (2003)). Therefore, without appropriate maintenance and control, it is not evident to assume a correct functioning. Good reliability, ease of maintenance and extended operational life are extremely important to assure the success of ventilation design.

2.5 Ventilation strategies

It is common to classify ventilation systems in 2 categories:

- Natural ventilation
- Mechanical ventilation

A third system is the hybrid ventilation, which intents to combine the features of natural and mechanical ventilation.

Various configurations of mechanical ventilation are in use:

- **Mechanical extract ventilation:** A fan is used to mechanically remove air from a space. This induces a 'suction' or 'under' pressure which promotes the flow of an equal mass of 'make-up' or 'fresh' air into the space through purpose provided air inlets or infiltration openings.

- **Mechanical supply ventilation:** Supply (outdoor air) is mechanically introduced into the building where it mixes with the existing air. This process induces a positive (i.e. above atmospheric) pressure in the building. Indoor air is displaced through purpose provided and/or infiltration openings.

- **Mechanical balanced 'mixing' ventilation:** Combines extract and supply systems as separately ducted networks. An air flow pattern is established between the supply areas to the extract areas.

- **Mechanical balanced 'displacement' ventilation:** Displacement ventilation is a form of balanced ventilation in which the supply air 'displaces' rather than mixes with the room air. Gravitational effects encourage the incoming air to creep at floor level until it reaches a thermal source (occupant, electrical load, etc.). The air then rises around the heat source and into the breathing zone prior to extraction at ceiling level.

- **Demand Controlled Ventilation (DCV):** Demand controlled ventilation (DCV) systems provide a means by which the rate of ventilation is automatically controlled in response to variations in indoor air quality. Ventilation is therefore provided only when and where it is needed while, at other times, it may be reduced to minimise space heating or cooling losses. Essentially, a 'sensor' is used to track indoor air quality and to modulate the rate of ventilation to ensure air quality does not deteriorate.

2.5.1 Space air diffusion

Room air distribution systems are intended to provide thermal comfort and ventilation for space occupants and processes.

Room air diffusion methods can be classified as one of the following (ASHRAE (2009)):

- **Mixed systems** produce little or no thermal stratification of air within the space. Overhead air distribution is an example of this type of system (see fig. 2.15).

- **Fully (thermally) stratified systems** produce little or no mixing of air within the occupied space. Thermal displacement ventilation is an example of this type of system (see fig. 2.16).

- **Partially mixed systems** provide some mixing within the occupied and/or process space while creating stratified conditions in the volume above. Most underfloor air distribution designs are examples of this type of system (see fig. 2.17).

- **Task/ambient conditioning systems** focus on conditioning only a certain portion of the space for thermal comfort and/or process control. Examples of task/ambient systems are personally controlled desk outlets (sometimes referred to as personal ventilation systems) and spot-conditioning systems (see fig. 2.18).

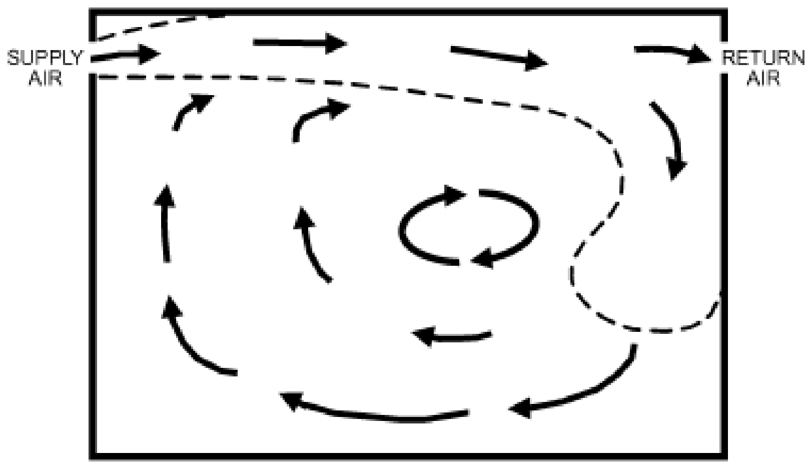

Figure 2.15: Entrainment Flow Within a Space. ASHRAE (2009)

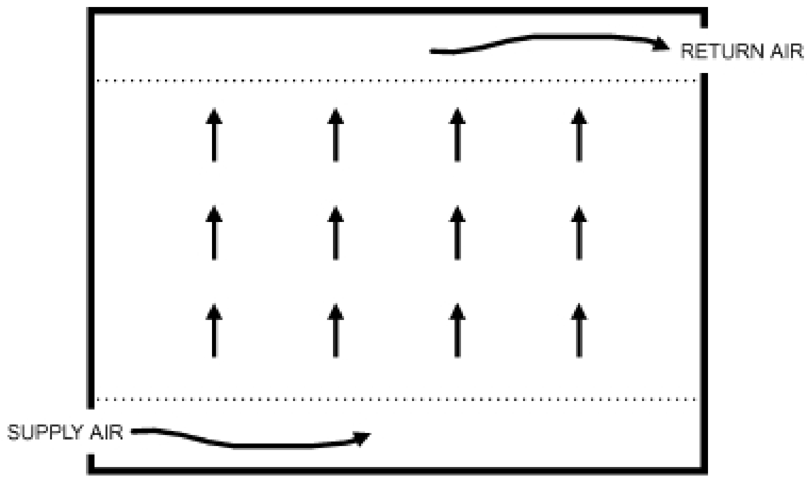

Figure 2.16: Displacement Flow Within a Space. ASHRAE (2009)

2.6 Ventilation heat recovery

Ventilation heat recovery is the process by which thermal energy is recovered from exhaust air for reuse within the building. The benefit of heat recovery improves as the climate becomes more severe, since the operating cost is majority dependent on the capacity of the system.

A heat recovery unit transfers heat (some units also moisture) from the exhaust air stream over to the supply air stream, thus reducing the heat loss due to ventilation, and reducing the need to condition the cold supply air. The choice of the most suitable system to be installed depends of the ventilation design criteria.

The principal methods used for ventilation heat recovery are:

- air-to-air heat recovery

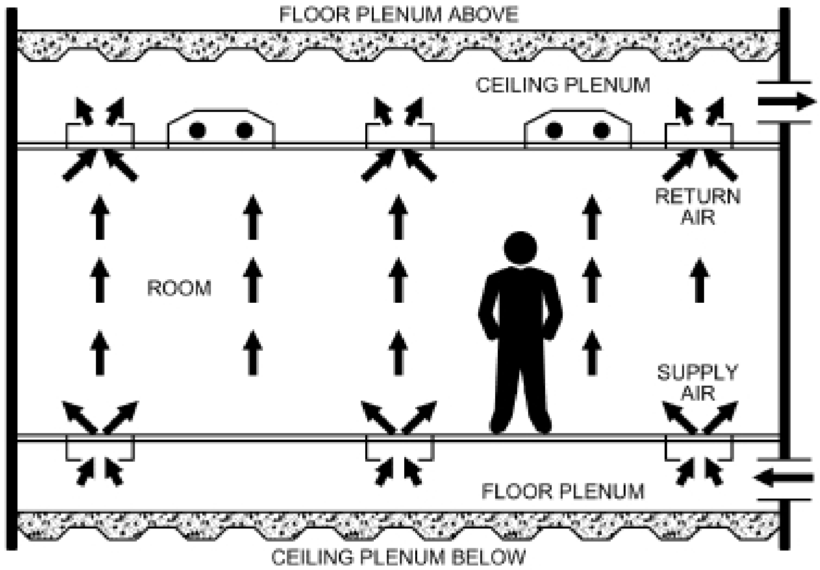

Figure 2.17: Underfloor Air Distribution to Occupied Space Above. Rock and Zhu (2002)

- flue gas heat recovery
- exhaust air heat pumps
- combined air-to-air heat recovery with heat pumps
- dynamic insulation
- preconditioning of supply air using buried or 'ground' ducting.

The most use is the air-to-air heat recovery, because of the relatively low initial cost and the capacity of offering high efficiency. There are two main types of heat recovery unit: regenerative and recuperative.

Regenerative heat exchangers transfer heat via heat-accumulating surfaces that are repeatedly exposed to either the exhaust air or supply air stream. The heat-accumulating surfaces are normally metal. These heat exchangers can also recover moisture. Undesirable leakage between the two air streams can occur, though this problem can be reduced by judiciously locating the fans to counteract the leakage. We can see an example of a regenerative rotary heat exchanger in fig. 2.19.

Recuperative heat exchangers transfer heat across a dividing plate by means of thermal conduction (plate or tube heat exchanger), or with an intermediate fluid (run-around heat exchanger, heat pipe or heat pump). Since the two air streams are kept separate, these exchangers can theoretically have zero transfer of odours. We can see an example of a recuperative cross-flow plate heat exchanger in fig. 2.20.

Schild (2004) gives us a comparison of different types of heat recovery system for balanced ventilation.

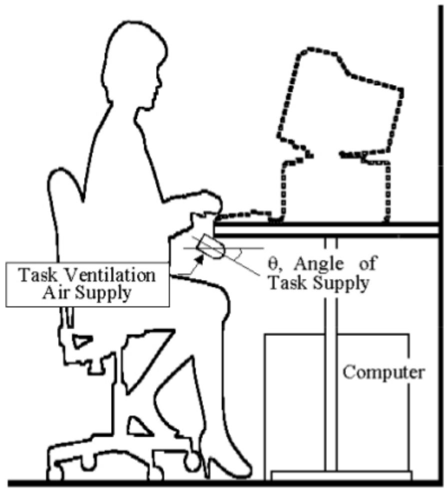

Figure 2.18: Side view of a task ventilation. Faulkner et al. (2002)

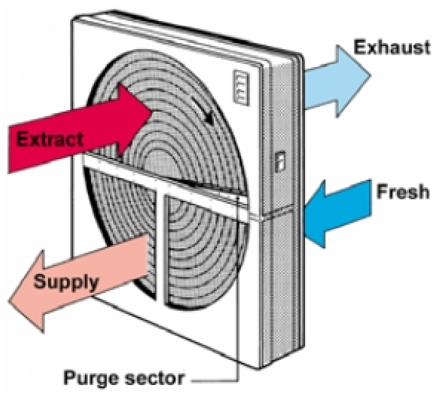

Figure 2.19: Rotary heat exchanger. Schild (2004)

2.7 Air cleaning by filtration

Filtration is a method by which particulates and, sometimes, gaseous pollutants may be removed from the air by passing the contaminated air through a medium. The filter intercepts the pollutant while allowing clean air to pass through. This method of air cleaning is especially necessary when high concentrations of particulates are present or when the outside air is contaminated.

As we can see earlier the level of metabolic CO_2 present in a room is good indicator of the IAQ. EN 13779 (2007) defines levels of indoor air quality for spaces with occupancy due to the concentration of CO_2. This standard classified the outdoor air (ODA) in 3 levels (see table 2.1).

Table 2.2 gives the category of indoor air (IDA) according with the CO_2 level above level of outdoor air.

In Portugal as we can see earlier in fig. 2.14, the ventilation systems have to ensure

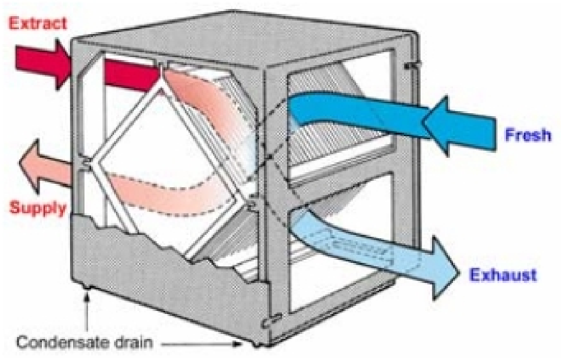

Figure 2.20: Cross-flow plate heat exchanger. Schild (2004)

Category	Description
ODA 1	Pure air which may be only temporarily dusty (e.g. pollen)
ODA 2	air with high concentrations of particulate matter and/or gaseous pollutants
ODA 3	air with very high concentrations of gaseous pollutants and/or particulate

Table 2.1: Classification of outdoor air (ODA). EN 13779 (2007)

that the limits indicated in RSECE (2006) are not exceeded, therefore the lowest authorized category of indoor air is IDA2.

The regulations relating to ventilation are prepared on the assumption that outdoor air is clean and healthy. However, as a result of urban pollution, the outside air is not always a source of clean air suitable for diluting pollutants within the buildings. Therefore, ventilation systems should be carefully designed to minimize potential contamination of the air intakes abroad and, in addition, consider filtering levels appropriate to promote the introduction of new air quality (Sarmento (2012)).

The appropriate level of filtering outside air depends on:

- outdoor air quality
- indoor air quality required

CO_2-level above level of outdoor air in ppm			
Category	Description	Typical range	Default value
IDA 1	High indoor air quality	<400	350
IDA 2	Medium indoor air quality	400 - 600	500
IDA 3	Moderate indoor air quality	600 - 1000	800
IDA 4	Low indoor air quality	>1000	1200

Table 2.2: Basic classification of indoor air quality (IDA). EN 13779 (2007)

- but also the specific use of the building and the degree of protection of equipment and systems

The EN 779 (2012) defines efficiency classes of air filters. There are also special air filters for high efficiency applications specific, particularly in hospitals, laboratories, museums, industries pharmaceutical, electronics and nuclear, etc.. The classification of this filters are given by EN 1822 (2009).

Ultraviolet light (UVC) has been used in HVAC hospital installations as a germicide. There are also filters for gaseous pollutants as is the case of the active carbon filters (CF), in particular to control odors (VOC's) but also other gaseous pollutants (SO_2, NO, NO_2, O_3). The process of filtering is based on adsorption (gas molecules adhere to the surface of the filter material) (Sarmento (2012)).

The EN 13779 (2007) recommends the use of air filters depending on the level indoor air quality and desirable outdoor air quality recorded (see table 2.3).

Outdoor	Indoor air quality			
air quality	IDA 1(high)	IDA 2 (medium)	IDA 3 (moderate)	IDA 4 (low)
ODA 1	F9	F8	F7	F5
ODA 2	F7 + F9	F6 + F8	F5 + F7	F5 + F6
ODA 3	F7 + GF + F9	F7 + GF + F9	F5 + F7	F5 + F6

Table 2.3: Recommended minimum filter classes per filter section. EN 13779 (2007)

Despite the investment and operational costs associated with filtering the outside air, it becomes essential its use for the maintenance of the conditions of IAQ, under penalty of loss of productivity as we saw in section 2.2.4.

2.8 Ventilation efficiency

Ventilation efficiency may be regarded as a series of indices or parameters which characterise the mixing behaviour of air and the distribution of contaminants within a space. During the previous decades, researchers defined and evaluated more than a dozen different air quality indicators without comparative guidelines for ventilation designers, resulting in confusion to a certain degree. This indices can be divided into 2 types:

- Indices representing the ability of a system to exchange the air in the room
- Indices representing the ability of a system to remove air-borne contaminants

Nevertheless, the four most widely used indicators of indoor air quality are (Novoselac and Srebric (2003)):

- Number of air changes (n_{AC})
- Contaminant removal effectiveness (ε^c)

- Zone Air Distribution Effectiveness (E_z)
- Air change efficiency (ε^a)

The values of ventilation efficiency parameters are normally determined by measurement (see section 2.9). This can be restrictive, since the flow and pollutant fields are unique to each enclosed space. The evolution of computational fluid dynamics for flow field analysis provides an opportunity to apply ventilation efficiency concepts at the design stage. However, further validation and boundary data are needed before these CFD methods can be more generally applied.

Some indices are based on room averaged values, while others refer to conditions at specific points or locations within the space. Room values provide guidance to the overall performance of a ventilation system while point values are necessary to indicate regions where ventilation might be inadequate. The concepts of ventilation efficiency may be applied to entire buildings, single zones or locations within a single zone (Liddament (1996)).

2.8.1 Number of Air Changes

The number of air changes in a space per unit of time (n_{AC}) also known as air exchange (or change) rate (I) is widely used to provide information about intensity of ventilation. It compares airflow to volume and is:

$$I = Q/V \tag{2.1}$$

Q = volumetric flow rate of air into space, m^3/s
V = interior volume of space, m^3

The air exchange rate has units of 1/time, usually h^{-1}. When the time unit is hours, the air exchange rate is also called air changes per hour (ach). The air exchange rate may be defined for several different situations. For example, the air exchange rate for an entire building or thermal zone served by an air-handling unit compares the amount of outside air brought into the building or zone to the total interior volume. This nominal air exchange rate I_N is:

$$I_N = Q_{oa}/V \tag{2.2}$$

where Q_{oa} is the outdoor airflow rate including ventilation and infiltration. The nominal air exchange rate describes the outside air ventilation rate entering a building or zone. It does not describe recirculation or the distribution of the ventilation air to each space within a building or zone.

Another indicator used to describe ventilation and infiltration is the time constant τ, which have units of time (usually in hours or seconds). One time constant is the time required for one air change in a building, zone, or space if ideal displacement flow existed. It is the inverse of the air exchange rate:

$$\tau = 1/I = V/Q \tag{2.3}$$

The nominal time constant compares the interior volume of a building or zone to the volumetric outdoor airflow rate:

$$\tau_n = V/Q_{oa} \tag{2.4}$$

The n_{AC} value does not provide information on the quality of the fresh air distribution or contaminant removal from the space. Therefore, the number of air changes provides incomplete information on perceived air quality.

In Portugal the n_{AC} is used to quickly check if a space meets the requirement of having air velocities less than 0,2 m/s in the occupied zone (ADENE (2011)).

2.8.2 Contaminant Removal Effectiveness

One of the first indicators that actually define a perceived air quality is the contaminant removal effectiveness (ε^c) (Yaglou and Witheridge (1937)). The ε^c is a measure of how quickly an air-borne contaminant is removed from the room. This indicator is based on the room average contaminant concentration $< C >$, the contaminant concentration at supply C_s, and the contaminant concentration at exhaust C_e:

$$\varepsilon^c = \frac{C_e - C_s}{< C > - C_s} \tag{2.5}$$

Mundt et al. (2003) and Brouns and Waters (1991) shows that can be use net concentrations in preference to absolute concentrations with no loss in generality. This means that all concentrations values are taken as values above the value in the outside air or supply duct C_s. The use of volumetric measures of air and contaminant imply that temperature and pressure are constant throughout the ventilation system. The errors caused by this assumption are sufficiently small to be ignored in the majority of practical cases. The equation 2.5 becomes:

$$\varepsilon^c = \frac{C_e}{< C >} \tag{2.6}$$

It should be noted that the theory and definitions described refer to an air tight room where all the air enters and leaves via single inlet and exhaust ducts. However, these ducts may be taken as the summation of all possible inlet and outlet paths which means that the theory and definitions are applicable to any room, regardless of the method of ventilation.

Nominal time constant for the contaminant

The time constant for the contaminant (τ_n^c) is also called the turnover time of the contaminant, or the transit time for the contaminant flow through the room (Brouns and Waters (1991)). The nominal time constant for the contaminant is defined as the ratio between the equivalent volume of contaminant in the room and the contaminant injection rate.

$$\tau_n^c = \frac{V_c}{q} \tag{2.7}$$

V_c =equivalent volume of contaminant in the room, m^3
q = contaminant injection rate, m^3/s

V_c is defined by the expression:

$$V_c = < C > .V \tag{2.8}$$

The nominal time constant for the contaminant may also be defined as the average time it takes for the contaminant to flow from its source to the exhaust duct.

As we can see in Brouns and Waters (1991) and Mundt et al. (2003) the ε^c may also be defined as the ratio between the nominal time constant for the ventilation air and the nominal time constant for the contaminant.

$$\varepsilon^c = \frac{\tau_n}{\tau_n^c} \tag{2.9}$$

The ventilation effectiveness depends on the air distribution and the location of the pollution sources in the space. It may, therefore, have different values for different pollutants.

In certain situations it may be useful to have an indicator that reflects the ventilation effectiveness in a point. The local Air quality Index (ε_p^c) is defined as the ratio between the steady state concentration of contaminant at the exhaust duct and the steady state concentration of contaminant at a point p in the room.

$$\varepsilon_p^c = \frac{C_e}{C_p} \tag{2.10}$$

To estimate the ε^c, it is often useful to divide a space into two zones. One is the air supply zone and the other zone comprises the rest of the room. In mixing ventilation the supply zone is usually above the breathing zone. If there is complete mixing of air and pollutants, the ventilation effectiveness is one. If the air quality in the breathing zone is better than in the exhaust, the ventilation effectiveness is higher than one, and the desired air quality in the breathing zone can be achieved with a lower ventilation rate.

If the air quality in the breathing zone is poorer than in the exhaust air, the ventilation effectiveness is lower than one and more ventilation is required (ECA 11 (1992)).

The contaminant removal effectiveness is a function of location and characteristics of air terminal devices and of pollution sources (see fig 2.21). It is furthermore a function of temperature and flow rate of the supply air. The ε^c may be calculated by numerical simulation or measured experimentally. When such data are not available, there are several sources with typical values. In fig. 2.22 we can see typical values of contaminant removal effectiveness.

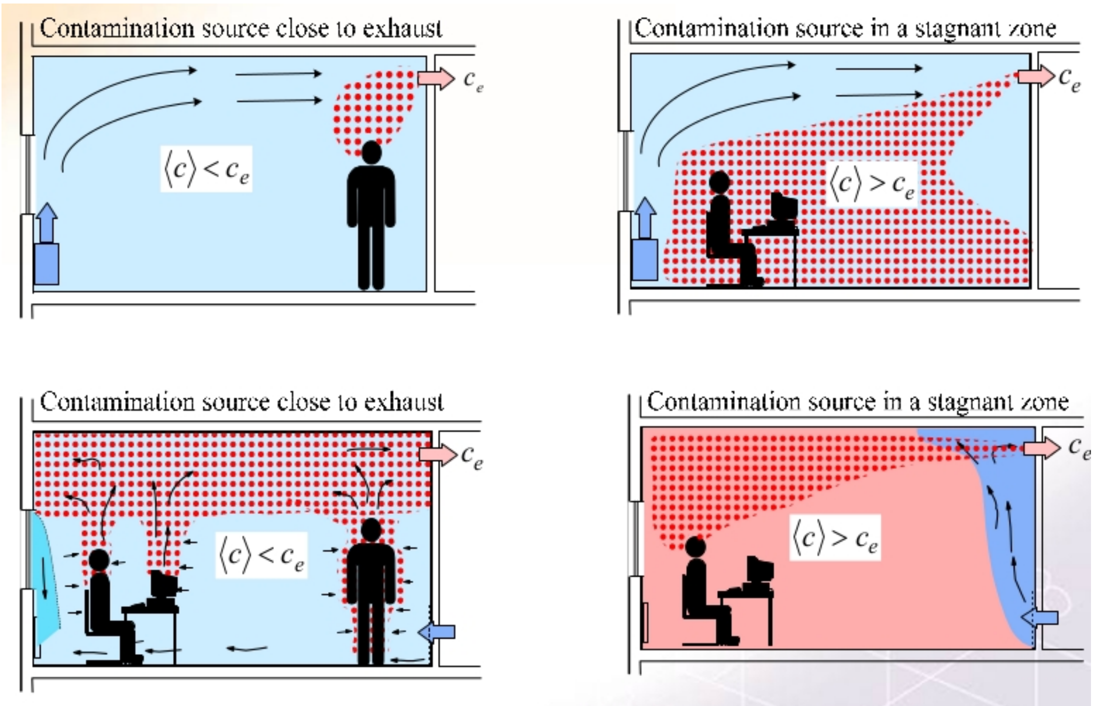

Figure 2.21: Steady state concentration in the exhaust and steady state mean concentration in a room in four different situations. Mundt et al. (2003)

Method of measurement

In order to measure any of the indices, it is necessary to measure the concentration of the contaminant at the appropriate points in the room. If it is not possible to measure the contaminant itself, then a tracer gas which imitates the behaviour of the contaminant may be used instead (see section 2.9).

The indices may be obtained either by measuring the equilibrium concentration due to continuous injection of contaminant at a constant rate, or by monitoring contaminant history due to different methods of contaminant injection (see section 3.3.2).

2.8.3 Air Change Efficiency

Air change efficiency (ε^a) is a measure of how effectively the air present in a room is replaced by fresh air from the ventilation system (Sandberg and Skaret (1985)).

Air Diffusion	Cold jet $\Delta\theta < 0$K		Hot jet		
	Effective velocity	Ventilation effectiveness	$\Delta\theta$ (supply-indoor)	Low ceiling	High ceiling
Mixing Horizontal jet	> 1,5 m/s	0,9 – 1,1	< 10 °C	0,8 - 1	Not advised
	< 0,5 m/s	0,7 – 0,9	> 15 or 20 °	0,4 – 0,8	Not advised
Mixing Vertical jet	All diffusers	0,9 – 1,1	< 10 °C	0,6 – 0,8	0,8 – 1 [a]
			> 15 °C	0,4 – 0,8	
Displacement ventilation		1,0 - 2		0,2 – 0,7	Not advised

[a] applying this value intends that the diffusers used are powered geometry or swirling. If fixed geometry diffusers are used, it's restricted to heating only (no cooling) and appropriate and careful selection taking into account $\Delta\theta$.

Figure 2.22: Typical values for CRE. EN 13779 (2007)

As in the ε^c it should be noted that the theory and definitions described refer to an air tight room where all the air enters and leaves via single inlet and exhaust ducts. However, these ducts may be taken as the summation of all possible inlet and outlet paths which means that the theory and definitions are applicable to any room, regardless of the method of ventilation Sutcliffe (1990).

Age of air

The age of air concept was introduced by Sandberg (1981) and has proven to be a useful tool in evaluating ventilation effectiveness. The age of air is the length of time t that some quantity of outside air has been in a building, zone, or space. Units are of time, usually in seconds or minutes, so it is not a true efficiency or effectiveness measure. The age of air concept, however, has gained wide acceptance in Europe and is used increasingly in North America ASHRAE (2009).

The mean age of air is a statistical concept based on the age distribution of the air components in a point. The age is counted from the time the air enters the room (see fig. 2.23). The air in a point is a mixture of components that has spent different time in the room during which they have been contaminated. The local mean age of air ($\bar{\tau}_p$) is defined as the average time it takes for air to travel from the inlet to any point p in the room. In a fully mixed situation the local mean age of air will be the same in the whole room. If there is a shortcut from supply device to the exhaust device the local mean age of air will be low in the shortcut zone and high in the stagnant zone.

The average age of all air present in the room, the room mean age of air ($< \bar{\tau} >$) is equal to the spatial average of the local mean ages of air $\bar{\tau}_p$.

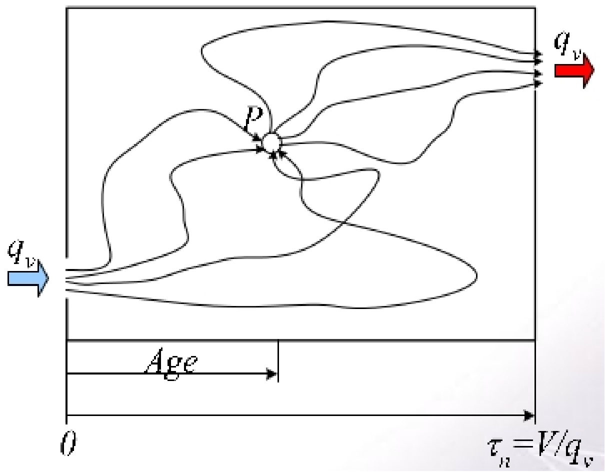

Figure 2.23: Age of air concept. Mundt et al. (2003)

Air change time

The air change time ($\bar{\tau}_r$) for all the air in the room is equal to twice the room mean age of all the air in the room:

$$\bar{\tau}_r = 2. < \bar{\tau} > \tag{2.11}$$

As we can see in Mundt et al. (2003), normally to facilitate the understanding of these different ages, a comparison with a human population is often made. The mean age of air leaving the room can be compared to the mean age at the time of death and the mean age of the air in the room compared to the mean age of a population living at any time. The birth rate, i.e. the air flow rate entering the room is constant.

Air Change Efficiency

The local mean age of air leaving the room, always equal to τ_n is theoretically equal to the shortest possible air change time for the air in the room, only valid for piston flow (see fig 2.24). The actual air change time is directly related to the room mean air age $< \bar{\tau} >$.

The air change efficiency is defined as the ratio between the shortest possible air change time for the air in the room, the nominal time constant, τ_n, and the actual air change time $\bar{\tau}_r$.

$$\varepsilon^a = \frac{\tau_n}{\bar{\tau}_r} = \frac{\tau_n}{2. < \bar{\tau} >} \tag{2.12}$$

The air change efficiency is a function of location and characteristics of air terminal devices and of pollution sources (see fig 2.24).

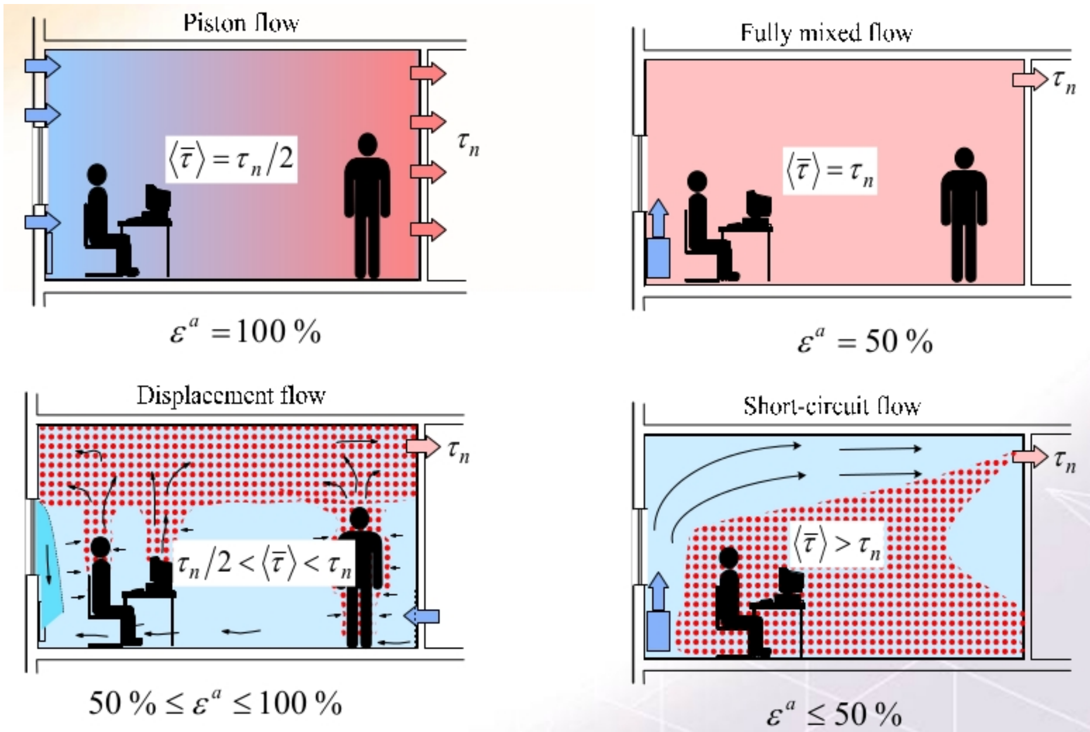

Figure 2.24: Room mean age of air and local mean age of air in the exhaust for four different situations. Mundt et al. (2003)

Coefficient of Air Change Performance

The coefficient of air change performance (η) is defined as the ratio between the nominal time constant and the room mean age:

$$\eta = \frac{\tau_n}{<\bar{\tau}>} \tag{2.13}$$

It should be noted that the coefficient of air change performance equals twice the air change efficiency:

$$\eta = 2.\varepsilon^a \tag{2.14}$$

This term is equivalent to the ventilation effectiveness definition of ANSI/ASHRAE 62.1 (2007).

Local Air Change Index

This index characterises the conditions at a particular point and may be large due to the position in the room of the measurement point.

$$\varepsilon_p^a = \frac{\tau_n}{\bar{\tau}_p} \tag{2.15}$$

Method of measurement

In order to measure any of the indices, it is necessary to measure the concentration of the contaminant at the appropriate points in the room. If it is not possible to measure the contaminant itself, then a tracer gas which imitates the behaviour of the contaminant may be used instead (see section 2.9).

The indices may be obtained either by measuring the equilibrium concentration due to continuous injection of contaminant at a constant rate, or by monitoring contaminant history due to different methods of contaminant injection (see section 3.3.1).

2.8.4 Zone Air Distribution Effectiveness

According with ASHRAE (2009), the HVAC design engineer usually does not have knowledge or control of actual pollutant sources within buildings. For most projects, therefore, air change effectiveness is of more relevance to HVAC system design than ventilation effectiveness. Of the various indicators and definitions proposed for air change effectiveness, ASHRAE refers to ANSI/ASHRAE 62.1 (2007) zone air distribution effectiveness (E_z) as the value to use. As mentioned above this term is equivalent to the coefficient of air change performance (η).

In the fig. 2.25 we can see the typical values indicated for the E_z. As an alternative to using the values, E_z may be regarded as equal to air change effectiveness determined in accordance with ANSI/ASHRAE 129 (1997) for all air distribution configurations except unidirectional flow.

Air-Change Effectiveness

The calculation of air-change effectiveness according with ANSI/ASHRAE 129 (1997) needs calculation of the age of air. This could be obtain by two ways: from a tracer gas decay and from a tracer gas step-up. The method used in this study were the tracer gas decay.

The age of air from a tracer gas decay is calculated from eq. 2.16:

$$A_i = \frac{(t_{stop} - t_{start})C_{i,avg}}{C_{i(t_{start})}} \tag{2.16}$$

where
A_i = the age of air at location i
t_{stop} = the time of the final tracer gas measurement at location i during the tracer gas decay
t_{start} = the time when outdoor airflow is started or tracer injection is stopped at the

Air Distribution Configuration	E_z
Ceiling supply of cool air.	1.0
Ceiling supply of warm air and floor return.	1.0
Ceiling supply of warm air 15°F (8°C) or more above space temperature and ceiling return.	0.8
Ceiling supply of warm air less than 15°F (8°C) above space temperature and ceiling return provided that the 150 fpm (0.8 m/s) supply air jet reaches to within 4.5 ft (1.4 m) of floor level. ***Note:*** For lower velocity supply air, E_z = 0.8.	1.0
Floor supply of cool air and ceiling return provided that the 150 fpm (0.8 m/s) supply jet reaches 4.5 ft (1.4 m) or more above the floor. ***Note:*** Most underfloor air distribution systems comply with this proviso.	1.0
Floor supply of cool air and ceiling return, provided low-velocity displacement ventilation achieves unidirectional flow and thermal stratification.	1.2
Floor supply of warm air and floor return.	1.0
Floor supply of warm air and ceiling return.	0.7
Makeup supply drawn in on the opposite side of the room from the exhaust and/or return.	0.8
Makeup supply drawn in near to the exhaust and/or return location.	0.5

Figure 2.25: Typical values for Zone Air Distribution Effectiveness. ANSI/ASHRAE 62.1 (2007)

beginning of tracer gas decay
$C_{i,avg}$ = the time-averaged tracer gas concentration at location i between time t_{start} and t_{stop}
$C_{i(t_{start})}$ = the tracer gas concentration at location i at time t_{start}

The parameter $C_{i,avg}$ equals the arithmetic average of the measured tracer gas concentrations at location i when concentration is measured at approximately equally spaced intervals of time. When the concentration measurements are made with uneven time intervals $C_{i,avg}$ is calculated from eq. 2.17:

$$C_{i,avg} = \frac{\sum\limits_{n=first}^{n=last-1} \left[\frac{C_{i,n}+C_{i,n+1}}{2}(t_{n+1}-t_n)\right]}{t_{last}-t_{first}} \tag{2.17}$$

Its necessary to calculate de nominal time constant using eq. 2.18:

$$\tau_n = \frac{\sum\limits_m (Q_{ex,m}A_{ex,m})}{\sum\limits_m (Q_{ex,m})} \tag{2.18}$$

where
τ_n = nominal time constant
m = an identification number unique for each exhaust air-stream

$Q_{ex,m}$ = rate of airflow in exhaust air-stream m
$A_{ex,m}$ = age of air in exhaust air-stream m

The air-change effectiveness is calculated form eq. 2.19:

$$E = \frac{\tau_n}{A_{avg}} \tag{2.19}$$

where
E = air-change effectiveness
A_{avg} = arithmetic average of the ages of air measured at breathing level within the test space[1]

Method of measurement

The method of measurement of this indicators is similar to the indicated in section 2.8.3.

2.8.5 Comparison of ventilation efficiency indicators

Although there are several indicators of ventilation efficiency, the two that appear to be the most suitable for use in design and standards are ε^c and ε^a (E_z is derived from ε^a). These two indicators are applicable to all types of ventilation systems (airflow patterns) and they can be more easily measured in the field or laboratories.

For comparison of the ε^c and the ε^a, the main question is which one of these two perceived air quality indicators is more appropriate for general use, such as in design and standards, especially because these perceived air quality indicators can show conflicting values for indoor air quality in the same room.

The discrepancies in the concentration and age of air distribution patterns result in conflicting values of ε^a and ε^c with different positions of the contaminant source (see fig. 2.26).

Different values of ε^a and ε^c are due to the different "nature" of these two indicators. The air exchange efficiency is an indicator of air distribution quality because it quantifies how good the airflow pattern is. This efficiency indicator accounts for the size and intensity of the recirculation in the room by comparing the room airflow pattern with the airflow pattern of the ideal piston flow. On the other hand, the contaminant removal effectiveness is the indicator of contamination level in a room. The effectiveness indicator depends not only on the airflow pattern but also on the intensity, area, and positions of contaminant sources relatively to this airflow pattern. Consequently, for well-known positions and intensities of contaminant sources, contaminant removal effectiveness provides good indication of air quality. However, in the spaces where contaminants are unknown, the air exchange efficiency is a more useful indicator because it provides general indication

[1] The region within an occupied space between planes 75 and 1800 mm above the floor and more than 600 mm from the walls or fixed air-conditioning equipment (ANSI/ASHRAE 62.1 (2007)). This concept is similar to occupied zone defined in EN 13779 (2007) but differs in the limits impose.

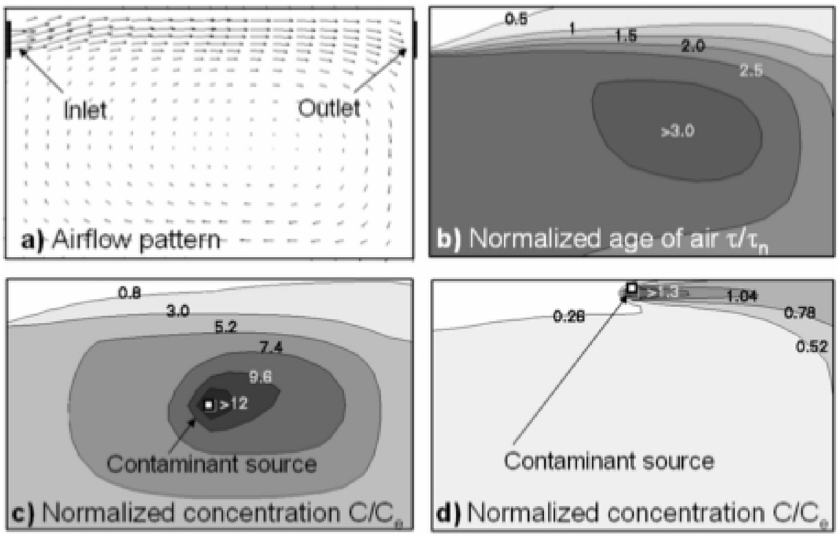

Figure 2.26: Normalized local age of air and contaminant concentration distributions for the same airflow field and different positions of the contaminant source; (a) airflow pattern, (b) $\varepsilon^a = 0.21$ for this airflow pattern, (c) $\varepsilon^a = 0.21$ and $\varepsilon^c = 0.19$ for contaminant source at room center, (d) $\varepsilon^a = 0.21$, and $\varepsilon^c = 2.20$ for contaminant source in the supply air jet. Novoselac and Srebric (2003)

of air quality independently of contaminant source positions.

In sections 2.8.2 and 2.8.3 we have seen Equations the different physical meanings of contaminant removal effectiveness and air exchange efficiency, rendering a direct comparison meaningless. The air exchange efficiency has values from 0 to 1, while the contaminant removal effectiveness takes values from 0 to infinity. Fig. 2.27) shows ventilation cases where ε^a and ε^c values approach their lower and upper limits.

Air exchange efficiency	Upper limit	$\varepsilon_a = 1$	Ideal piston flow
	Perfect mixing	$\varepsilon_a = 0.5$	Complete and instantaneous mixing
	Lower limit	$\varepsilon_a \rightarrow 0$	Bypass area and recirculation area are completely separated
Contaminant removal effectiveness	Upper limit	$\varepsilon \rightarrow \infty$	Contaminant source at the outlet Flow field does not have influence
	Perfect mixing	$\varepsilon = 1$	Complete and instantaneous mixing Position of contaminant does not have influence
	Lower limit	$\varepsilon \rightarrow 0$	Contaminant source is in the recirculation area, which is completely separated from the bypass area

Figure 2.27: Limits for Air Exchange Efficiency and Contaminant Removal Effectiveness. Novoselac and Srebric (2003)

The contaminant removal effectiveness depends not only on the position of the contaminant relative to the airflow pattern but also on the area of the source region. For example, a contaminant source might be released from a point source such as tobacco smoke or from a large area source such as pollutants from a floor finish. Furthermore, contaminant removal effectiveness depends on source properties such as contaminant density. Consequently, it is hard to expect a relationship between ε^c and ε^a (Sandberg and Sojeberg (1984); Skaret (1984)) because the number of combinations of flow patterns and the type and positions of the contaminant sources are unlimited.

2.8.6 Ventilation efficiency in Portuguese legislation

In Portugal the legislation defines ventilation efficiency as the "ratio between the flow rate of fresh air that is blown or enters a given area and flow rate of fresh air that arrives in fact to the occupied space, defined as the volume corresponding to the useful area to a useful height of 2m" (RSECE (2006)).

Although this definition, is not indicated any methodology for the measurement or calculation of the efficiency of ventilation. Because of that situation ADENE, entity that manages the application of SCE, in order to harmonize, published indications of values to adopt (ADENE (2011)). It is the understanding of the group of experts integrate the Scientific Coordination of the SCE, that should be only use the values of 60, 70, 80 and 90% for the efficiency of ventilation in accordance with the situations described below:

- 60% - where supply and extraction are both made by a ceiling, or together to ceiling and close together (situations where the supply jet hits the spot where the extraction is done) and without specific measures to reduce the risk of short-circuit between them;
- 70% - in cases which fall in an intermediate situation between those described for utilization efficiency values of 60% and 80%, for example, occurs only when short-circuit for a part of supply diffusers/grilles;
- 80% - in the spaces with good strategy of distributing the supply air, including situations where supply and extraction are made or near the ceiling, provided that is satisfied all the conditions below:
  - the risk of short-circuit is minimizing through the maximizing of the distance of supply-extraction, or strategies that optimize the route of the effective air jet blowing in the occupied zone;
  - high induction diffusers, well distributed;
  - extraction in stagnant zones of the flow field.
- 90% - situations in which the supply is made in a zone near the floor and the extraction along the ceiling without possibility of short-circuit, or other type of supply where the mixture is excellent and approaches the efficiency of systems like displacement.

In the case of airflow strategy of displacement or equivalent , the efficiency should always be considered of 100%, if the introduction of air is taken at the floor level in the occupied zone, cooler than the room temperature at low speed.

2.9 Measurement methods

2.9.1 Tracer gases techniques

The techniques known as tracer gases is a versatile tool for determination of air flow rates in ventilation systems for buildings, offices, etc.. These techniques allow determining ventilation rates and patterns of air circulation. They also permit measuring the air flow in ventilation systems where conventional methods are not practical to use and accurate.

Implementation of tracer gases techniques

The ASTM-E741 (2006) described a method know as "Standard Test Method for Determining Air Exchanges in the Single Zone by Means of a Tracer Gas Dilution". This test method uses the measurement of tracer gas dilution to determine air change within a building or other enclosure that is characterized as a single zone. The measurement of the concentration, and sometimes the volume rate of the tracer gas that is injected into the zone, allows calculation of the volume rate of outgoing air from the zone. Three techniques are used:

- concentration decay
- constant injection
- constant concentration

Each technique employs specific tracer gas injection and sampling strategies. The choose technique depends of the speed of measurement[2], time-varying air change[3], quantity to be measured, the comparative capabilities of the techniques, the complexity of zone geometry[4] and the complexity of the required equipment[5]. Table 2.4 summarizes the three techniques.

Concentration decay test method

This method is used when you want to determine the rate of air exchange in short time. Consists of to inject a small amount of tracer gas in place to study, mixing it well with the air at the site with the help of fans. Then is removed the source of emission and registered the decrease of the concentration of the tracer gas over time. An advantage of this method is that the emission rate not need be measured, but should be controlled so that the concentration is in the range of measurement equipment used. The biggest problem of this technique is the poor mixing of the tracer gas with the indoor air. This is the method used in this study.

[2] A one-time measurement of air change is most quickly acquired with the concentration decay technique and least quickly with the constant concentration technique.

[3] The constant concentration and constant injection techniques may be useful for measuring air change rates that vary with time.

[4] Whereas all the techniques require uniform tracer gas concentration, the constant concentration technique may be useful to achieve this in a zone with complex geometry.

[5] The complexity of the required equipment is lowest for the tracer gas decay technique and highest for the constant concentration technique.

Technique	Type of Air Change Measurement	Steady-State Assumption Required	Volume Control of Tracer Gas	Concentration Measurement Relative to
Concentration Decay				
Average	Rate	No	Approx. initial target	Other samples
Regression	Rate	Yes	Approx. initial target	Other samples
Constant Injection				
Average	Flow	No	Flow rate within 2%	Absolute standard
Constant Concentration				
	Flow	No	Mean Concentration within 2% of target	Absolut standard

Table 2.4: Summary of Air Change Measurement Techniques. ASTM-E741 (2006)

Constant injection test method

In this method the flow rate of gas emission is adjusted to be constant so there is no need for an initial period to mixture the tracer gas and indoor air. It is used in continuous measurements of rates of air exchange in one or more zones. It particularly useful measurements in dwellings. Another advantage is that the emission of gas in each zone of the building can be controlled separately so that the external air flow into each zone can be determined. Has the disadvantage of require the measurement of absolute emission rates and concentration of the tracer gas.

Constant concentration test method

This method is widely used in habited areas, when it is necessary to carry out measurements over long periods of time. Consist in getting the greatness that quantifying the amount of air from the values of the flow released, in every instant for the concentration remains constant in the area studied. Has the advantage that it can used in places where the air change rate varies over time, thus obtaining a register of the development rate of the same.

2.9.2 Properties of tracer gases

A gas to be classed as a good tracer should be colourless, odourless, inert and non meet present in the environment. In addition to the physical properties the tracer gases should fulfil some other requirements in regard to their practical suitability. These include health safety aspects, low environmental burden, high availability and good handling in practical use at the lowest costs possible and, not least, tracer gases should be well recordable with established measurement techniques over a wide concentration range and with high selectivity (Raatschen (1995)).

None tracer gas have all that properties, the table 2.5 shows the properties of the most common used tracer gases. The most commonly used tracer gas is Sulphur Hexafluorid (SF_6). They are also used other trace gases such as carbon dioxide (CO_2) and Nitrous

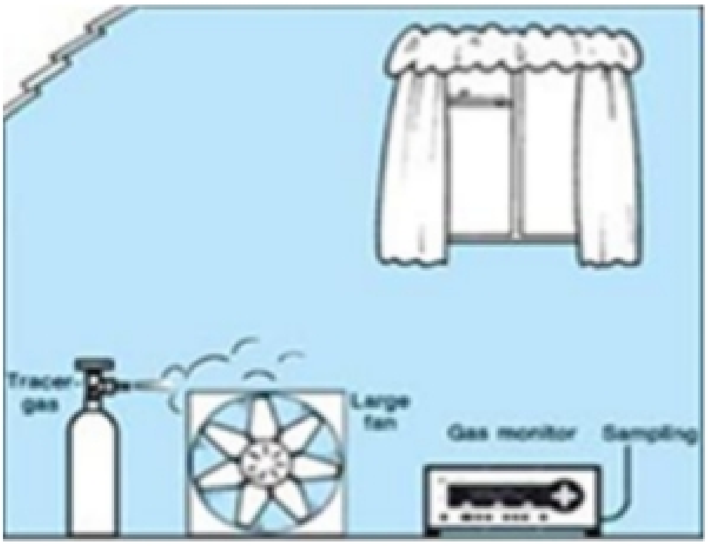

Figure 2.28: Concentration decay Test Method. Lucas (2011)

Oxide (N_2O).

In the past, a number of gaseous substances such as helium, hydrogen, oxygen, carbon monoxide, methane, acetone, and the radioactive noble gases argon-41 and krypton-85, have been studied and tested for the determination of ACR, mostly in comparison with other tracer gases (Laussmann and Helm (2011)). Reviews are given by Grimsrud et al. (1980), Shaw (1984), and Sherman (1990). Until the early 1990's, krypton-85 was still used for air change measurement Schulze and Schuschke (1990), but later skipped for safety reasons (radiation protection).

Tracer gas	Density (Air is 1,2 kg/m^3)	Toxicity threshold value	Chemical stability	Cost	Absorption of IR light
SF_6	6,3	1000 ppm	Slightly soluble in water	50 €/kg in 10 l bottle	Stronger
CO_2	1,9	5000ppm	Soluble in water		
N_2O	1,9	50 to 100 ppm		15 €/kg in 10 l bottle	

Table 2.5: Some properties of three common tracer gases. Mundt et al. (2003)

2.9.3 The use of carbon dioxide as a tracer gas

CO_2 is one of the gaseous organic compounds always detectable in the indoor air. Since humans exhale metabolic carbon dioxide in considerable quantities, its concentration can increase to several thousand ppm within a short time. CO_2 concentration is often used to assess the air quality of occupied rooms. In this context we remind Pettenkofer (1858)

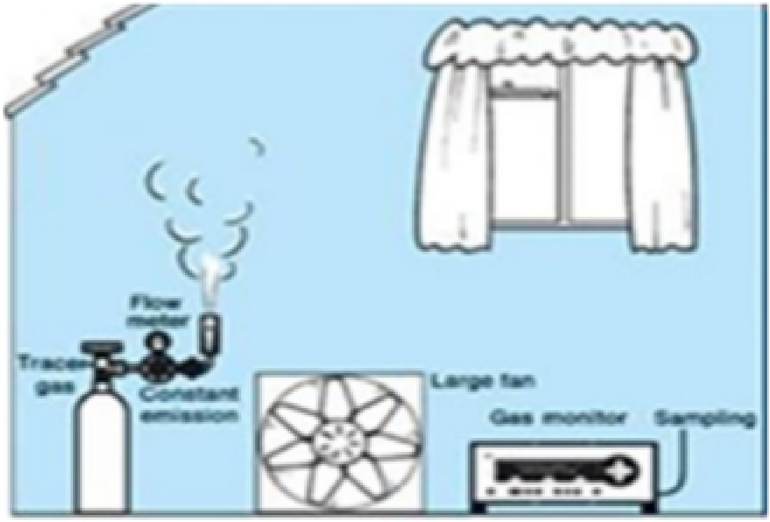

Figure 2.29: Constant injection Test Method. Lucas (2011)

reference concentration. Already in 1858 the German chemist and hygienist pointed out that a CO_2 concentration of 1000 ppm (0,1 vol %) is the upper tolerable limit in indoor environments. Nowadays CO_2 measurements are often used for the determination of the indoor ACR, because it can be easily quantified and the required devices are reasonably priced and easy to operate. Moreover, CO2 fulfils a number of the above mentioned specifications of a good tracer gas. Laussmann and Helm (2011)

2.9.4 Tracer gas analyser apparatus

The two most used technologies to measure the concentration of a tracer gas are:

- PAS - Photoacoustic Spectroscopy;
- NDIR - Non dispersive infrared absorbance.

Photoacoustic Spectroscopy

Photoacoustic spectroscopy involves irradiating intermittent light onto a sample and then detecting the periodic temperature fluctuations in the sample as pressure fluctuations. It is being applied in a diverse range of fields as Fourier transform infrared spectroscopy (FTIR) becomes more widespread.

Photoacoustic spectroscopy measurement is based on the photoacoustic effect. The photoacoustic effect was discovered by Alexander Graham Bell in 1880. This is the phenomenon whereby, when intermittent light is irradiated onto a substance, the substance emits acoustic waves of the same frequency as the light pulse frequency. It took several decades from that time until the photoacoustic effect was subsequently applied as a measurement technique. With the development of highly-sensitive microphones and other advances in electronics, research progressed into the

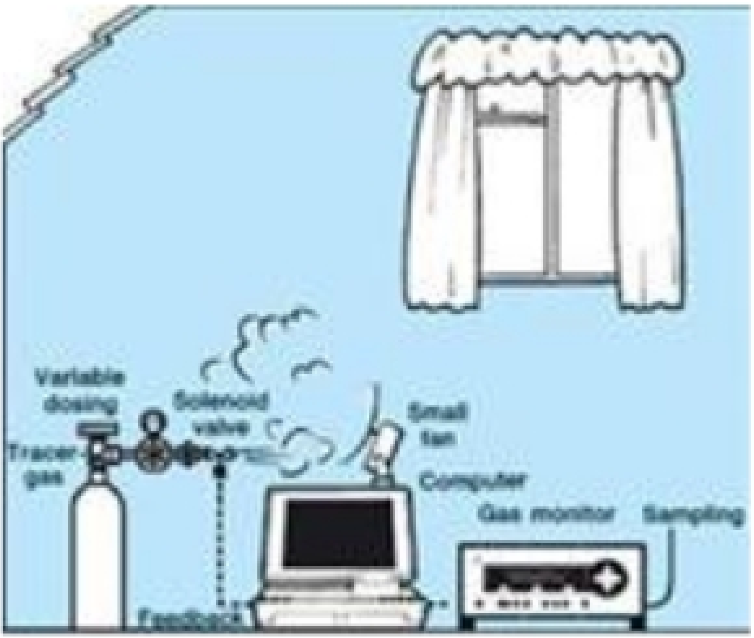

Figure 2.30: Constant concentration Test Method. Lucas (2011)

measurement of gas samples, in particular.

When a modulated infrared light beam is absorbed by the sample, heat is generated due to the incident light. This heat causes pressure changes in the surrounding gaseous layer, which can be detected by the high-sensitivity microphone. The signals from the microphone are acoustic interference waves. Applying Fourier transformation to these signals produces a spectrum similar to an absorption spectrum (see figure 2.31). The theory of photoacoustic spectroscopy differs according to the sample form and photoacoustic cell construction (Sawada (1982)).

With PAS, the absorption (proportional to the concentration) is measured directly and not relative to a background. This means that PAS is highly accurate and stable.

Non dispersive infrared absorbance

Non-dispersive Infrared sensors are simple spectroscopic devices used for gas analysis. The key components are an infrared source (lamp), a sample chamber or light tube, a wavelength filter, and an infrared detector (see figure 2.32). The gas is pumped or diffuses into the sample chamber, and gas concentration is measured electro-optically by its absorption of a specific wavelength in the infrared (IR). The IR light is directed through the sample chamber towards the detector. The detector has an optical filter in front of it that eliminates all light except the wavelength that the selected gas molecules can absorb. Other gas molecules do not absorb light at this wavelength, and do not affect the amount of light reaching the detector. The IR signal from the source is usually chopped or modulated so that thermal background signals can be offset from the desired signal. For greater optical efficiency, a reflector assembly can surround the lamp used for

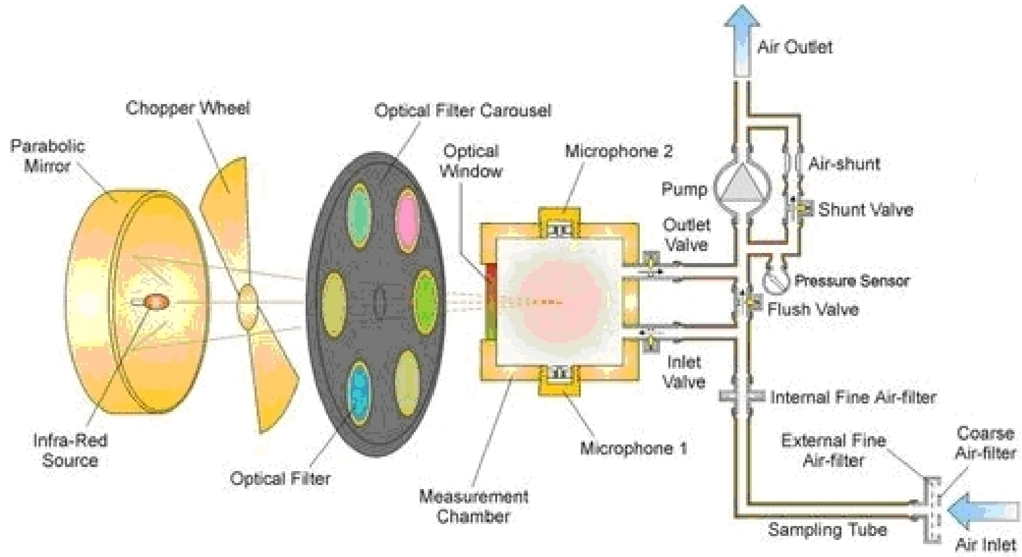

Figure 2.31: Scheme PAS sensor. LumaSense Technologies (2012)

the NDIR sensor. The reflector is usually parabolic in shape to collimate the IR light through the sample chamber towards the detector. The use of a reflector can increase available light intensity by two to five times. The reflector surface can also be gold-coated to further enhance its efficiency in the infrared. The intensity of IR light that reaches the detector is inversely related to the concentration of target gas in the sample chamber . When the concentration in the chamber is zero, the detector will receive the full light intensity. As the concentration increases, the intensity of IR light striking the detector decreases. Beer's Law describes the exact relationship between IR light intensity and gas concentration:

$$I = I_0 e^{kP} \tag{2.20}$$

I = the intensity of light striking the detector
I_0 = the measured intensity of an empty sample chamber
k = a system dependent constant
P = the concentration of the gas to be measured

NDIR sensors can be used to measure practically all inorganic and organic gases, but are most often used for measuring carbon dioxide because no other sensing method works as simply and reliably for this gas. Calibration gases of specific concentration are available for determining the system constant k for any particular sensor design (InternationalLight Technologies (2013)).

2.10 Maintenance and design for maintenance

Evidence suggests that the maintenance of ventilation systems is often inadequate and that the need for maintenance may even be ignored in the course of building design. Inaccessibility of system components, poor durability and a lack of awareness of servicing

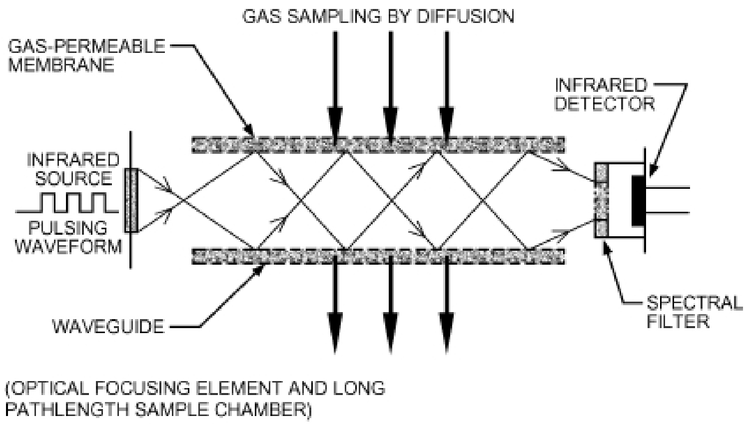

Figure 2.32: Scheme NDIR sensor. ASHRAE (2009)

needs have all contributed to reduced ventilation performance (Liddament (1996)).

In various study about the long term performance of mechanical ventilation, servicing was found to be inadequate both in dwellings and in large buildings (Rask (1989); Pallari and Luoma (1993); Werner et al. (1994); Granqvist and Kronvall (1994)). Typical problems included:

- lack of maintenance planning
- worn gaskets
- fans and grilles dirty
- missing filters
- clogged filters
- inaccessible components
- components representing a hazard to safety locations
- inaccessible control systems
- lack of operating and maintenance instructions

2.10.1 Design for easy maintenance

Maintenance is needed to ensure the reliability of the ventilation system and to secure the economic operation of the ventilation plant. There are systems that been installed without a clear idea of how maintenance is to be accomplished. The ease with which a system can be maintained is strongly influenced by the degree to which the issue has been considered at the design stage (Liddament (1996)).

In a ventilation system design, the main precautions to take to be easy maintenance are:

- **Controls:** shall be easy to reach, understand and operate.
- **Location of components:** components which require attendance shall be sited so that they are readily accessible and replaceable, and are mounted so that work can be carried out easily and safely.
- **Cleanability:** it shall be possible for both supply and extract ventilation air installations to be cleaned in their entirety. Installations shall be cleaned sufficiently frequently to ensure that neither the magnitude of air flows nor the quality is adversely affected by deposited dirt.
- **Components and materials:** components shall be made of materials which stand up to the intended use and maintenance and do not emit pollutants such as particles or gases which may adversely affect the quality of the supply air. The choice of materials and construction shall be such that the growth of micro-organisms is prevented.
- **Air tightness and pressure conditions:** is recommended that installations shall have the required air tightness. Pressure conditions between supply and extract air installations shall be adjusted so that there is no unintentional flow from the extract air to the supply air.
- **Commissioning:** ventilation systems shall be balanced so that the intended flow rates and tolerances are obtained. When an installation is handed over, it shall be demonstrated that it has been constructed and functions in the way intended. The installation shall be handed over in a clean state ready for operation.
- **Documentation:** the necessary drawings and specifications shall be produced for a building and the ventilation installation. The materials used including make and type designation shall be documented. Air flow rates through individual rooms shall be specified. Instructions for the operation and maintenance of the ventilation installation shall be prepared and shall be available when the building is put into service. User instructions in easily understandable language, which provide information on attendance, cleaning and maintenance, shall be affixed within easy reach of each terminal or appliance which is capable of being controlled by the user.
- **Inspection:** at all stages of the design, construction and operation of a building, checks shall be made to ensure that the intended quality is secured. Buildings shall be regularly inspected to ensure the correct functioning of the ventilation system and of other factors which influence good indoor climate.

2.10.2 Maintenance of specific components

Reliable ventilation system performance depends on maintenance of the component parts. Major items include:

- Fans
- Air filters
- Ductwork and air distribution systems

- Air treatment plant
- Terminal units

Of these components, the one that presents major challenges is the ductwork and air distribution systems, especially in large buildings, because they present particular problems since the air system is necessarily large and the distribution system complex.

Maintenance of ductwork and air distribution systems

The cleaning of ductwork systems is a subject which has gained increased attention over recent years. Accumulation of dust within ductwork systems can provide a site for the development of microbiological growth. This can result in bacteria or fungal spores being released into the occupied space with potential impacts on occupant comfort and health.

The control of dust build up should begin in the installation and should remain during the operation. This control must be enhanced by good filtration regimes, but occasional duct cleaning may also be required. To facilitate cleaning, access doors must be provided. EN 12097 (2006) presents the requirements for ductwork components to facilitate maintenance of ductwork systems.

Four steps are essential to carry out an air duct cleaning operation. These different phases are: diagnosis, cleaning, disinfection process and monitoring (Barbat and Feldman (2010)).

HVAC system inspection. The diagnosis is carried out to assess the general state of the duct system and to look for local and deposits of dust. Depending on the results, the diagnosis can justify or not a complete and deep air duct cleaning. Ductwork diagnosis requires three main phases: visual inspection, dust accumulation measurements and microbiological analysis.

Visual inspection allows evaluating dust accumulation along the ductwork. In addition, this diagnosis can highlight defaults of the installation: bad connection between ducts elements, corrosion and internal moisture. Depending on the network shape different tools may be used. A simple camera is used for vertical ducts. A four wheel mobile camera is particularly suitable for horizontal ducts (see fig. 2.33). But this type of diagnosis is quite limited because when the deposit duct looks like a thin layer it is very difficult to judge on its importance and therefore to assess the effect on comfort and health in the indoor spaces.

Other methods may be used to assess objectively the dust accumulation level. The method mostly used to make dust accumulation measurement is the weighing method, known as the Vacuum Test (Holopainen et al. (2003)). This method consists in vacuuming dust on the inner surface of an air duct through a template delimiting the sampling are. Dust is collected on a filter which will be weighed by an accurate laboratory weighting machine.

Figure 2.33: Four wheel mobile camera unit. Barbat and Feldman (2010)

Microbiological analysis aims to characterise the degree of contamination of the system. Three different sampling points are required. A reference sampling point must be located near the outdoor air intake. A second sampling has to be carried out in the supply air part of the ductwork right after the Air Handle Unit. The third sampling point must be located n the extract air duct. In order to obtain reliable results air sampling must be made with an iso-kinetic probe in which air velocity is equal to air flow velocity in the duct (Barbat and Feldman (2010)).

Mechanical cleaning methodology. Cleaning procedure consists of two complementary tasks: unstick dust particles from the duct surface and collect dust in suspension in the air flow. To achieve a good cleaning of the ductwork air velocity must be much higher than the velocity design values.

The two most popular duct cleaning techniques are mechanical brushing and compressed air blowing (Barbat and Feldman (2010)).

Mechanical brushing is based on the rotary motion of a brush. Rotation speed and direction of the brush may generally be controlled. Depending on the type of deposit brush material may be more or less stiff (see fig. 2.34).

Compressed air blowing is mainly used for fragile duct wall materials (i.e glass wool). It can be also used for the total completion of the cleaning process. Glass wool wall ducts require special care when cleaning.

The cleaning process used depends on internal protection of the insulation material of the duct and the type of material of the duct. The table 2.6 gives the recommended means of cleaning in regard to protection of the insulation material.

The mechanical brushing technique is the most efficient method especially in metal ducts. The compressed air technique is slower and noisier compared with the brushing

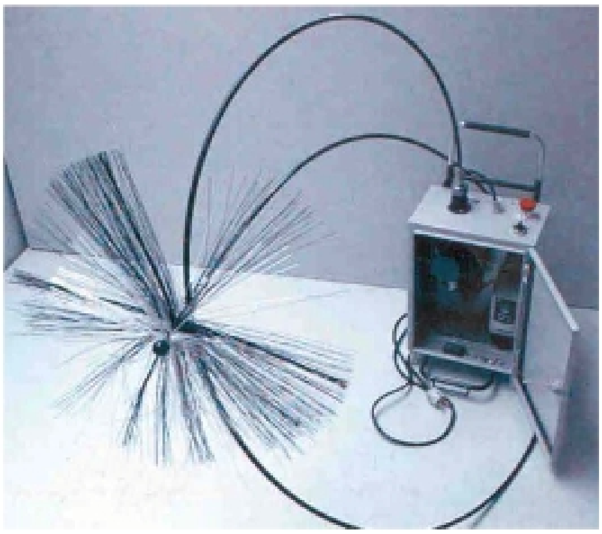

Figure 2.34: Mechanical brushing. Barbat and Feldman (2010)

Year	Insulating	Mechanical cleaning methodology
Before 1980	Insulation material without protection	No techniques
Since 1986	Insulation material with a light protection	Compressed air
Since 1993	Insulation material with an aluminium protection	Mechanical brushing
Since 1996	Insulation material with a thick protection	Mechanical brushing and compressed air

Table 2.6: Cleaning methodology vs insulation of mother the duct. Barbat and Feldman (2010)

methods, but presents good results especially in plastic ducts (Holopainen et al. (2003)).

Disinfection methods. The disinfection process is carried out after the cleaning of the air duct network. The aim of the disinfection process is to remove or kill micro-organisms. Micro-diffusion is the most common way of doing this. The method consist of producing a non-wetting fog (particles below 1 mm) having a wide anti-bacteria activity spectrum into the ductwork. Another method is based on the combustion of active substances. Fumes emitted by combustion provide an anti-bacteria effect. Due to deposit of small particles on the duct surface a re-cleaning of ducts is needed when using this process (Barbat and Feldman (2010)).

Cleanliness checking. Visual inspection, microbiological sampling and analysis and dust accumulation measurement must be carried out before and after the cleaning operation.

2.10.3 Implementing maintenance regulations and standards

Various Standards and Regulations are being introduced to ensure the quality and reliability of ventilation systems.

In the United States, proposed guidelines for ANSI/ASHRAE 62.1 (2007) cover the maintenance of HVAC systems to maintain good indoor air quality. The fig. 2.35 shows the standard recommendations.

Item	Activity Code	Minimum Frequency*
Filters and air-cleaning devices	A	According to O&M Manual
Outdoor air dampers and actuators	B	Every three months or in accordance with O&M Manual
Humidifiers	C	Every three months of use or in accordance with O&M Manual
Dehumidification coils	D	Regularly when it is likely that dehumidification occurs but no less than once per year or as specified in the O&M Manual
Drain pans and other adjacent surfaces subject to wetting	D	Once per year during cooling season or as specified in the O&M Manual
Outdoor air intake louvers, bird screens, mist eliminators, and adjacent areas	E	Every six months or as specified in the O&M Manual
Sensors used for dynamic minimum outdoor air control	F	Every six months or periodically in accordance with O&M Manual
Air-handling systems except for units under 2000 cfm (1000 L/s)	G	Once every five years
Cooling towers	H	In accordance with O&M Manual or treatment system provider
Floor drains located in plenums or rooms that serve as air plenums	I	Periodically according to O&M Manual
Equipment/component accessibility	J	
Visible microbial contamination	K	
Water intrusion or accumulation	K	

ACTIVITY CODE:
A Maintain according to O&M Manual.
B Visually inspect or remotely monitor for proper function.
C Clean and maintain to limit fouling and microbial growth.
D Visually inspect for cleanliness and microbial growth and clean when fouling is observed.
E Visually inspect for cleanliness and integrity and clean when necessary.
F Verify accuracy and recalibrate or replace as necessary.
G Measure minimum quantity of outdoor air. If measured minimum airflow rates are less than 90% of the minimum outdoor air rate in the O&M Manual, they shall be adjusted or modified to bring them above 90% or shall be evaluated to determine if the measured rates are in conformance with this standard.
H Treat to limit the growth of microbiological contaminants.
I Maintain to prevent transport of contaminants from the floor drain to the plenum.
J Keep clear the space provided for routine maintenance and inspection around ventilation equipment.
K Investigate and rectify.

* Minimum frequencies may be increased or decreased if indicated in the O&M Manual.

Figure 2.35: Minimum Maintenance Activity and Frequency. ANSI/ASHRAE 62.1 (2007)

Nowadays in Europe, all countries have developed recommendations for maintenance. In Portugal with the approval of RSECE (2006), began obligation to conduct maintenance tasks in buildings with HVAC power exceeding 25kW. Under the same legislation, were created the minimum qualifications required of technicians to perform installation tasks and maintenance of HVAC systems. It has also created the mandatory inspections on HVAC equipment, particularly boilers and air conditioning equipment.

2.11 Ventilation and acoustics

2.11.1 Acoustic design in buildings

There is a close relationship between the three factors, ventilation (in general, the heating, ventilation and air conditioning (HVAC) concept/system), indoor air quality (IAQ) and acoustics, and the design of each influences the performance of the others significantly (Khaleghi et al. (2008)).

The implications of noise can be different for natural and mechanical ventilation. The noise from the energy systems of the building may disturb the occupants and prevent the intended use of the space or building (Olesen (2007)). Natural ventilation, although accepted as a sustainable design strategy, may conflict with the control of ingress of external noise through ventilation openings (Field (2008)).Ventilation should not rely on opening of windows in the areas with high outdoor noise where it is not possible to reach the target level when airing or if the building is located in an area with a high outdoor noise level compared to the level the designer wishes to achieve in the indoor zone (EN 15251 (2007)).

Recent studies shows that mechanically ventilated schools did not meet the British requirements of internal ambient noise levels. This indicates that mechanical ventilation does not automatically provide a complete solution, and that good acoustic attenuation of the ventilation system is required (Mumovic et al. (2009))

2.11.2 Acoustic comfort requirements in European standards and national regulations

Noise requirements in European countries vary in a wide range. Limit noise levels as defined in national regulations and standards are very inconsistent, because they are given in three different units, which cannot be compared. Inconsistency in the use of units since countries use maximum level (L_{AFmax}), equivalent level (L_{eq}), and noise rating curves (NR). Due to the different definitions used in the definitions of the three units, they are not directly comparable (Brelih (2013)).

EN 15251 (2007) defines that or design of ventilation the required sound levels shall be specified in the design documents based on national requirements. If this is not the case this standard recommended values that may be applied if appropriate. Often national requirements exist for noise from service equipment inside or outside assuming windows are closed. These criteria apply to the sources from the building as well as the noise level from outdoor service equipment. The criteria should be used to limit the sound pressure level from the mechanical equipment and to set sound insulation requirements for the noise from outdoors and adjacent rooms.

According with EN 15251 (2007) noise is evaluated with a representative sample from different air handling systems, zones, windows, and orientation. Normally the criteria for noise do not influence the energy performance of buildings. It could, however, occur in naturally ventilated buildings, that the required amount of outside air cannot be obtained by opening of windows because noise from outside would violate the criteria. Also in the case of mechanical ventilation and cooling, providing the required amount of air would result in unacceptable noise levels from fans.

2.12 Ventilation performance prediction for buildings

Designing ventilation systems for buildings requires a suitable tool to assess the system performance. This assess is made with seven types of models:

- Analytical
- Empirical
- Small-scale experimental
- Full-scale experimental
- Multizone network
- Zonal
- CFD

2.12.1 The analytical model

Analytical models are derived from fundamental equations of fluid dynamics and heat transfer, such as mass, momentum, energy, and chemical-species conservation equations. The analytical models use simplifications in both geometry and thermo-fluid boundary conditions in order to obtain a solution. As a result, the final equations obtained for one case may not be used for another without modifications. However, the methodology and approximations could be similar for difference cases.

The analytical models are probably the oldest method for predicting ventilation performance. This method is still widely used today due to its simplicity, rich in physical meaning, and little requirement in computing resources, although it may not be accurate for complicated ventilation cases and the results may not be informative.

2.12.2 The empirical model

Similar to the analytical models, the empirical models are developed from the conservation equations of mass, energy, and chemical species. In many cases, the data of experimental measurements or advanced computer simulations are also used in the development of the empirical models to obtain some coefficients that make empirical models work in a certain scope. In theory, the analytical and empirical models do not differ very much. The perception is that the empirical models may use more approximations than the analytical models.

There are thousands of empirical models for different ventilation performance assessments. These applications of the empirical models demonstrate that the models are effective, cost-cutting tools for ventilation engineers and designers to predict ventilation performance in buildings. The performance of the empirical models is similar to that of the analytical models.

2.12.3 The small-scale model

The small-scale experimental models use measuring techniques to predict or evaluate ventilation performance with a reduced scale of the buildings or rooms. It is much more economical to use a small-scale experimental model than a full-scale building or room. One can get realistic ventilation performance by directly measuring thermo-fluid conditions in a small-scale model if the flow in the model is similar to these in reality. In order to achieve flow similarity between a small-scale experimental model and a real building or room, important dimensionless flow parameters in a small-scale experimental model such as Reynolds number, Grashof number, Prandtl number, etc. must remain the same as those in the actual building or room. When heat transfer is involved in a room with ventilation, it is difficult to obtain the same Reynolds and Grashof numbers. One possibility is to use liquid with different density, such as water or Freon to simulate thermal buoyancy. Otherwise, the small-scale model may not simulate the actual flow in buildings or rooms. Even though, the flow in the small-scale may not be the same as that in the actual room.

Figure 2.36: Photograph of a small-scale model filled with water for flow visualization of a two-storey building with a common atrium. Livermore and Woods (2007)

The small-scale experimental models are very effective and economical to study ventilation performance in buildings. However, in addition to scaling issues associated with thermo-fluid dimensionless parameters, it can be rather challenging to scale down complex flow geometry, for example a complex diffuser. Therefore the small-scale experimental models were mainly used to validate analytical, empirical, or numerical models. The validated analytical, empirical, or numerical models were then scaled up for studying the ventilation performance in real buildings.

2.12.4 The full-scale model

The full-scale experimental models have been widely used for predicting ventilation performance in buildings. The full-scale models were mainly used to generate data to validate numerical models, especially CFD models. The full-scale experimental models can be further classified into two categories: laboratory experiment and in-situ measurements.

Laboratory experiment often uses an environmental chamber to mimic a room or a single storey building with several small rooms. If outdoor wind conditions have to be considered, the chamber should be placed in a wind tunnel, which would make the facility very expensive.

It may not be realistic to construct a full-scale model, if one wants to predict ventilation performance in a theater or in an entire multi-storey building. One solution is to use an existing building of similar kind to predict the ventilation performance. Such in-situ measurements can be difficult because the thermo-fluid boundary conditions are not controllable in most cases. There may be unexpected disturbances during an experimental measurement. The resolution of data is often very low because it may not be practical to measure ventilation parameters in many locations in a large building. In addition, the data obtained from one building may not be applicable to a similar building nearby.

The recent applications indicate that the full-scale models by laboratory experiment or in-situ measurements gave the most realistic prediction of ventilation performance for buildings. However, they were generally very expensive and time consuming. In addition, these experimental measurements were not free from errors. Current trend seems to use full-scale experimental models of laboratory experiment and in-situ measurements to obtain data for validating computer models, such as CFD models, and then use the validated computer models to conduct the predictions of ventilation performance or design ventilation systems. The in-situ measurements were more frequently used to evaluate the performance of existing buildings.

2.12.5 The multizone model

The multizone network models are mainly used to predict air exchange rates and airflow distributions in buildings with or without mechanical ventilation systems. They can also be used to calculate ventilation efficiency, energy demand, pollutant transport, and smoke control (Axley (2007)).

The multizone models solve mass, energy and chemical-species conservation equations. However, the models assume quiescent or still air in a zone so that the momentum effect can be neglected. The models further assume uniform air temperature, and chemical-species concentration in a zone. That assumptions could cause significant errors in some cases (Wang and Chen (2008)).

The two more popular non comercial multizone programs are COMIS and CONTAM developed by two national laboratories in United States (Chen (2009)). It is applied to calculate the effect of the wind speed velocity on the stack pressure in a building, to predict airflow, pressure, and contaminant distribution in a building and to determine airflows between zones due to temperature differences (Khoukhi et al. (2007); Maatouk (2007); Sohn et al. (2007))

Although multizone models were not very accurate in each zone due to the assumptions used, they were very powerful design tools, especially for calculating airflow in a large building. The accuracy in each zone could be remedied with more detailed airflow program, such as a zonal model or a CFD model.

2.12.6 The zonal model

The well-mixing assumption used in the multizone models is not valid for large indoor spaces or a room with stratified ventilation system, such as displacement ventilation. Therefore, zonal models have been used to remedy the problem in predicting the distributions of air temperature. Zonal models divide a room into a limited number of cells, typically less than 1000 for a threedimensional space. Air temperature is calculated in each cell to determine its non-uniform distribution in the space (Chen et al. (2010)).

The zonal models had been developed based on measured airflow patterns or mass and energy balance equations. Those based on measured airflow patterns relied on the patterns to calculate air temperature distributions. Their applications were limited by the availability of airflow patterns (Chen (2009)).

If the flow momentum were strong, the accuracy of the zonal model simulations would suffer considerably. This is because the zonal models based on mass and energy balance equations do not solve momentum equation in order to reduce computing costs. In the jet region or thermal plume region where the momentum is strong, special treatments are needed that would increase significantly the complicities of the zonal models. The complicities would also increase the computing costs and make the equation system in the zonal models less stable. In this case, a modified zonal model may require similar computing effort as CFD with the same number of cells. In the future, the zonal models could be replaced by the CFD models as computers become even faster and CFD interface is more user-friendly.

2.12.7 The CFD model

The Computational Fluid Dynamics (CFD) numerically solves a set of partial differential equations for the conservation of mass, momentum (Navier–Stokes equations), energy, chemical-species concentrations, and turbulence quantities. The solution provides the field distributions of air pressure, air velocity, air temperature, the concentrations of water vapor (relative humidity) and contaminants, and turbulence parameters for both indoor and outdoor spaces. Despite having some uncertainties in the models, requiring sufficient knowledge on fluid mechanics from a user and demanding a high capacity computer, the CFD models have become more and more popular in predicting ventilation performance due to the rapid increase in computer capacity and the development of user-friendly CFD program interfaces.

The CFD models have been widely used to study indoor air quality, thermal comfort, fire safety, HVAC system performance, etc. in various buildings (commercial buildings, residential buildings, schools, health care facilities, institutional buildings, and industrial buildings), underground facilities, public transportation vehicles, greenhouses, animal facilities, etc.

The CFD models generally include Reynolds Averaged Navier–Stokes equation (RANS) modeling and Large Eddy Simulation (LES).

Applications of CFD models for predicting ventilation performance in buildings at present can be generally divided into three categories: for indoor air quality studies in spaces with nonuniform distributions of contaminant concentrations, for natural ventilation designs, and for investigations on stratified indoor environments.

2.12.8 Conclusions

The analytical models can only be applied to very simple problems. They are good tools in understanding ventilation mechanism.

The empirical models are probably the bread and butter tools for ventilation design. Most of design handbooks, design guidelines, and product catalogs for designing ventilation in buildings use empirical models.

The small-scale experimental models are less expensive than the full-scale experimental models. For this reason, the small-scale models still have their ground, despite of difficulties in modeling simultaneously inertial and buoyancy forces. The small-scale models often provide excellent flow visualization.

The full-scale models in laboratory and in-situ measurements are very popular in ventilation performance studies. Most of the experimental measurements are again for generating data for CFD model validation, as they are very time-consuming and expensive. High quality experimental data are still difficult to obtain due to the complexity of airflow in building ventilation. Many of the measurements do not provide detailed information, which makes it difficult for other researchers to use the data. This is especially true for in-situ measurements where experimentists may not even know the boundary conditions in their tests.

The multizone network models are widely used in predicting ventilation performance for entire buildings. The models, after more than two-decade development, are reasonably solid. However, because of the heavy use of assumptions, serious efforts have been made to improve the models. The models are also improved with added capacities or by coupling them with other computer models. The multizone models will continue to be an important tool for designers.

The zonal models are reduced order models that are intended for not-so-well-mixed air in buildings. Since the models do not solve the conservation equation for momentum, the accuracy suffers for the flow regions with strong momentum. Many remedy methods have been imposed, but they slow down the computation and increase the difficulty level in using the models. The zonal models have still a long way to go to be a reliable and user-friendly tool for ventilation designers.

The CFD models accounts for 70% of the ventilation performance studies published in the last years (Chen (2009)). It is interesting to note that about 2/3 of the studies used some kinds of experimental data to validate the CFD results, which clearly indicates that the community has yet trusted the CFD modeling without validation. That is why the researchers continue to seek for more reliable, more accurate, and faster CFD models. However, the effort has yet to produce fruitful results. CFD will continue to be a research tool for predicting ventilation performance in buildings. The CFD models are becoming more and more popular in design practice. It is essential that the CFD models should be more reliable and faster. Reliability is a major issue at present since the validation by experimental data requires a lot of efforts regardless the data are obtained by the CFD modelers or from the literature. The CFD speed will continue to improve as computers are becoming faster. However, the demand on computing time will continue to increase as engineers would like to use more sophisticated CFD models to handle more complicated ventilation problems with even more grid cells.

Chapter 3

Experimental measurements

3.1 SATEC laboratorial instalation

3.1.1 Room geometry

The experimental measurements were performed on SATEC located in the department of Mechanical Engineering (DEM). This is an environmental chamber with a length of 4900 mm, a width of 4020 mm and a heigth of 2950 mm. The chamber is made from aluminium sandwich plates with polyurethane insulation with a thickness of 60 mm. The ceiling is made of aluminium radiant panels. The chamber have no windows and the access is made by a door as we can see on figure 3.1. The fig 3.2 shows the interior geometry of the SATEC.

Figure 3.1: SATEC exterior

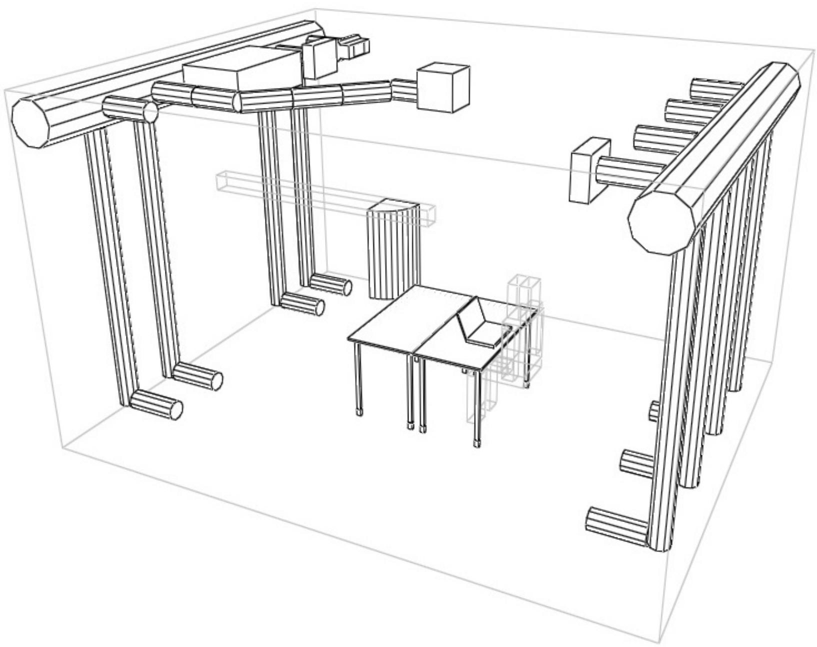

Figure 3.2: SATEC interior geometry

3.1.2 SATEC thermal systems

The thermal plant of SATEC is located above the chamber and the installation is a typical 4 pipe system. The hot water is produced by an electrical boiler with 3 levels of 5-10-15 kW. The cold water is produced by an air-to-water chiller with 6,43 kW. We can see the plant in figure 3.3.

We have several possibilities to perform the air conditioning of SATEC. In this study we used a constant volume (CV) ventilation schemes, with different types of grille/diffuser and variation of flow rate. The SATEC have a system of dampers that allows the choice of grille/diffuser tested according to the scheme.

3.1.3 Air supply and air extract system

The SATEC enables that we can supply air from 8 different locations as well we can extract air for 8 locations. The locations used in this study are represented in figure 3.4 and described in table 3.1.

The supply air is treated by an Air Handling Unit (AHU). In this type of units is possible to variate the quantity of outdoor air supplied to the room, but in this study it was always used 100% outdoor air. This unit is composed by the following modules:

- Mixing chamber
- Pre-filter module

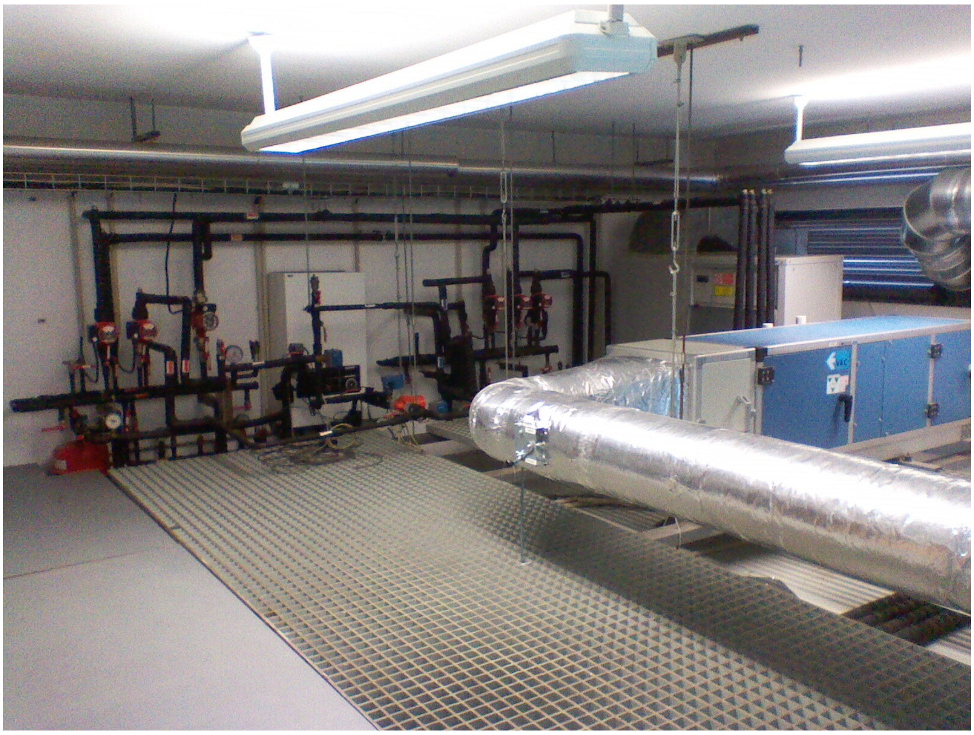

Figure 3.3: SATEC thermal plant

- Heat water coil module
- Cold water coil module
- Humidification module
- Supply ventilator module
- Filter module
- Extract ventilator module

The grilles and diffusers used in this are from TROX company. The table 3.2 shows the models used in the experimental measurements.

3.2 Measurement systems

3.2.1 Air flow

The air flow in the grilles/diffusers was measured with a air flow cone. This is a device widely used by HVAC technicians. The direction and the homogeneity of the incoming and outcoming air flow are often disrupted by the geometry of the HVAC grills. Therefore, it is necessary to canalize the flow to the sensing element of the probe. The probe and its sensing element are located in a well known section of the cone which guarantees a good measurement (see fig. 3.5).

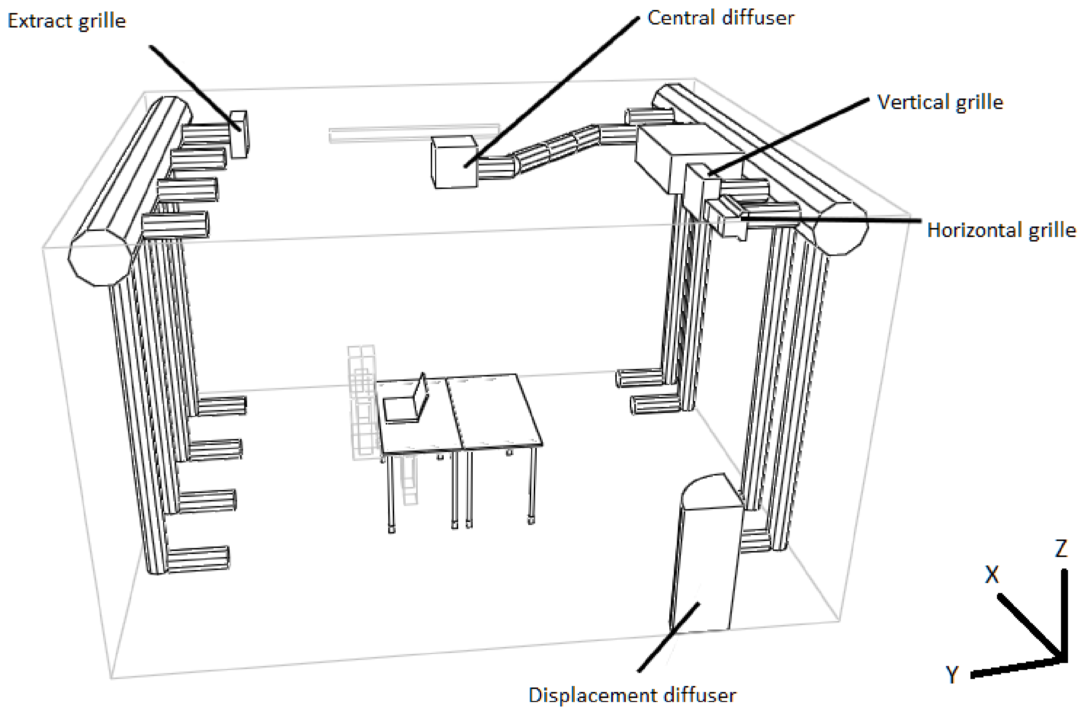

Figure 3.4: Model geometry

The sensing element used in the air flow cone is the hotwire thermo-anemometer. In fig. 3.6 we can see the measured principle. A wire is continuously heated at a superior temperature than ambient and continuously cooled by airflow. Constant temperature is maintained by a regulation circuit. The heating current is proportional to the airflow velocity. The temperature is measured with a PT100 probe. PT100 is a resistance with a positive temperature coefficient which varies according to the temperature. In this sensor the temperature increase involves increasing the resistance.

The use of this probe is very useful because we can measure the air flow in a grille/diffuser and at the same time we can measure de temperature of the supply air.

3.2.2 Temperature

The temperature was measured was in the center of the chamber during the tests. The equipment used was the KIMO datalogger (see fig. 3.7). This equipment have an NTC temperature probe. In this type of sensors the higher the temperature is, the value of the resistance decreases.

3.2.3 CO_2

The CO_2 sensors use NDIR technology (see section 2.9.4). This technology is based on the absorption of light in a gold-plated reflective light pipe or waveguide diffusion gas

Scheme	Type of Ventilation	Supply	Extract
1	mixture	horizontal grille	extract grille
2	mixture	horizontal grille	extract grille
3	mixture	vertical grille	extract grille
4	mixture	central diffuser	extract grille
5	displacement	displacement diffuser	extract grille
6	mixture	horizontal grille	extract grille
7	mixture	horizontal grille	extract grille
8	mixture	vertical grille	extract grille
9	mixture	central diffuser	extract grille
10	mixture	vertical grille	extract grille
11	mixture	central diffuser	extract grille
12	displacement	displacement diffuser	extract grille
13	displacement	displacement diffuser	extract grille

Table 3.1: Experimental schemes

	Model	Dimension mm
Horizontal grille	AT-AG/S1	325x125
Central diffuser	ADLQ-A/AK008ZM	250x250
Vertical grille	AT-AG/S1	325x125
Displacement diffuser	QL-WV-RO-1x160	1000x340
Extract grille	AT/S1	325x125

Table 3.2: Grille/diffusers models

chamber. A gas permeable PTFE filter prevents particulate and water contamination of the sensor. Light is absorbed in proportion to the CO2 concentration and the remaining light is measured and converted into an analogue signal.

The sensors used are designed to be installed in HVAC return air ducts. This product offers simplicity in design and installation. This transmitters were selected because they have de sensor element more exposed and the different air velocities cause minor disturbs. The measurement range chosen were 0 to 2000 ppm. This values are within the normal range of the occupied spaces. The values are registered on the KIMO datalogger. The data can be displayed and download to a PC

In order to know the concentration of carbon dioxide in the supply air, it was used a portable KIMO AQ200 probe. This probe use the NDIR technology as described previously. At the same time this equipment measures the temperature of the supply air. This equipment have the datalogger function, allowing the registration of the measurements for future processing.

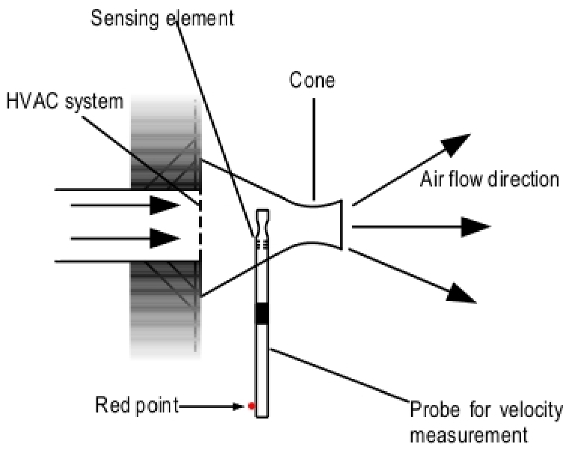

Figure 3.5: Air Flow cones measurement principle. KIMO Instruments (2010)

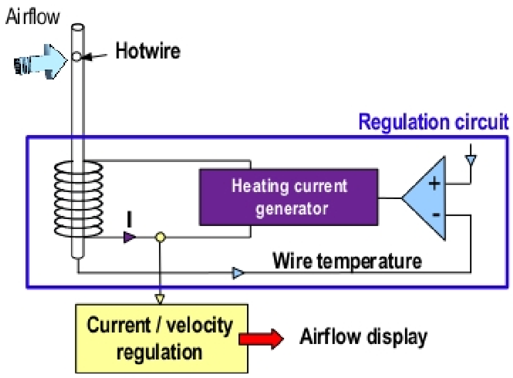

Figure 3.6: Hotwire measurement principle. KIMO Instruments (2008)

3.3 Tests

3.3.1 Air-Change Efficiency

The tests conducted to determine the ACE value are made with recommendations of the Nordtest standards (NTvvs 019 (1988); NTvvs 047 (1985)).

The tests were made with 4 CO_2 sensors. Three sensors were placed at a height of 0,83 m, 1,25 m and 1,68 m from the chamber floor and the other close to de extract grille as we can see in fig. 3.9. Table 3.3 shows the position coordinates of the locations of probes. The position of the sensors follows the recommendations of NTvvs 114 (1997) for the measurement of indoor CO_2 (see fig. 3.10).

The test used were the concentration decay method, because the reasons mentioned in section 2.9.1. The objective of the test is to determine the room-average age of air and the local mean age air. This test is accomplished by liberating a small volume of tracer gas uniformly into the chamber, with the ventilation system turned off. Subsequently the mixing of the tracer gas was promoted by mechanical means (by a fan-coil) to its

Figure 3.7: KIMO datalogger. KIMO Instruments (2012)

Figure 3.8: Carbon dioxide transmitter. Dico Filtro (2013)

dispersion by air. When this condition is guarantee, the ventilation systems is turn on and then tracer gas concentration is measure at known times. The table 3.4 recommends the minimum duration test to ensure that the decay concentration method is well applied. This test was repeated for the 13 ventilation schemes.

In this test, incandescent lamps were used as a heat source to simulate a person (see fig 3.9). It was necessary to resort to this solution because this test does not allow any additional emission of CO_2. The interior conditions of the chamber was maintain according with the values of the table 3.5.

As we have seen earlier the Zone Air Distribution Effectiveness can be obtained by the same test, by just using the ANSI/ASHRAE 129 (1997) equations.

Probe	x (m)	y (m)	z (m)
1	2	3,63	0,83
2	2	3,63	1,25
3	2	3,63	1,68
4	3,30	4,08	2,52

Table 3.3: Locations of CO_2 probes

Figure 3.9: SATEC interior

Air Change Rate (1/h)	Minimum Duration of Test (h)
0,25	4
0,5	2
1	1
2	0,5
4	0,25

Table 3.4: Examples of Minimum Durations Between the Initial and Final Samples. ASTM-E741 (2006)

3.3.2 Contaminant Removal Effectiveness

In this test the objective was to measured the steady-state concentrations of the metabolic CO_2 in the same locations that the ACE test.

In this test the ventilation system was turn on, with one person in the chamber emitting CO_2. The level of activity of the person was considered office work. The concentration of the metabolic CO_2 was measured at known times until the steady-state was reached. The interior conditions of the chamber was maintain according with the values of the table **??**. This test were repeated for the 13 ventilation schemes.

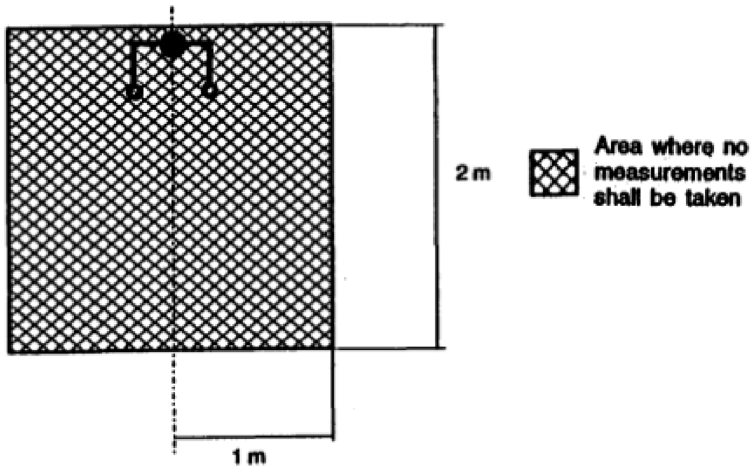

Figure 3.10: Room air Concentration sampling position around a person. NTvvs 114 (1997)

Scheme	Supply Ventilation rate [l/s]	Effective velocity grille diffuser [m/s]	Extract Ventilation rate [l/s]	θ_a [°C]	θ_{SUP} [°C]	c_{SUP} [ppm]
1	35	1,6	35	25	18	410
2	10	0,45	10	25	18	410
3	20	0,91	20	25	18	410
4	20	0,76	20	25	18	410
5	20	0,04	20	25	20	410
6	20	0,91	20	22	25	410
7	20	0,91	20	22	40	410
8	20	0,91	20	22	25	410
9	20	0,72	20	22	25	410
10	20	0,91	20	22	40	410
11	20	0,72	20	22	40	410
12	20	0,04	20	22	25	410
13	20	0.04	20	22	40	410

Table 3.5: SATEC interior conditions

Chapter 4

Numerical simulations

4.1 Software

The software used in this study was *IES-VE* with the module *MicroFlo*. This program was developed by IES - Integrated Environmental Solutions, and is oriented to energy studies in buildings.

Computational Fluid Dynamics (CFD) is concerned with the numerical simulation of fluid flow and heat transfer processes. The objective of CFD applied to buildings is to provide the designer with a tool that enables them to gain greater understanding of the likely air flow and heat transfer processes occurring within and around building spaces given specified boundary conditions which may include the effects of climate, internal energy sources and HVAC systems.

There are 3 main steps in a typical CFD analysis (IES-VE (2013)):

1- Pre-processing Stage – Definition of the Problem:

- Define the model geometry
- Define the computational domain
- Define the boundary and initial conditions
- Define the grid/mesh
- Define all the necessary solver parameters

2- Solution Stage – Solving the Governing Equations:

- Inspect the progress of the run
- Adjust solver parameter criteria if necessary to achieve convergence

3- Post-processing Stage – Analysis of Results:

- Visualisation of results and reporting.

The two first steps will be described in this chapter, and the third step will be discussed in chapter 5.

4.2 Inputs

Model geometry

The model geometry was made in de *ModelIT* module and then was imported to the *MicroFlo* module. The dimensions of the room were assigned according to what is presented in section 3.1.1. The figure 4.1 shows the model geometry with the location of the supplies and extracts.

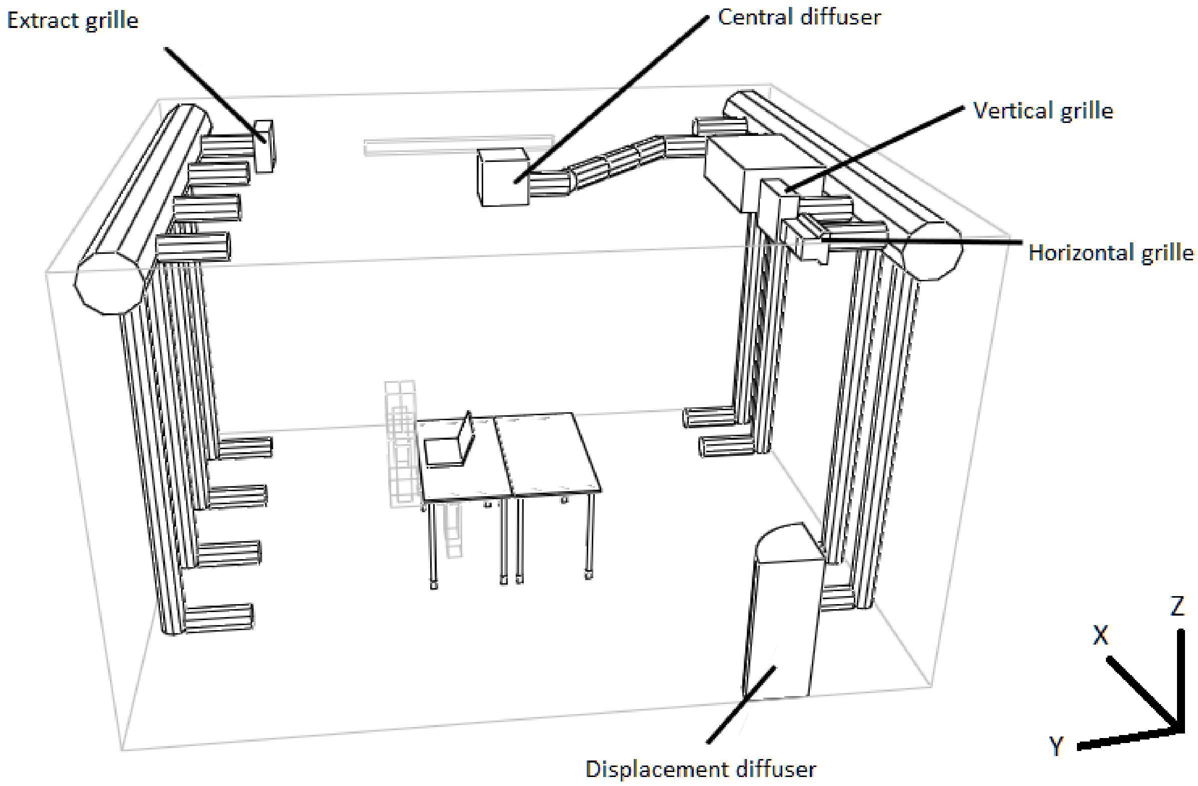

Figure 4.1: Model geometry

Ventilation schemes

The table 4.1 shows the schemes used in this study. The 13 schemes represents the most wide schemes used on HVAC installations in Portugal.

Turbulence model

In this software there are two types of turbulence models:

- κ-ϵ: the most generally accepted and widely used turbulence model. The κ-ϵ model calculates turbulent viscosity for each grid cell throughout the calculation domain by solving two additional partial differential equations, one for turbulence kinetic energy and the other for its rate of dissipation.

Scheme	Type of Ventilation	Supply	Extract
1	mixture	horizontal grille	extract grille
2	mixture	horizontal grille	extract grille
3	mixture	vertical grille	extract grille
4	mixture	central diffuser	extract grille
5	displacement	displacement diffuser	extract grille
6	mixture	horizontal grille	extract grille
7	mixture	horizontal grille	extract grille
8	mixture	vertical grille	extract grille
9	mixture	central diffuser	extract grille
10	mixture	vertical grille	extract grille
11	mixture	central diffuser	extract grille
12	displacement	displacement diffuser	extract grille
13	displacement	displacement diffuser	extract grille

Table 4.1: Simulations schemes

- Constant effective viscosity: This model does not attempt to account for the transport of turbulence but offers the user a much faster, much more approximate method of accounting for turbulence than the κ-ϵ model. The turbulent viscosity is assumed constant throughout the calculation domain and it can be defined either by specifying an absolute value or a multiplier, which is applied to the molecular laminar viscosity. This specification of turbulent viscosity is at best approximate but does allow a number of scenarios to be investigated for key features, prior to using the κ-ϵ model.

We used the first option. The software use de κ-ϵ model like is describe in Versteeg and Malalasekera (1995).

Surface heat transfer

There are two options:

- MicroFlo will calculate heat transfer between solid surfaces and the air
- Users can define the surface heat transfer coefficients

It was used the first option.

Boundary conditions

We have seen in the table 3.5 the boundary conditions used in the 13 ventilation schemes.

Grid settings

We can define the default grid spacing and merge tolerance. The merge tolerance enables grid lines that are separated by a distance less than the tolerance, to be merged into a single grid line to minimise superfluous gridding. Ensure Grid merge tolerance is less than or equal to the thickness of the smallest component. The table 4.2 shows the number of cells and the aspect ratio used in the simulations.

Scheme	Nr: of Cells X	Nr: of Cells Y	Nr: of Cells Z	Total Cells	Aspect Ratio
1	100	96	47	451200	17:1
2	100	96	47	451200	17:1
3	100	96	48	460800	17:1
4	98	96	48	451584	17:1
5	98	95	51	474810	17:1
6	100	96	47	451200	17:1
7	100	96	47	451200	17:1
8	100	96	48	460800	17:1
9	99	96	48	456192	18:1
10	100	96	48	460800	17:1
11	99	96	48	456192	18:1
12	98	95	51	474810	17:1
13	98	95	51	474810	17:1

Table 4.2: Grid used in simulations schemes

Initial conditions

The initial velocity in the x, y and z directions was set to 0 m/s in all schemes. The initial room air temperature was set to 22°C in schemes 1 to 5 and 25°C in the other ones. Quicker convergence may be achieved if we choose an initial values close to that of the converged solution.

Discretisation

The software give us three options:

- Upwind
- Hybrid
- Power law

Early attempts to derive CFD solution schemes using the traditional central difference approach to discretisation were found to fail for flows with high absolute value of Peclet number, due to the highly non-linear relationship between the transported variable and the transport distance. The basic remedy for this behaviour is to allow the finite volume cell interface values of the convected properties to take on the upwind grid point values; this method is known as the 'upwind' scheme. Advanced users who wish to use an alternative scheme may opt for the arguably more accurate but more computationally expensive hybrid and power-law schemes.(IES-VE (2013))

4.3 Characterization of sources

Heat sources

According with Wilkins (1998), ASHRAE (2009) and Wilkins and Hosni (2011), nameplate values on computers should be ignored when performing load calculations. Power consumption of laptop computers is relatively small: depending on processor speed and screen size, it varies from about 15 to 40 W. Actual power consumption for laptops is about 25% of nameplate value. Typical heat gain values for an computers with a 2.0 GHz processor, 2 GB RAM, 430 mm screen is 36 W.

The room lighting is achieved through a luminaire consisting of a fluorescent lamp with the power consumption of 58 W.

Figure 4.2 shows the thermal interaction of the human body with its environment. The total metabolic rate M within the body is the metabolic rate required for the person's activity M_{act} plus the metabolic level required for shivering M_{shiv} (should shivering occur). A portion of the body's energy production may be expended as external work W; the net heat production $M - W$ is transferred to the environment through the skin surface q_{sk} and respiratory tract q_{res} with any surplus or deficit stored S, causing the body's temperature to rise or fall. The heat emission of a person is calculated with equation (4.1).ASHRAE (2009)

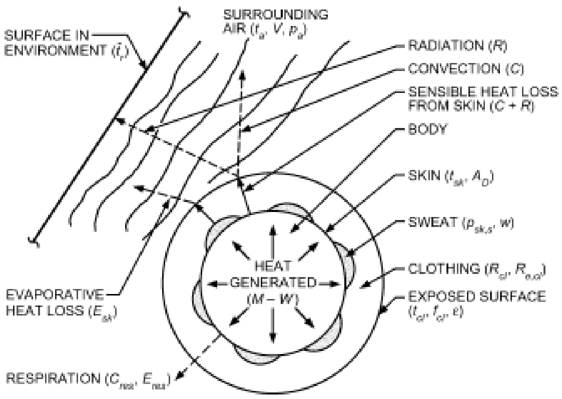

Figure 4.2: Thermal Interaction of Human Body and Environment. ASHRAE (2009)

$$M - W = q_{sk} + q_{res} + S = (C + R + E_{sk}) + (C_{res} + Eres) + (S_{sk} + S_{cr}) \tag{4.1}$$

where:

- M = rate of metabolic heat production, W/m^2;
- W = rate of mechanical work accomplished, W/m^2;
- q_{sk} = total rate of heat loss from skin, W/m^2;
- q_{res} = total rate of heat loss through respiration, W/m^2;

- $C + R$ = sensible heat loss from skin, W/m^2;
- E_{sk} = total rate of evaporative heat loss from skin, W/m^2;
- C_{res} = rate of convective heat loss from respiration, W/m^2;
- E_{res} = rate of evaporative heat loss from respiration, W/m^2;
- S_{sk} = rate of heat storage in skin compartment, W/m^2;
- S_{cr} = rate of heat storage in core compartment, W/m^2.

Heat dissipates from the body to the immediate surroundings by several modes of heat exchange: sensible heat flow $C + R$ from the skin; latent heat flow from sweat evaporation E_{rsw} and from evaporation of moisture diffused through the skin E_{dif} ; sensible heat flow during respiration C_{res} ; and latent heat flow from evaporation of moisture during respiration Eres. Sensible heat flow from the skin may be a complex mixture of conduction, convection, and radiation for a clothed person; however, it is equal to the sum of the convection C and radiation R heat transfer at the outer clothing surface (or exposed skin). Sensible and latent heat losses from the skin are typically expressed in terms of environmental factors, skin temperature t_{sk} , and skin wettedness w_{sk}. Factors also account for the thermal insulation and moisture permeability of clothing. The independent environmental variables can be summarized as air temperature t_a, mean radiant temperature $\bar{t_r}$, relative air velocity v, and ambient water vapor pressure p_a. The independent personal variables that influence thermal comfort are activity and clothing. The rate of heat storage in the body equals the rate of increase in internal energy (ASHRAE (2009))

Estimating metabolic rates is difficult. A unit used to express the metabolic rate per unit DuBois area is the met, defined as the metabolic rate of a sedentary person (seated, quiet): 1 met = 58.1 W/m^2. DuBois and DuBois (1916) gives an estimation of this body surface area based on the weight and the height of a person. It is defined by the equation (4.2).

$$A_{du} = 0,202 W_b^{0,425} H_b^{0,725} \tag{4.2}$$

where:

- A_{du} = DuBois surface area, m^2;
- W_b = body mass, kg;
- H_b = body height, m.

In this study was considered that the occupation of the person is fixed. Being the study space an office, it was considered that the occupant exerted an fixed activity: filing seated. For this activity Buskirk (1960), Webb (1964) and Passmore and Durnin (1967) consider the mean value of 70 W/m^2 (1,2 met). We used the following assumptions: W_b = 70 Kg and H_b = 1,70 m. Wich give us the value of 126,38 W for the total heat generation of the person.

Heat loss by persons is not uniform, because of the different resistances of the garment to transfer heat generated inside the bodies. Typically in male subjects when they are inside buildings, and in this particular case an office, the only part of the body where the skin is directly exposed to the ambient air is the head. Froese and Burton (1957) estimates that the head heat loss was linearly related to air temperature by the regression equation (4.3).

$$q_H = 284,8 - 7,55\theta_a \tag{4.3}$$

where:

- q_H = head heat loss, $KCal/m^2$;

- θ_a = air temperature, °C.

O'Sullivan and Schmitz (2007) refers that the area of human skin on the head corresponds to 9% of the total area of the body. In this study we divide de human body in 3 parts: front head, rear head and body. We can see in the table 4.3 the corresponding surface area.

	Percentage of skin area [%]	Surface area m^2
Front Head	4,5	0,081
Rear Head	4,5	0,081
Body	91	1,642

Table 4.3: Percentage of skin surface area of the person model

Sensible Heat W					
Scheme	Body	Front head	Rear head	Laptop	Luminaire
1	54,87	9,03	9,03	36	58
2	54,87	9,03	9,03	36	58
3	54,87	9,03	9,03	36	58
4	54,87	9,03	9,03	36	58
5	54,87	9,03	9,03	36	58
6	47,78	12,57	12,57	36	58
7	47,78	12,57	12,57	36	58
8	47,78	12,57	12,57	36	58
9	47,78	12,57	12,57	36	58
10	47,78	12,57	12,57	36	58
11	47,78	12,57	12,57	36	58
12	47,78	12,57	12,57	36	58
13	47,78	12,57	12,57	36	58

Table 4.4: Values of the model heat sources.

Moisture g/h			
	Body	Front head	Rear head
All schemes	34,58	13,71	1,71

Table 4.5: Values of the model moisture sources.

Moisture sources

According with ASHRAE (2009) the latent heat of an moderately active office work is about 42,3% of total heat, which corresponds to 53,46 W.

People add water vapour to the indoor air by respiration and transpiration. McCutchan and Taylor (1951) present data on the effect of humidity on human respiration. They determined that with room temperature in the range of 20°C to 24°C the expired air is at 33,2°C and 88,2% relative humidity (RH). The RH of the expired air drops slightly with a drop in indoor air humidity. Typical respiration rates are 0,3 kg/day. Ferguson and Martin (1991) provide measured transpiration rates from burn wounds, but, for comparison reasons, also provide transpiration rates from healthy skin, in terms of skin diffusion resistance (the diffusion resistance was defined in terms of absolute humidity gradient expressed in terms of weight per unit of volume). Taking a typical range transpiration rates calculated for ambient conditions of 20°C at 50% RH are in the range of 0,5 to 1,4 kg/day per adult person at rest. The sum of respiration and transpiration is in the range of 0,8 to 1,7 kg/day for an adult at rest, or approximately 30 to 70 g/h. TenWolde and Pilon (2007) indicate that moisture release from humans is categorized as independent of humidity, and 50 g/h is used for an average adult.

CO_2 sources

For modelling the person has been necessary to determine the CO_2 emission. It is possible to relate the emission of CO_2 to the O_2 consumption. This relation is give by the respiratory quotient (RQ). This ratio is the rate of oxygen O_2 used and transformed into carbon dioxide when humans are breathing. Its value varies between 0,7 and 1 but the normal value for moderate activity and normal diet is 0,82 (McIntyre (1980)) or 0,83 (Alfano et al. (2010)). We can see this relationship in equation (4.4).

$$\dot{V}_{CO_2} = RQ * \dot{V}_{O_2} \tag{4.4}$$

The flow rate of O_2 consumption depends on the level of activity or metabolic rate M of the person and the size of this person via the body surface area. As an indicator the Table 4.6 gives average values of oxygen consumption for different levels of work and for a normal adult man.

Alfano et al. (2010) give us a relationship between the flow rate of O_2, the DuBois surface area, the respiratory quotient and the metabolic rate, as we can see in equation (4.5). In this equation the oxygen consumption of a person is given in [l/s].

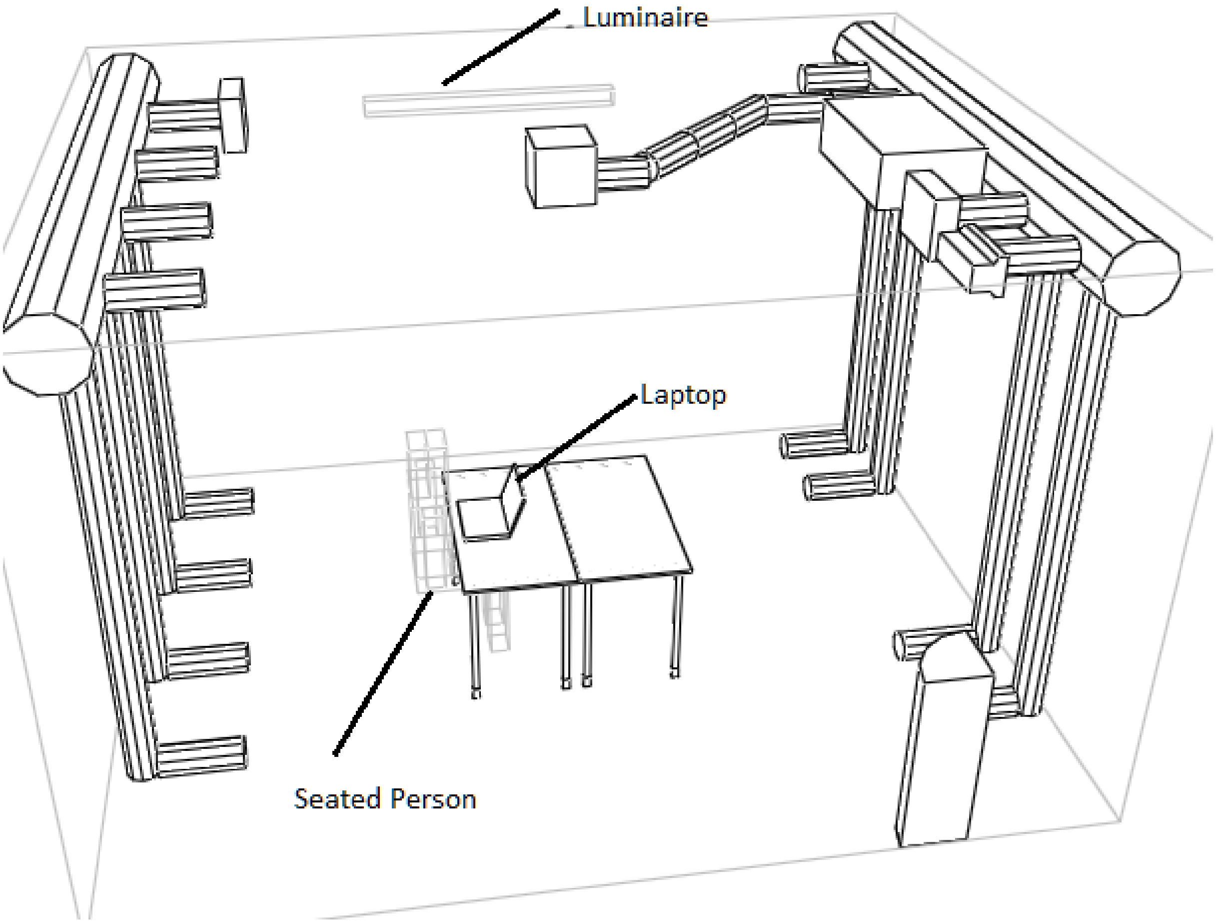

Figure 4.3: Heat sources - SATEC

$$\dot{V}_{O_2} = \frac{0,00276 A_{du} M}{0,23 RQ + 0,77} \tag{4.5}$$

From the previous equations (4.4) and (4.5) we can establish the equation for the CO_2 emission of one person [l/s].

$$\dot{V}_{CO_2} = RQ.\frac{0,00276 A_{du} M}{0,23 RQ + 0,77} \tag{4.6}$$

Using (4.6) the $\dot{V_{CO_2}}$ of the person is 5,17 l/s. We can see in Linde Group (2011) that the value of the density of CO2 used in the experimental measurements is 1,528, which give us an CO_2 emission rate of 28,41 g/h. The emission of CO_2 is only in front head (figure 4.4). The table 4.7 resumes the values used on the CFD simulations.

We can see in figure 4.3 the location of the sources and in tables 4.4, 4.5 and 4.7 the value of the sources quantities for the different schemes simulated.

Level of Exertion	Heart Rate bpm	Oxygen Consumed ml/s
Light work	<90	<8
Moderate work	90 to 110	8 to 16
Heavy work	110 to 130	16 to 24
Very heavy work	130 to 150	24 to 32
Extremely heavy work	150 to 170	>32

Table 4.6: Heart Rate and Oxygen Consumption at Different Activity Levels. Astrand and Rodahl (1977)

CO_2 emissions g/h			
	Body	Front head	Rear head
All schemes	0	28,41	0

Table 4.7: Values of the CO_2 emissions.

4.4 Outputs

The MicroFlo viewer application can be used to display the CFD results. A combination of vectors and contours can be chosen for create the images. In addition to studying the air flow through the space, it is possible to model the concentrations of CO_2, moisture and CO, and the local mean age of air. The CO, CO_2 and moisture calculations are performed automatically when the appropriate boundary condition is set on a supply diffuser or a source term is introduced via a fluid component. The local mean age of air is only calculated for ventilated rooms, where there is an exchange with the outside. This is a measure of the time a parcel of air has been in the simulation domain, after allowing for advection and diffusion. Regions of high values indicate places of poor ventilation. The air change effectiveness is derived from the local mean age according to ASHRAE (2005).

The full list of output variables the MicroFlo viewer displays is listed below:

- Velocity vector
- Velocity contour
- Temperature contour
- Pressure contour
- H_2O mass fraction contour
- CO_2 mass fraction contour
- Local Mean Age of Air contour
- Filled velocity contour
- Filled temperature contour

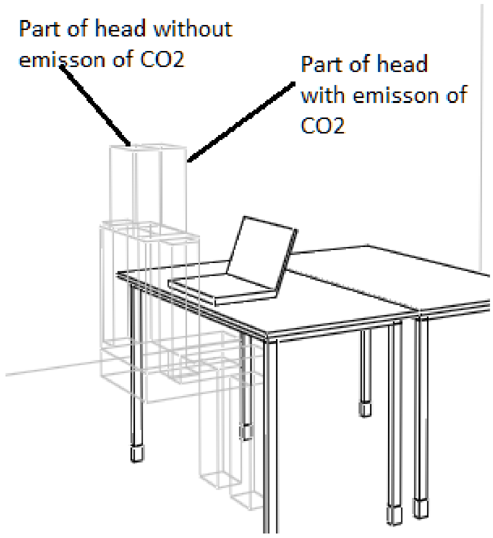

Figure 4.4: Model of person

- Filled pressure contour
- Filled H_2O mass fraction contour
- Filled CO_2 mass fraction contour
- Filled Local Mean Age of Air contour
- Dry resultant temperature contour
- PMV contour
- PPD contour
- Comfort contour
- Filled dry resultant temperature contour
- Filled PMV contour
- Filled PPD contour
- Filled comfort contour

The MicroFlo viewer can also be used to create surfaces of temperature, velocity, pressure, H_2O and CO_2 mass fractions, and the local mean age of air. These are displayed as virtual 'nets'. The MicroFlo viewer can display animated particle tracking where the user can specify the number and length of the particle tracks.

Chapter 5

Results

5.1 Experimental results

5.1.1 Contaminant Removal Effectiveness

As an example we show the calculation procedure for scheme 2. Following the procedure mentioned in the section 3.3.2, we obtain de values of fig. 5.1.

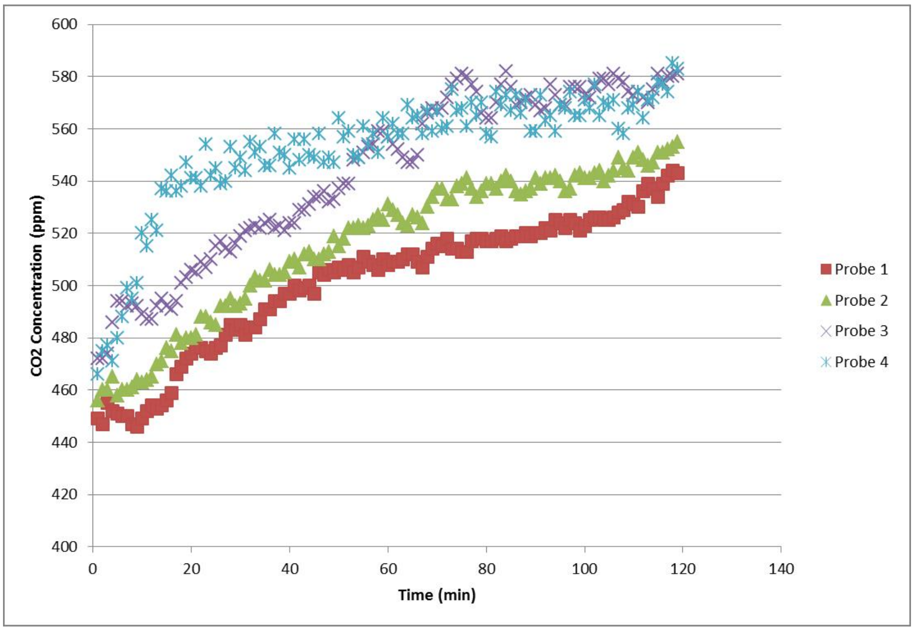

Figure 5.1: CO_2 values measured in CRE test scheme 2

The fig. 5.2 shows the final part of the measurements of fig. 5.1, where the CO_2 concentration measurements are in steady state.

Then the average of these values were calculated, as we can see in table 5.1.

Then we can calculate the ε^c with the equation 5.1:

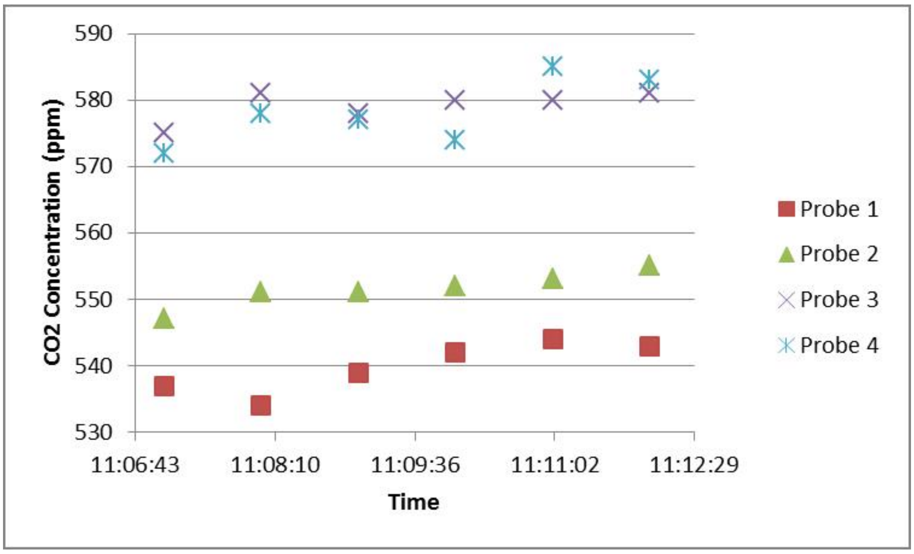

Figure 5.2: CO_2 values measured in CRE test scheme 2 - steady state zone

Time	Probe 1 (ppm)	Probe 2 (ppm)	Probe 3 (ppm)	Probe 4 (ppm)
11:06:01	539	546	570	571
11:07:01	537	547	575	572
11:08:01	534	551	581	578
11:09:01	539	551	578	577
11:10:01	542	552	580	574
11:11:01	544	553	580	585
11:12:01	543	555	581	583
avg value	539,71	550,71	577,86	577,14

Table 5.1: Average values of CO_2 concentrations - scheme 2

$$\varepsilon^c = \frac{C_e}{< C >} \tag{5.1}$$

where $< C >$ is equal to the room average concentration:

$$< C >= \frac{539,71 + 550,71 + 577,86}{3} = 556,10ppm \tag{5.2}$$

then:

$$\varepsilon^c = \frac{577,14}{556,10} = 1,04 \tag{5.3}$$

As we have seen before the local air quality Index (ε_p^c) is defined as:

$$\varepsilon_p^c = \frac{C_e}{C_p} \tag{5.4}$$

To probe 1, 2 and 3 we have:

$$\varepsilon_{p1}^c = \frac{C_e}{C_p} = \frac{577,14}{539,71} = 1,07 \tag{5.5}$$

$$\varepsilon_{p2}^c = \frac{C_e}{C_p} = \frac{577,14}{550,71} = 1,05 \tag{5.6}$$

$$\varepsilon_{p3}^c = \frac{C_e}{C_p} = \frac{577,14}{577,86} = 0,99 \tag{5.7}$$

According with NTvvs 114 (1997) the accuracy can be divided into the following two components:

- accuracy of gas analyser, m_G (at a 95% confidence level)
- Fluctuation of CO_2 concentration, m_C, $m_C = 2s$

The total accuracy of each measuring point, m [ppm], is then calculated as:

$$m = \sqrt{m_G^2 + m_C^2} \tag{5.8}$$

The total accuracy of the measurements is then calculated as:

$$\Delta\varepsilon^c = \sqrt{\left(\frac{m_{Ce}}{C_e}\right)^2 + \left(\frac{m_{<C>}}{<C>}\right)^2} \tag{5.9}$$

The calculation of CRE was performed similarly for the other 12 schemes. The fig. 5.3 and table 5.2 show the results for all the 13 schemes.

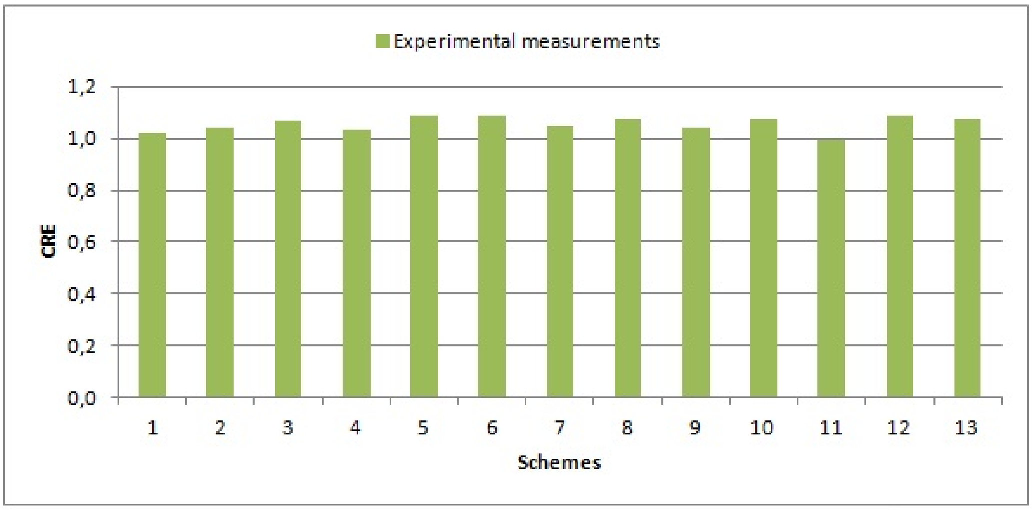

Figure 5.3: CRE experimental results

	Scheme												
	1	2	3	4	5	6	7	8	9	10	11	12	13
CRE	1,02	1,04	1,07	1,04	1,09	1,09	1,05	1,07	1,04	1,07	0,99	1,09	1,08
Error	0,12	0,11	0,12	0,12	0,12	0,12	0,11	0,12	0,12	0,11	0,12	0,12	0,12

Table 5.2: CRE experimental results

5.1.2 Air-Change Efficiency

Nordtest method

The calculations of the air-change efficiency according with the Nordtest method were done taking in count the recommendations of NTvvs 019 (1988) and NTvvs 047 (1985). As an example we show the calculation procedure for scheme 2. Following the procedure mentioned in the section 3.3.1, we obtain de values of fig. 5.4.

The fig. 5.5 shows the concentration readings in the extract duct in the decay zone.

The mean age of the room air is calculated from the weighted are under the curve by equation 5.10 (Mathisen and Skaret (1989); Sandberg et al. (1995)). For simplicity the terms in the equation are denoted I, II, III and IV.

The terms II and IV represent the last part of the decay that has to be extrapolated from the first part of the decay curve. λ is negative for step-down tests, but is used with positive sign in the formulas below.

Figure 5.6 explains the symbols used in the expressions for I and III.

The nominal time constant is then calculated by equation 5.11 and the air change efficiency by equation 5.12.

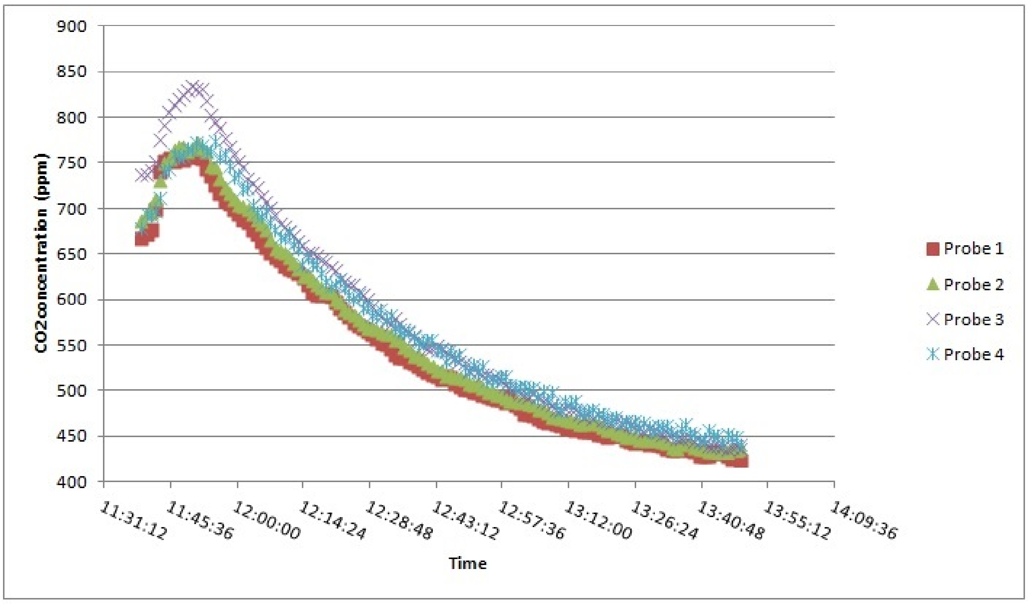

Figure 5.4: CO_2 values measured in ACE test (Nordtest method) scheme 2

$$< \bar{\tau} >= \frac{\sum_{i=1}^{i=n} \left[\frac{C_i + C_{i-1}}{2} \cdot (t_i - t_{i-1}) \cdot \frac{t_i + t_{i-1}}{2} \right] + \frac{C_n}{\lambda} \cdot \left[\frac{1}{\lambda} + t_n \right]}{\sum_{i=1}^{i=n} \left[\frac{C_i + C_{i-1}}{2} \cdot (t_i - t_{i-1}) \right] + \frac{C_n}{\lambda}} = \frac{I + II}{III + IV} \tag{5.10}$$

$$\tau_n = \frac{\sum_{i=1}^{i=n} \left[\frac{C_i + C_{i-1}}{2} \cdot (t_i - t_{i-1}) \right] + \frac{C_n}{\lambda}}{C_0} = \frac{III + IV}{C_0} \tag{5.11}$$

$$\varepsilon^a = \frac{\tau_n}{2. < \bar{\tau} >} \tag{5.12}$$

It was use a spreadsheet to analyse measurement data and calculate the air change efficiency according with the example in Mundt et al. (2003). The table A.1 shows the calculation example for scheme 2. According with the outputs of the spreadsheet the air change efficiency is:

$$\varepsilon^a = \frac{84,9}{2 \cdot 53,4} = 0,79 \tag{5.13}$$

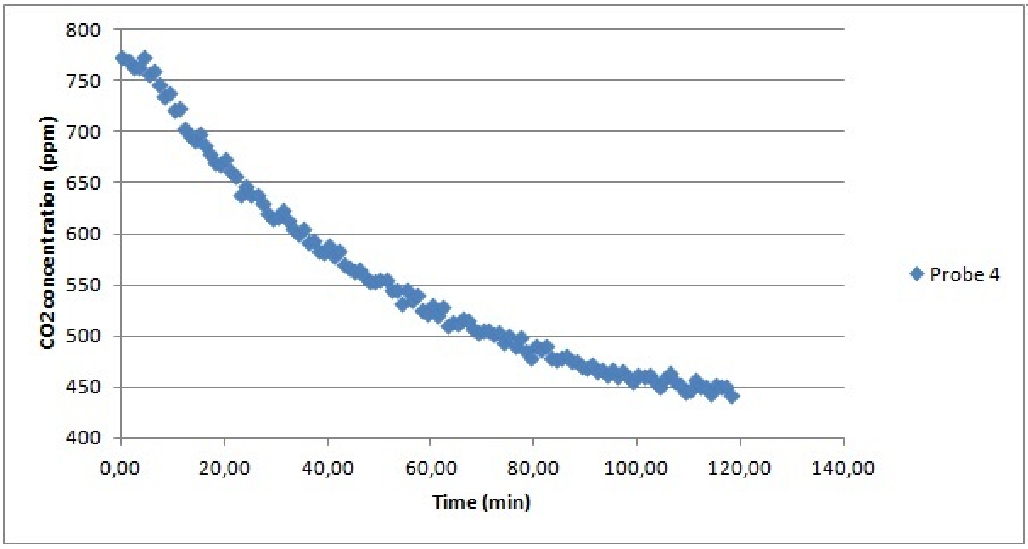

Figure 5.5: CO_2 values measured in ACE test scheme 2

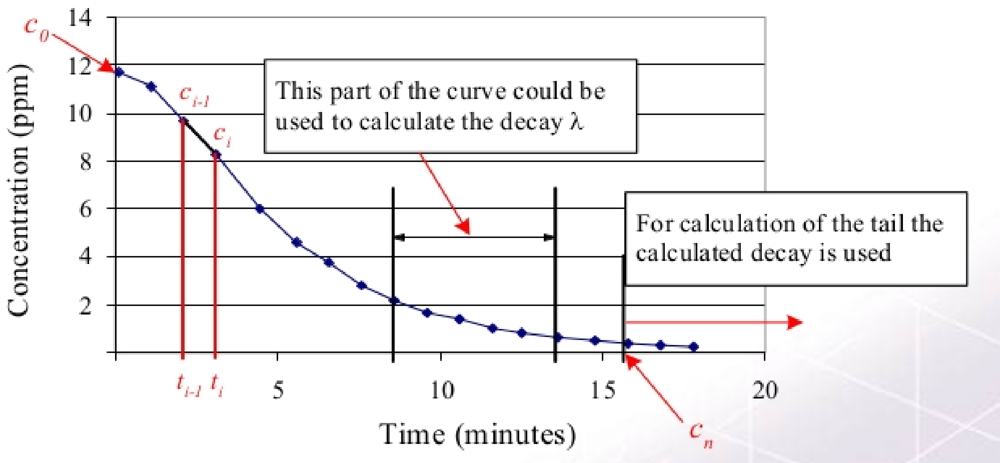

Figure 5.6: Decay of tracer gas concentration. Mundt et al. (2003)

As we have seen before the local air change index (ε_p^a) is defined as:

$$\varepsilon_p^a = \frac{\tau_n}{\bar{\tau}_p} \tag{5.14}$$

Using a spreadsheet like as we saw above, for the 3 locals the ε_p^a comes:

$$\varepsilon_{p1}^a = 0,77; \varepsilon_{p2}^a = 0,77; \varepsilon_{p3}^a = 0,75 \tag{5.15}$$

According with NTvvs 047 (1985), when we measuring in one extract duct only, the accuracy has been determinate to be ±16 %.

A systematic error occur when, due to the occurrence of leakages in the chamber structure, not all air leaving the ventilated space passes the measuring points. The mean-age predicted by the method will then be lower than the true value.

The calculation of ACE (Nordtest method) was performed similarly for the other 12 schemes. The fig. 5.7 and table 5.3 show the results for all the 13 schemes.

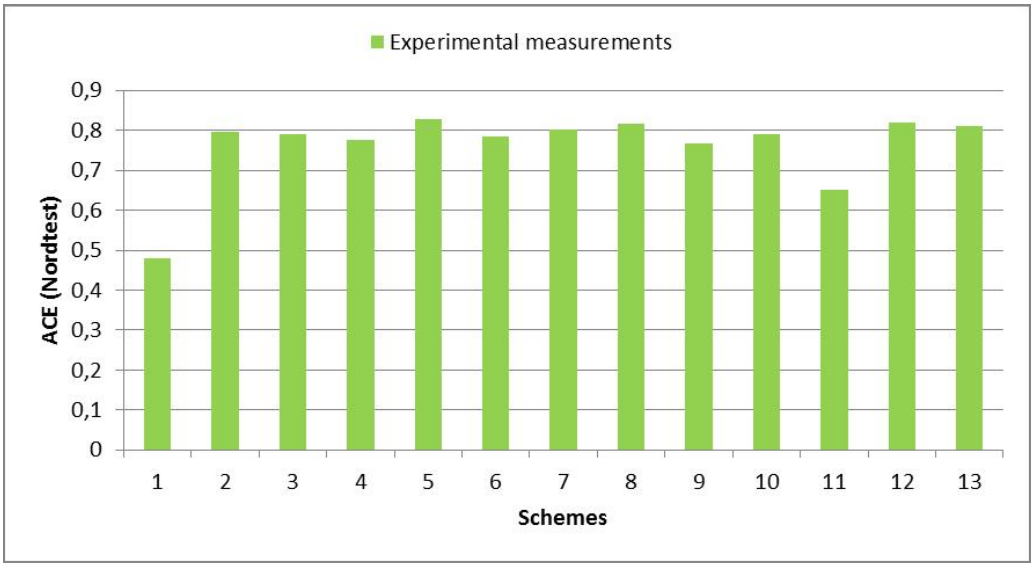

Figure 5.7: ACE (Nordtest method) experimental results

	Scheme												
	1	2	3	4	5	6	7	8	9	10	11	12	13
ACE	0,48	0,79	0,79	0,78	0,83	0,78	0,80	0,82	0,77	0,79	0,65	0,82	0,81
Error	0,08	0,13	0,13	0,12	0,13	0,13	0,13	0,13	0,12	0,13	0,10	0,13	0,13

Table 5.3: ACE (Nordtest method) experimental results

ASHRAE 129 method

The calculations of the air-change efficiency according with the ASHRAE 129 method were done taking in count the recommendations of ANSI/ASHRAE 129 (1997). As an example we show the calculation procedure for scheme 2. Following the procedure mentioned in the section 3.3.1, we obtain de values of fig. 5.4.

The ASHRAE 129 method is different of the Nordtest method only in the equations used. While the Nordtest method compares de nominal time constant with the local air age in the extract duct, the ASHRAE 129 method compares the nominal time constant with the arithmetic average of the ages of air measured at breathing level within the test space. For this reasons the same test were used to determine the two indicators.

In the same way to the Nordtest method, it was used a spreadsheet to analyse measurement data and calculate the air change efficiency according to the ASHRAE 129. The table A.2 shows the calculation example for scheme 2. According with the outputs of the spreadsheet the air change efficiency is:

$$E = \frac{\tau_n}{A_{avg}} = \frac{93,747}{81,231} = 1,15 \tag{5.16}$$

As we saw earlier in the ASHRAE 129 method there is no concept of local air change index.

According with ANSI/ASHRAE 129 (1997), the accuracy for the step down test has been determinate to be ±16 %.

The calculation of ACE (ASHRAE 129 method) was performed similarly for the other 12 schemes. The fig. 5.8 and table 5.4 show the results for all the 13 schemes.

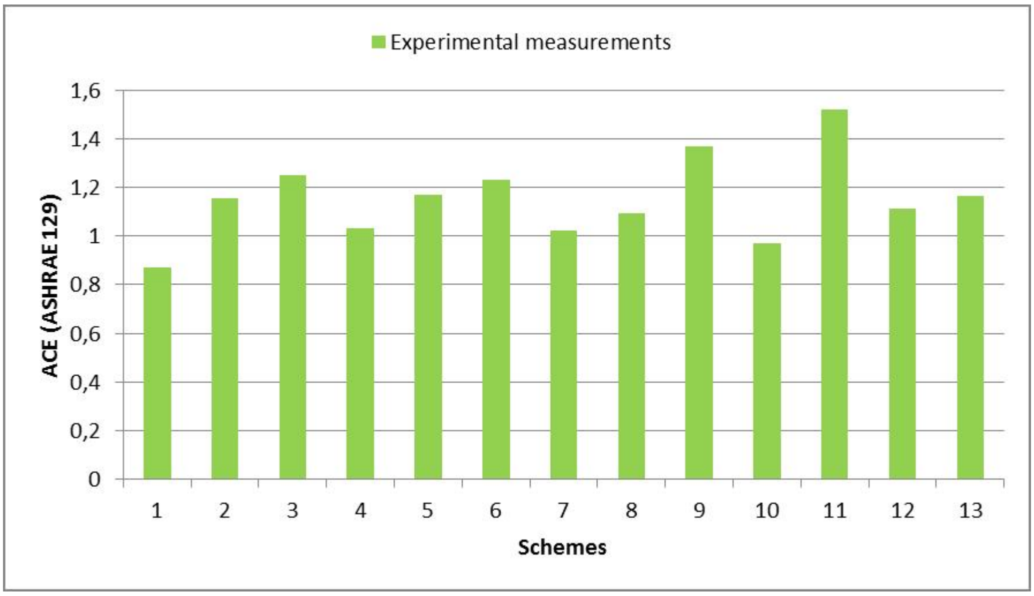

Figure 5.8: ACE (ASHRAE 129 method) experimental results

5.2 CFD results

5.2.1 Contaminant Removal Effectiveness

As we saw earlier, it was used the *MicroFlo* module from the *IES-ve* software. As an example we show the calculation procedure for scheme 2. The fig. 5.9 and 5.10 shows the

	Scheme												
	1	2	3	4	5	6	7	8	9	10	11	12	13
ACE	0,87	1,15	1,25	1,03	1,17	1,23	1,02	1,10	1,38	0,97	1,52	1,12	1,16
Error	0,16	0,22	0,18	0,19	0,21	0,19	0,18	0,17	0,23	0,21	0,29	0,19	0,19

Table 5.4: ACE (ASHRAE 129 method) experimental results

CO_2 concentration for the plan X=2m and Y=3,63m respectively.

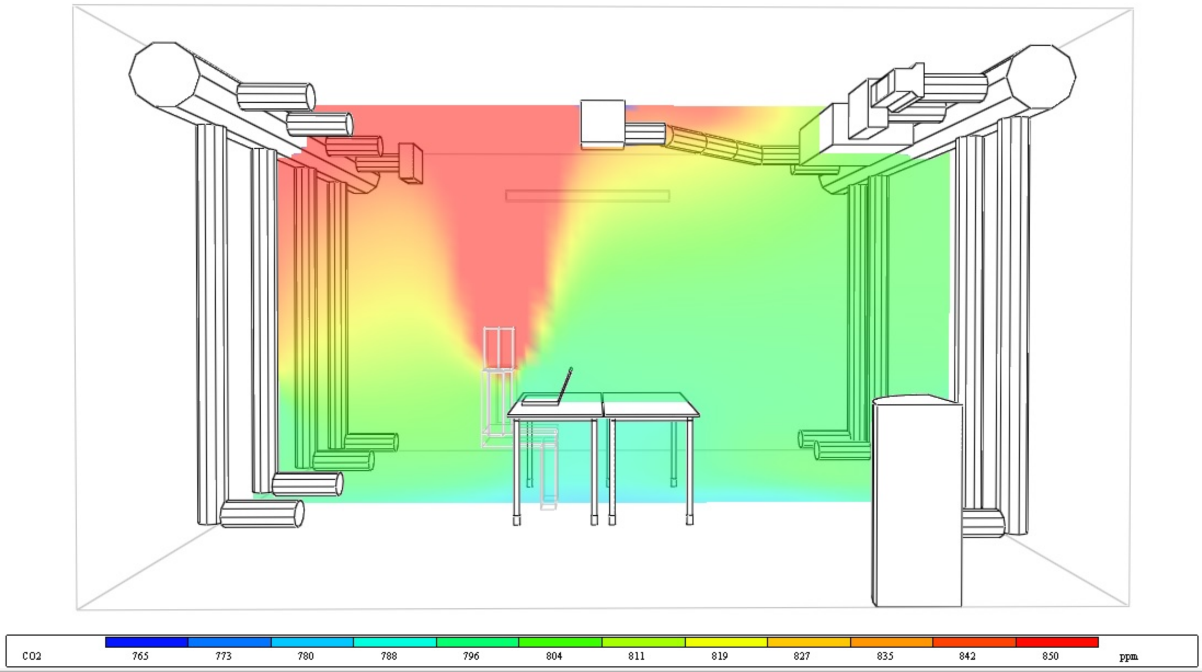

Figure 5.9: CO_2 concentration in plan x=2m - scheme 2

The value of the contaminant removal effectiveness comes:

$$\varepsilon^c = \frac{C_e}{<C>} \tag{5.17}$$

where $<C>$ is equal to the room average concentration:

$$<C> = \frac{807 + 826 + 846}{3} = 825,33ppm \tag{5.18}$$

then:

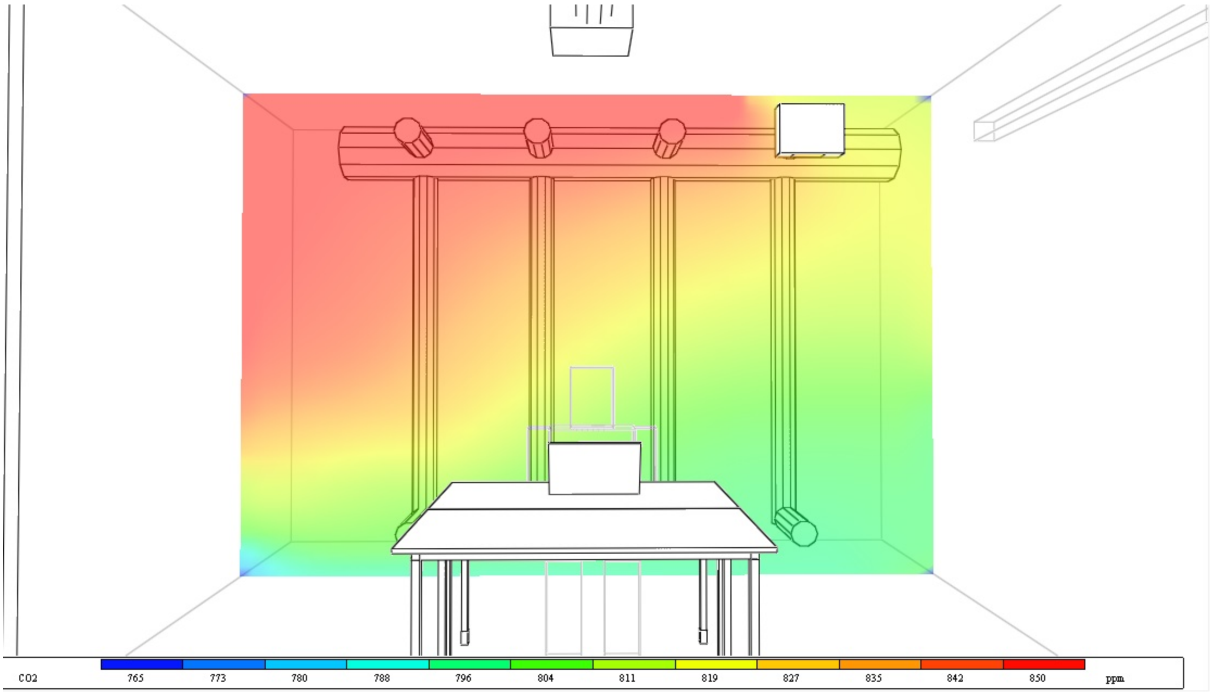

Figure 5.10: CO_2 concentration in plan y=3,63m - scheme 2

$$\varepsilon^c = \frac{828}{825,33} = 1,00 \tag{5.19}$$

As we have seen before the local air quality Index (ε_p^c) is defined as:

$$\varepsilon_p^c = \frac{C_e}{C_p} \tag{5.20}$$

To positions of probes 1, 2 and 3 we have:

$$\varepsilon_{p1}^c = \frac{C_e}{C_p} = \frac{828}{807} = 1,03 \tag{5.21}$$

$$\varepsilon_{p2}^c = \frac{C_e}{C_p} = \frac{828}{826} = 1,00 \tag{5.22}$$

$$\varepsilon_{p3}^c = \frac{C_e}{C_p} = \frac{828}{843} = 0,98 \tag{5.23}$$

The calculation of CRE was performed similarly for the other 12 schemes. The fig. 5.11 and table 5.5 show the results for all the 13 schemes.

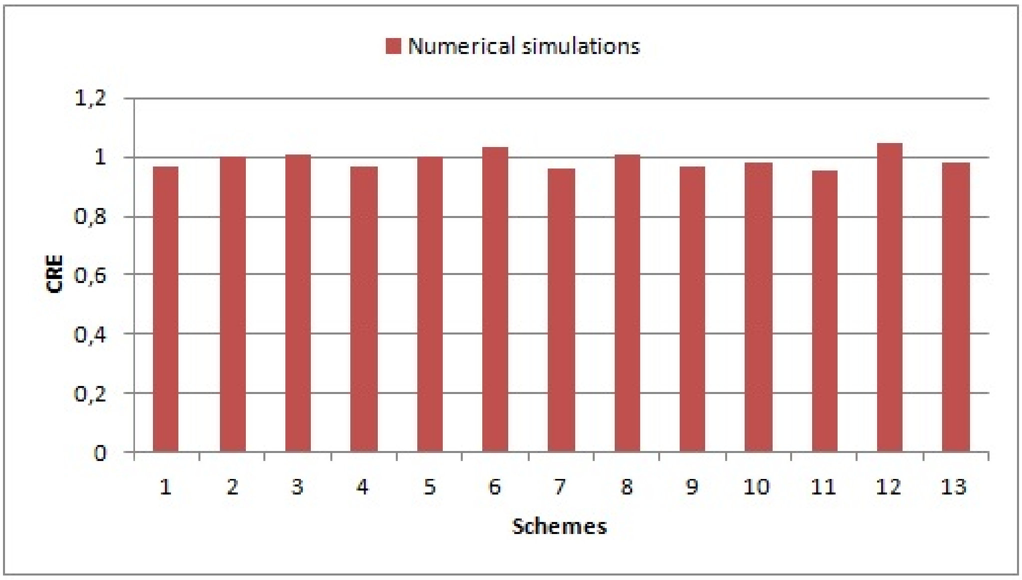

Figure 5.11: CRE numerical results

	Scheme												
	1	2	3	4	5	6	7	8	9	10	11	12	13
ACE	0,97	1,00	1,01	0,97	1,00	1,03	0,96	1,01	0,97	0,98	0,96	1,05	0,98

Table 5.5: CRE numerical results

5.2.2 Air-Change Efficiency

Nordtest method

As an example we show the calculation procedure for scheme 2. The fig. 5.12 and 5.13 shows the local air age for the plan X=2m and Y=3,63m respectively.

The value of the air change efficiency according with the Nordtest method comes:

$$\varepsilon^a = \frac{\tau_n}{2. < \bar{\tau} >} \tag{5.24}$$

then:

$$\varepsilon^a = 0,67 \tag{5.25}$$

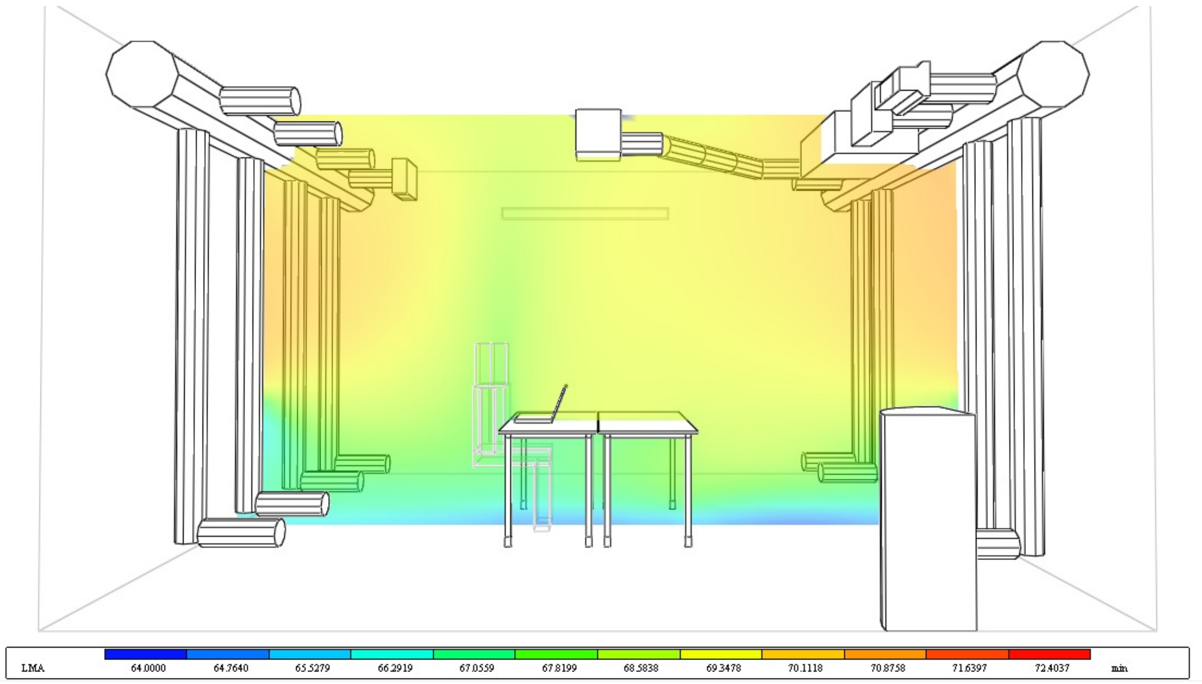

Figure 5.12: Local air age in plan x=2m - scheme 2

As we have seen before the local air change index (ε_p^a) is defined as:

$$\varepsilon_p^a = \frac{\tau_n}{\bar{\tau}_p} \tag{5.26}$$

Using a spreadsheet like as we saw above, for the 3 locals the ε_p^a comes:

$$\varepsilon_{p1}^a = 0,69; \varepsilon_{p2}^a = 0,68; \varepsilon_{p3}^a = 0,68 \tag{5.27}$$

The calculation of ACE (Nordtest method) was performed similarly for the other 12 schemes. The fig. 5.14 and table 5.6 show the results for all the 13 schemes.

	Scheme												
	1	2	3	4	5	6	7	8	9	10	11	12	13
ACE	0,50	0,67	0,52	0,56	0,61	0,57	0,60	0,52	0,74	0,66	0,99	0,53	0,59

Table 5.6: ACE (Nordtest method) numerical results

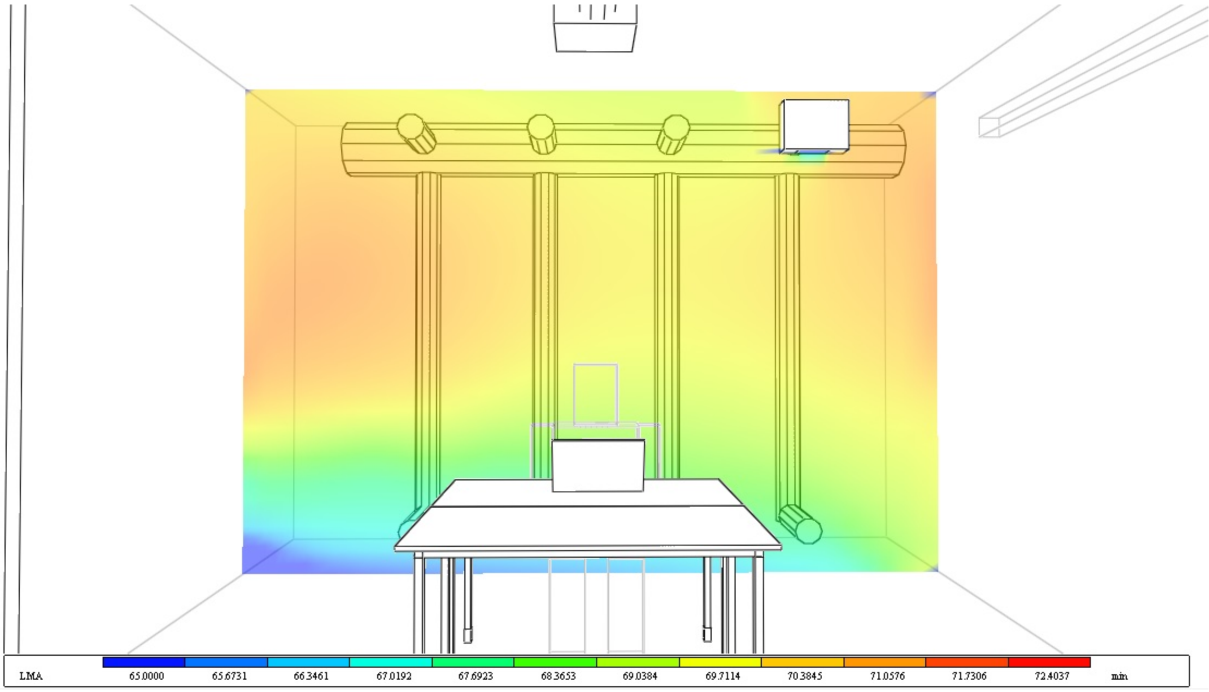

Figure 5.13: Local air age in plan y=3,63m - scheme 2

ASHRAE 129 method

As we saw earlier the ACE calculation through the Nordtest method and the ASHRAE 129 method differs only by the equations used. Consequently were used the values of the same CFD simulations. As an example we show the calculation procedure for scheme 2.

The value of the air change efficiency according with the ASHRAE 129 method comes:

$$E = \frac{\tau_n}{A_{avg}} \tag{5.28}$$

then:

$$E = \frac{\tau_n}{A_{avg}} = \frac{93,75}{68,78} = 1,36 \tag{5.29}$$

The calculation of ACE (Nordtest method) was performed similarly for the other 12 schemes. The fig. 5.15 and table 5.7 show the results for all the 13 schemes.

The overall results of tests and simulations are presented in table A.3.

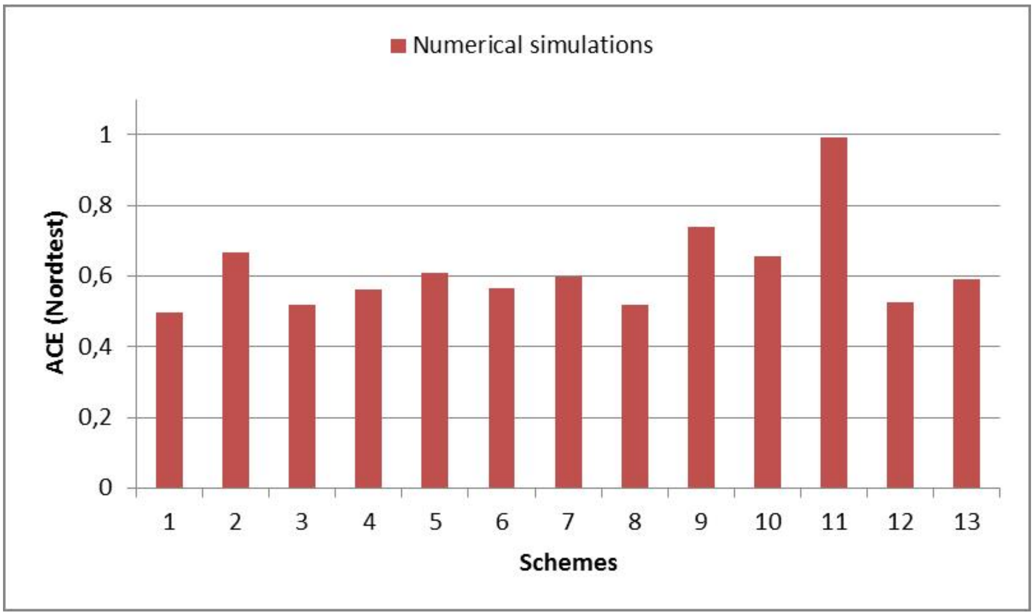

Figure 5.14: ACE (Nordtest method) numerical results

	Scheme												
	1	2	3	4	5	6	7	8	9	10	11	12	13
ACE	1,01	1,36	1,10	1,21	1,30	1,17	1,14	1,08	1,41	1,29	1,84	1,17	1,18

Table 5.7: ACE (ASHRAE 129 method) numerical results

5.3 Analysis of results

5.3.1 Contaminant Removal Effectiveness

The fig. 5.16 show the comparison of results between the experimental measurements and the numerical simulations. We can see that the values of the simulations are validated by experimental results. The differences in results are within the errors of experimental measurements.

We can see that the differences in the various schemes of diffusion are very small. In the experimental measurements the difference between the scheme with the highest value of CRE and the scheme with the lowest value is only 9,17%. In numerical simulations this difference is even smaller standing at 8,57%.

The fig. B.1 shows the comparison between the different schemes using the vertical grille. We can see that in the scheme 3 the concentration of CO_2 in the room presents a stratified distribution. This is advantageous because it allows the breathing zone to have a cleaner air that the extraction zone. In the schemes 8 and 10, this situation don't happen, because the supply air is warmer than the air of the room. We can see in this schemes that the concentration of CO_2 presents a more uniform distribution in the whole room. This situation is more evident in scheme 10, because the temperature difference is greater. Despite this situation these strategies provide a better distribution of the con-

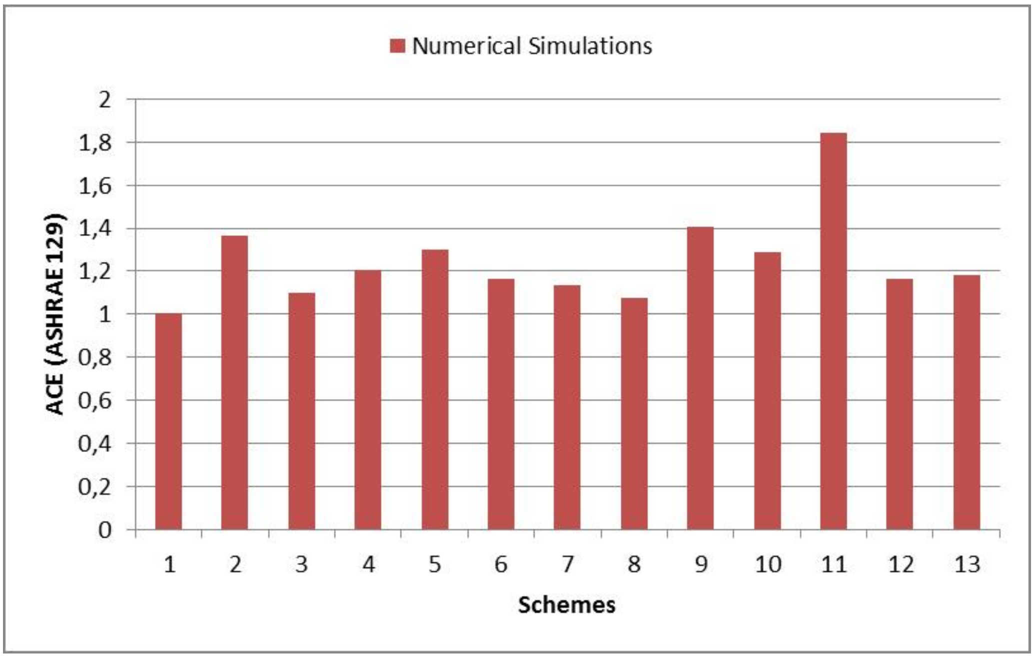

Figure 5.15: ACE (ASHRAE 129 method) numerical results

taminants that the strategies that use the horizontal grille.

The fig. B.2 shows the comparison between the different schemes using the horizontal grille. We can see that in scheme 6 and 7 there is plenty of fresh air that exits the extraction grille without crossing the breathing zone. This reflects the existence of a short circuit between the supply grille and the extraction grille. This situation is worse in the scheme 7 because the temperature difference between the supply air and the room is 18°C whereas in scheme 6 is only 3°C. This is also reflected in the CRE value wich is 1,03 and 0,96 for the scheme 6 and 7 respectively. We can also see that the CRE value of the scheme 2 is very slightly higher than the scheme 1, mainly due to the difference in flow velocity.

The fig. B.3 shows the comparison between the different schemes using the central diffuser. In the scheme 4 we can see that the breathing zone in front the person is fed with fresh air, which improve the conditions of air quality. In the schemes 9 and 11 the conditions are also good, but not so good that the conditions in scheme 4.

The fig. B.4 shows the comparison between the different schemes using the displacement diffuser. We can see in the scheme 5 that the concentration of CO_2 in the room presents a stratified distribution, similarly to what we have seen in the scheme 3. The schemes 12 and 13 presents a poor quality of the concentrations of CO_2 in the breathing zone, this situation is worse in scheme 13, where exists short circuit of the fresh air between the supply diffuser and the extraction grille.

The schemes that have better behaviour with respect to the distribution of CO_2 concentration are: 2, 3, 4, 5, 9 and 11. The other schemes are to be avoided. The table 5.9 gives us a overview of the schemes performance.

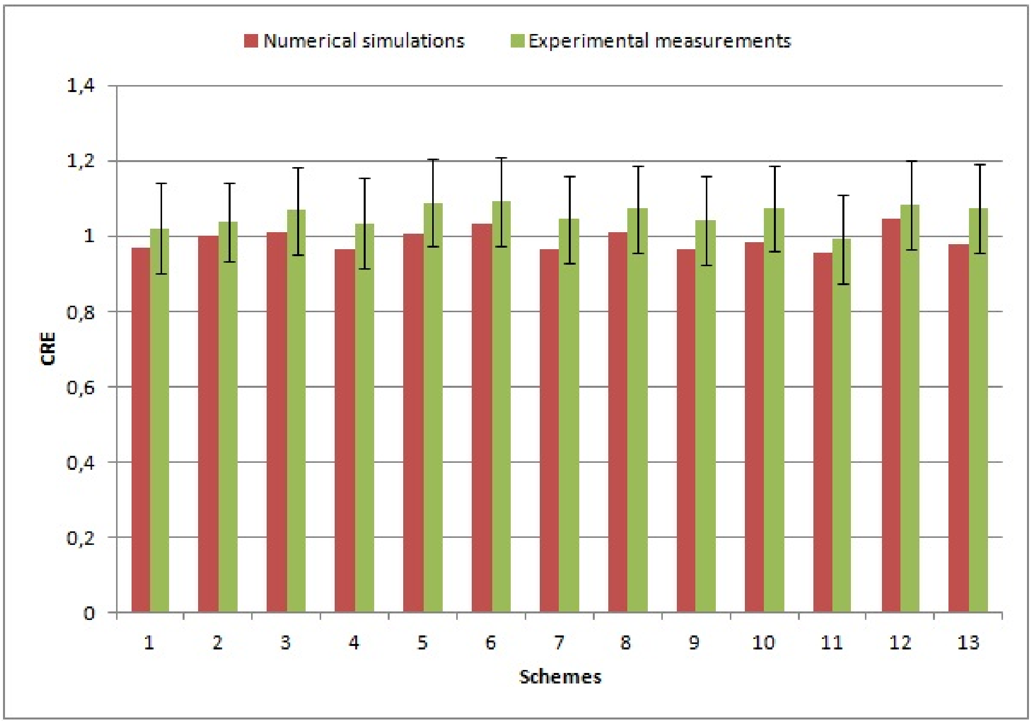

Figure 5.16: CRE - experimental measurements vs numerical simulations

5.3.2 Air Change Efficiency

The fig. 5.17 show the comparison of results between the experimental measurements and the numerical simulations for the Nordtest method. We can see that the values of the simulations don't correspond to the values of the experimental results. Only the schemes 1 and 9 are within the errors of the experimental measurements. Therefore, this software should not be used for calculating the ACE according to this method. This is consistent with the information provided by the manufacturer, since there is an indication that the software calculates the ACE according to the ASHRAE 129 method.

The fig. 5.18 show the comparison of results between the experimental measurements and the numerical simulations for the ASHRAE 129 method. We can see that the values of the simulations are validated by experimental results. The differences in results are within the errors of experimental measurements. We also can see that the schemes 6, 8, 12 and 13 presents a very good correspondence. The schemes 10 and 11 are the only that the value of the numerical simulations are slightly out of the error range of the experimental measurements (32% and 21% respectively).

Comparing to the values presented in the ANSI/ASHRAE 62.1 (2007) standard that we can see in fig. 2.25, we can conclude that the values of ACE of the schemes 1, 3, 5, 8 correspond to the expected. As for the other schemes the values have some differences (see table 5.8). The largest differences occur in schemes that use the central diffuser with supply of warm air (scheme 9 and 11). This is because in these schemes there is a large

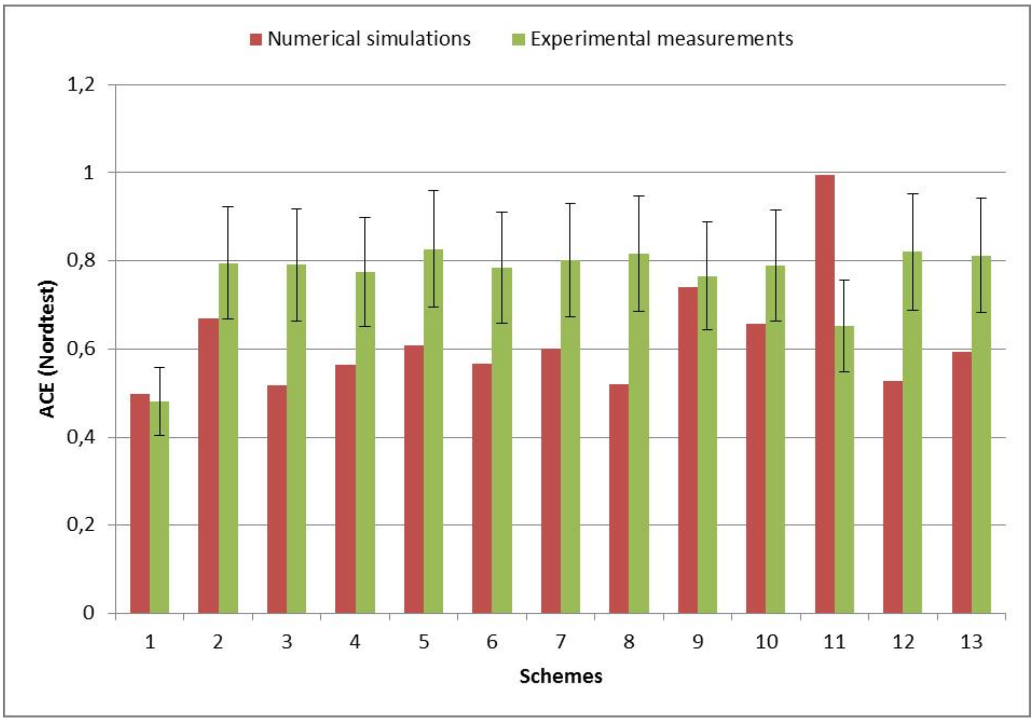

Figure 5.17: ACE (Nordest method) experimental results vs numerical simulation

amount of fresh air that reaches the extraction grille without going trough the breathing zone, causing that the mean air age of this air is small, making the value of ACE high.

Scheme	ASHRAE 62.1	Experimental Measurements	Numerical Simulations
1	1	0,87	1
2	1	1,15	1,36
3	1	1,25	1,10
4	1	1,03	1,21
5	1,2	1,17	1,30
6	0,8	1,23	1,17
7	0,8	1,02	1,14
8	0,8	1,10	1,08
9	0,8	1,37	1,41
10	0,8	0,97	1,29
11	0,8	1,52	1,84
12	0,7	1,12	1,17
13	0,7	1,16	1,18

Table 5.8: Comparison between the values of ASHRAE 62.1, experimental measurements and numerical simulations

The fig. C.1 shows the comparison between the different schemes using the vertical grille. We can see that in the scheme 3 the local air age in the room presents a stratified

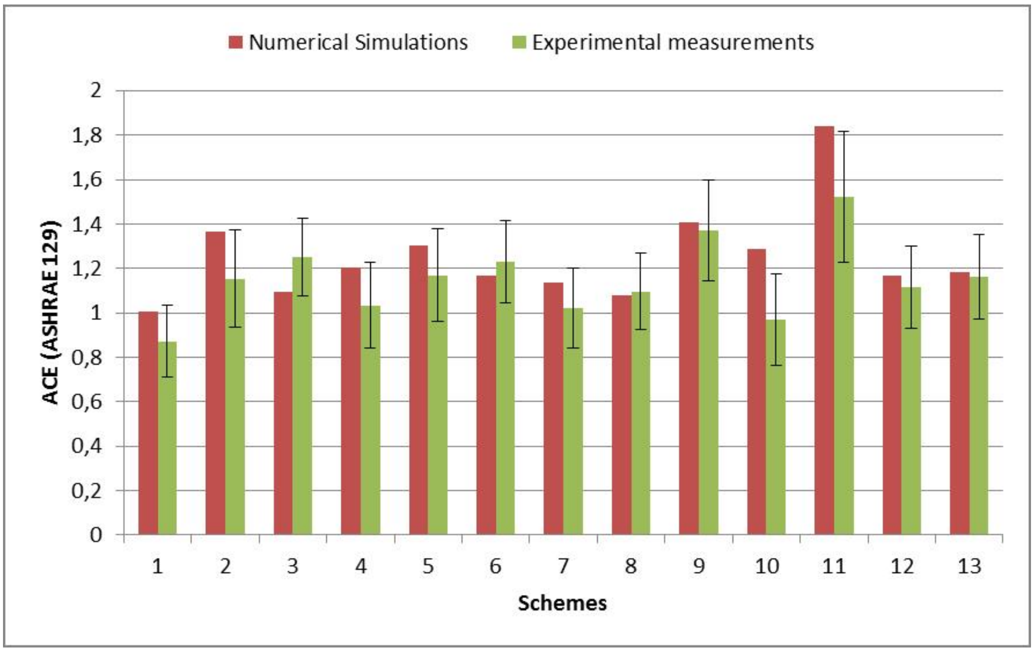

Figure 5.18: ACE (ASHRAE 129 method) experimental results vs numerical simulation

distribution. This is advantageous because it allows the breathing zone to have a younger air that the extraction zone. In the schemes 8 and 10, this situation don't happen, because the supply air is warmer than the air of the room. We can see in the scheme 8 that the the local air age of air is similar in the majority of the breathing zone and the extraction zone. This situation is more evident in scheme 10, because the temperature difference is greater. Despite this situation these strategies provide a better distribution of the air that the strategies that use the horizontal grille.

The fig. C.2 shows the comparison between the different schemes using the horizontal grille. We can see that in scheme 6 and 7 there is plenty of fresh air that exits the extraction grille without crossing the breathing zone. This reflects the existence of a short circuit between the supply grille and the extraction grille. This situation is worse in the scheme 7 because the temperature difference between the supply air and the room is 18°C whereas in scheme 6 is only 3°C. This is also reflected in the ACE value wich is 1,17 and 1,14 for the scheme 6 and 7 respectively. We can also see that the ACE value of the scheme 2 is higher than the scheme 1 (1,36 and 1 respectively), mainly due to the difference in flow velocity (0,45 m/s and 1,6 m/s respectively).

The fig. C.3 shows the comparison between the different schemes using the central diffuser. In the scheme 4 we can see that in the breathing zone in front of the person the local age of air is very younger, which improve the conditions of air quality. In the schemes 9 and 11 the conditions are not very good, exists the occurrence of short circuit of the supply air.

The fig. C.4 shows the comparison between the different schemes using the displacement diffuser. We can see in the scheme 5 that the local age of air in the room presents a stratified distribution, similarly to what we have seen in the scheme 3. The schemes 12 and 13 presents a poor distribution of the fresh air , this situation is worse in scheme 13, where exists short circuit between the supply diffuser and the extraction grille.

The schemes that have better behaviour with respect to the distribution of supply air are: 2, 3, 4, 5, 8 and 9. The other schemes are to be avoided. The table 5.9 gives us a overview of the schemes performance.

5.4 Results discussion

We can conclude that the value of the CRE is more dependent of the position of the contamination source (the person in this case), that the strategy used for the diffusion of air. All the strategies have difference between then smaller than 10%.

We can see that in the schemes with the supply air colder than the room temperature, that the value of the ACE is greater in schemes were the velocity of the supply air in smaller. The scheme 1 that have a velocity of the supply air of 1,6 m/s, the value of ACE is 0,87 while the scheme 2 that have a velocity of 0,45 m/s, the value of the indicator is 1,15. The schemes that promote the stratification of the air distribution, like scheme 3 e 5 have good values for the ACE.

In the schemes using the horizontal grille with the supply air warmer than the room temperature (scheme 6 and 7), the value of the ACE is larger in the scheme that the difference of temperature between the supply air and the room is lower.

In the schemes using the vertical grille with the supply air warmer than the room temperature (scheme 8 and 10), the value of the ACE is larger in the scheme that the difference of temperature between the supply air and the room is lower.

In the schemes using the central diffuser and the displacement diffuser with the supply air warmer than the room temperature (scheme 9, 11, 12 and 13), the value of the ACE is larger in the schemes that the difference of temperature between the supply air and the room is bigger. This can be explained because in this schemes there are a great quantity of supply air that short circuit the room in direction of the extraction grille, making the mean age of air in the extraction quite younger than the air in the breathing zone.

We can conclude that the schemes to be adopted, taking into account the ACE value, should be the schemes that operate with low speeds of supply air and small differences in temperature between supply air and the room. Should also be avoided all strategies that lead to the existence of short circuit situations.

The fig. 5.19 shows that in this study found no correlation between the value of ACE and the value of CRE.

Scheme	CRE	CO_2 concentration distribution	ACE	Fresh air distribution
1	0	0	0	0
2	0	+	+	+
3	0	++	+	++
4	0	+	+	++
5	0	++	+	++
6	0	- -	+	- -
7	0	- -	+	- -
8	0	-	+	-
9	0	+	++	-
10	0	- -	+	- -
11	0	+	++	- -
12	0	-	+	- -
13	0	- -	+	- -

Table 5.9: Overview of the schemes performance. ++ excellent +good 0 medium - poor - - very poor

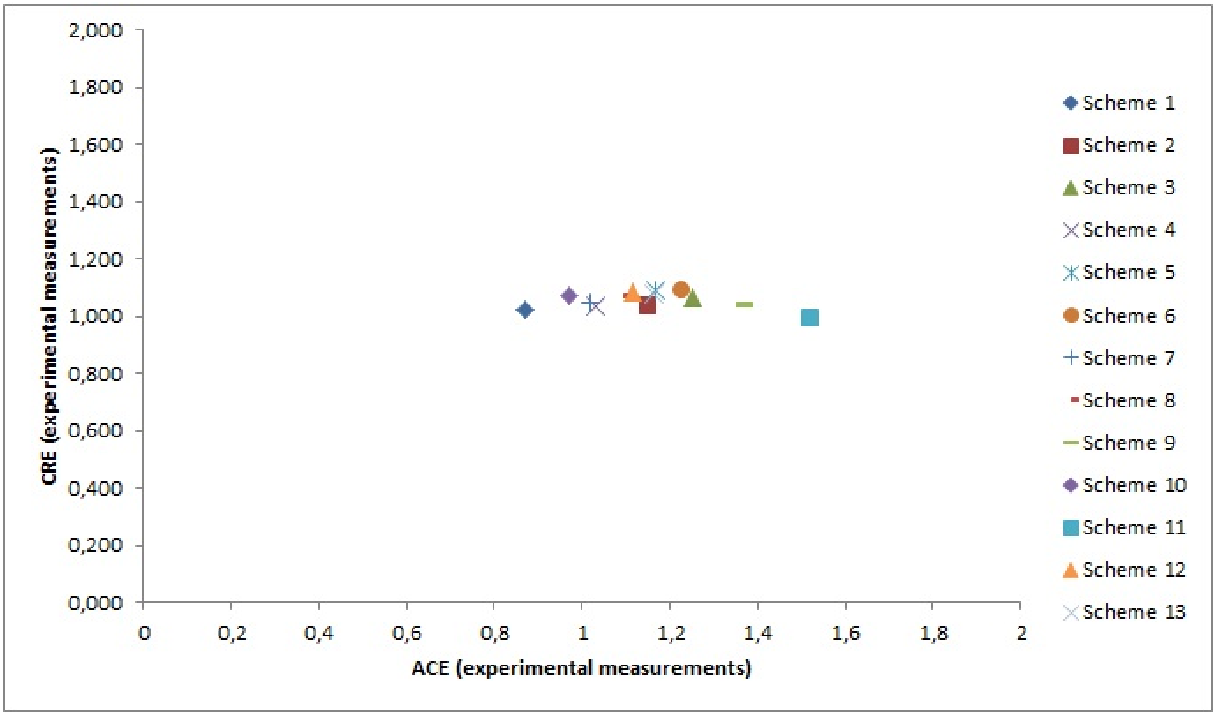

Figure 5.19: The correlation between air change efficiency and contaminant removal effectiveness

Chapter 6

Conclusions

In the present work, after the different experiments and simulations have been conducted, it the was possible to obtain the values of the ventilation efficiency for the different air diffusion strategies. The results of the work are leading to different conclusions and recommendations. The conclusions can be distinguished in the points listed under:

- The numerical simulations were validated by the experimental measurements, making the *Microflo* module of the *IES-VE* software suitable to predict the behaviour of the air stream in a occupied room for different air diffusion strategies.

- In the tests the location of the measurement points should be carefully chosen so they don't constrain the results.

- The different measurement sessions in the test chamber, revealed that, the ACE indicator is more appropriate for quantifying the quality of the air diffusion.

- The value of the CRE is more dependent of the position of the contamination source, that the strategy used for the air diffusion of air.

- We found no correlation between the numerical value of ACE and CRE. On these tests, those with higher levels of ACE does not necessarily correspond to those with higher values of CRE. The calculated values are not grouped by the various types of the air diffused used.

- The solutions to be adopted, to maximize the ventilation efficiency, should be the schemes that operate with low speeds of supply air and small differences in temperature between supply air and the room.

Nevertheless, some recommendations to further research this subject are presented now:

- Conducting tests for calculation the ventilation efficiency with the source of contamination in other locations of the room.

- Conducting test for calculation the ventilation efficiency in strategies that use the principle of task ventilation with the aim of verifying the possible increase of air quality in the breathing zone.

- Improve the relationship between the numerical value of ventilation efficiency and the quality of the air diffusion, to allow an effective reduction of energy consumption related to the ventilation system without diminishing the IAQ.

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Appendix A

ACE calculation (Nordtest method) Spreadsheet

A.1 ACE calculation (Nordtest method) Spreadsheet

Table A.1 shows the measured data, measured concentration and corresponding time in the two columns at the left. In fig. A.1 the last part of the usable decay curve is plotted against time. The exponential trend curve function in has been used to find the slope of the curve, -0.004.

First the area and the weighted area under the curve is calculated as shown in table A.1. Secondly the area and the weighted area under the tail are calculated by means of the calculated slope and the concentration of 442 ppm at time 118 minutes. Finally the total area under the curve is calculated by summing the 4th column and the total weighted area under the curve by summing the 5th column. The mean age of air and the nominal time constant can then be calculated and thus the air change efficiency.

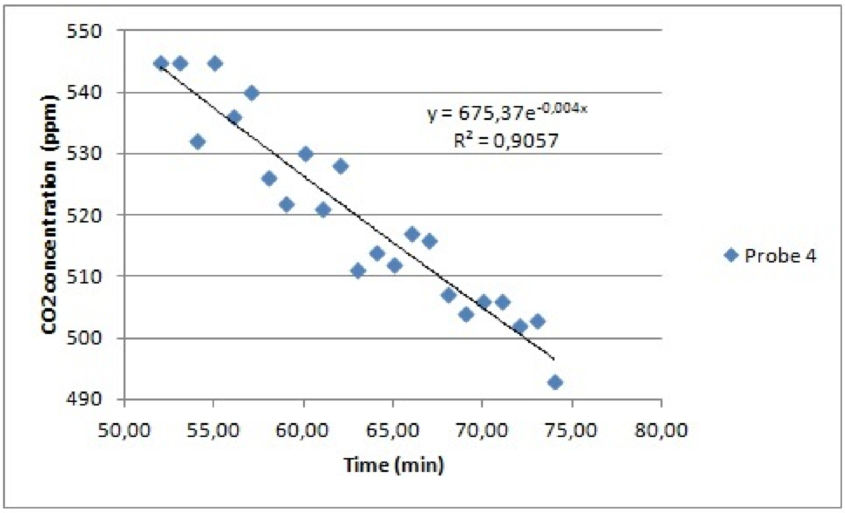

Figure A.1: Calculation of the slope of the step down test

Measured Values		Calculated Values			
Time (min)	Concentration (ppm)		Area under the curve	Weighted area under the curve	
0	773				
1	770		771,5	385,75	
2	764		767	1150,5	
3	764		764	1910	
4	774		769	2691,5	
5	757		765,5	3444,75	
6	760		758,5	4171,75	
7	747		735,5	4897,75	
8	735		741	5557,5	
9	738		736,5	6260,25	
10	722		730	6935	
11	724		723	7591,5	
12	704		714	8211	
13	697		700,5	8756,25	
14	692		694,5	9375,75	
15	698		695	10077,5	
16	686		692	10726	
17	678		682	11253	
18	670		674	11725	
19	674		671	13084,5	
20	662		668	13694	
21	656		659	14168,5	
...	...		...	...	
112	451		454,5	50676,75	
113	451		451	50737,5	
114	444		447,5	50791,25	
115	452		448	51296	
116	450		451	52090,5	
117	450		450	52425	
118	442		446	52405	
		Tail	1,768		Slope
		Weighted tail		650,624	-0,004
		Sum Area	65628,3		
		Sum Weighted area under the curve		3504740,4	
		Mean age of air	53,40		
		Nominal time constant	84,90		
		Air Change Efficiency	**0,79**		

Table A.1: Measured data and ACE calculation - scheme 2

A.2 ACE calculation (ASHRAE 129 method) Spreadsheet

Measured Values					Calculated Values		
Time	Concentration (ppm)				$\frac{(C_{i,n}+C_{i,n+1})}{2}(tn+1-t_n)$		
(min)	P1	P2	P3	P4	P1	P2	P3
0	762	773	832	773	758,5	770,5	831,5
1	755	768	831	770	749,5	766,5	825
2	744	765	819	764	740	757	810,5
3	736	749	802	764	731	747,5	798
4	726	746	794	774	721,5	740	791
5	717	734	788	757	713	729,5	782,5
6	709	725	777	760	707,5	721,5	772
7	706	718	767	747	702,5	714,5	763,5
...	...	...	...	...	...	...	...
113	432	433	439	451	431	434,5	439,5
114	431	437	438	444	431,5	435	438,5
115	428	433	437	452	429,5	435	437,5
116	425	438	438	450	426,5	435,5	437,5
117	426	436	440	450	425,5	437	439
118	424	437	436	442	425	436,5	438
$\sum_{n=first}^{n=last-1}\frac{(C_{i,n}+C_{i,n+1})}{2}(tn+1-t_n)$					65529	63678	65940
$C_{i,avg}=\frac{\sum_{n=first}^{n=last-1}\frac{(C_{i,n}+C_{i,n+1})}{2}(tn+1-t_n)}{(t_{last}-t_{first})}$					529,907	539,644	558,813
$A_i=\frac{(t_{stop}-t_{start})C_{i,avg}}{C_{i(t_{start})}}$					82,059	82,378	79,255
A_{avg}					81,231		
τ_n					93,747		
E					**1,15**		

Table A.2: Measured data and ACE calculation (ASHRAE 129) - scheme 2

A.3 Overall Results

Scheme	1	2	3	4	5	6	7	8	9	10	11	12	13
Contaminant Removal Effectiveness													
CFD	0,97	1,00	1,01	0,97	1,00	1,03	0,96	1,01	0,97	0,98	0,96	1,05	0,98
Tests	1,02	1,04	1,07	1,04	1,09	1,09	1,05	1,07	1,04	1,07	0,99	1,09	1,08
Air Quality Index													
CFD P1	1,03	1,03	1,03	0,99	1,04	1,06	1,00	1,02	0,99	1,01	0,99	1,05	1,02
CFD P2	0,95	1,00	1,01	0,96	1,00	1,03	0,95	1,01	0,97	0,97	0,95	1,06	0,96
CFD P3	0,94	0,98	0,99	0,94	0,97	1,01	0,94	1,00	0,93	0,96	0,93	1,04	0,96
Tests P1	1,07	1,07	1,09	1,06	1,13	1,13	1,12	1,10	1,08	1,13	1,08	1,13	1,11
Tests P2	1,03	1,05	1,07	1,04	1,10	1,09	1,05	1,08	1,04	1,08	0,97	1,10	1,08
Tests P3	0,98	1,00	1,04	1,01	1,04	1,05	0,97	1,04	1,01	1,02	0,95	1,03	1,04
Air Change Efficiency (Nordtest method)													
CFD	0,50	0,67	0,52	0,56	0,61	0,57	0,6	0,52	0,74	0,66	0,99	0,53	0,59
Tests	0,48	0,80	0,79	0,78	0,83	0,78	0,80	0,82	0,77	0,79	0,65	0,82	0,81
Air Change Index (Nordtest method)													
CFD P1	0,51	0,69	0,56	0,62	0,66	0,58	0,57	0,53	0,70	0,64	0,92	0,57	0,59
CFD P2	0,50	0,68	0,55	0,60	0,65	0,58	0,57	0,54	0,70	0,64	0,92	0,59	0,59
CFD P3	0,50	0,68	0,54	0,59	0,64	0,59	0,50	0,55	0,71	0,65	0,93	0,59	0,59
Tests P1	0,49	0,77	0,80	0,76	0,79	0,81	0,82	0,83	0,76	0,81	0,63	0,81	0,80
Tests P2	0,48	0,77	0,80	0,76	0,81	0,80	0,81	0,82	0,75	0,81	0,64	0,81	0,80
Tests P3	0,46	0,75	0,78	0,74	0,81	0,79	0,80	0,80	0,74	0,79	0,63	0,79	0,79
Air Change Efficiency (ASHRAE 129)													
CFD	1,01	1,36	1,10	1,21	1,30	1,17	1,14	1,08	1,41	1,29	1,84	1,17	1,18
Tests	0,87	1,15	1,25	1,03	1,17	1,23	1,02	1,10	1,37	0,97	1,52	1,12	1,63

Table A.3: Overall Results of measurements and simulations

Appendix B

CRE CFD images

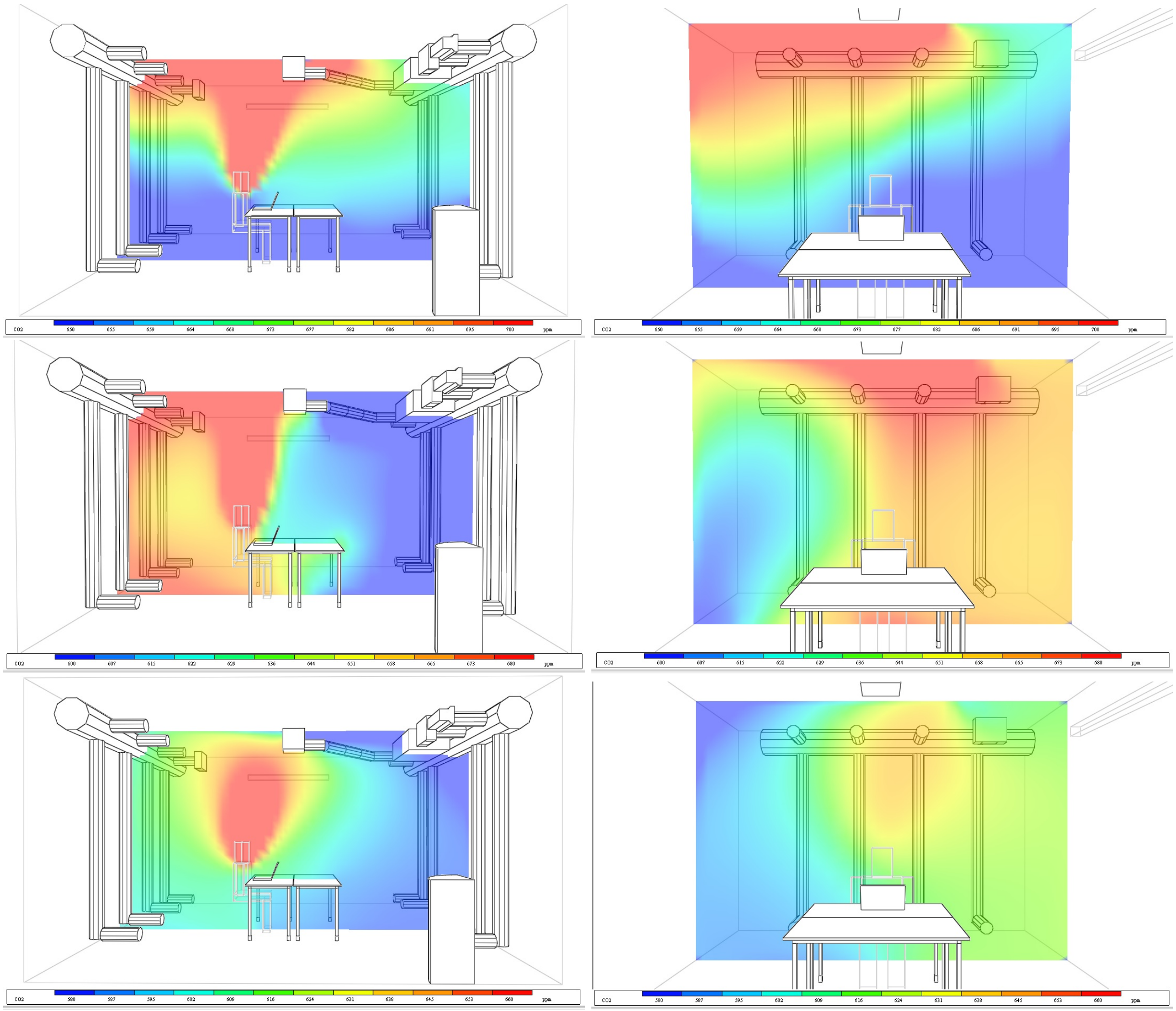

Figure B.1: Comparison between the different schemes using the vertical grille. From top to bottom: scheme 3, scheme 8 and scheme 10. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

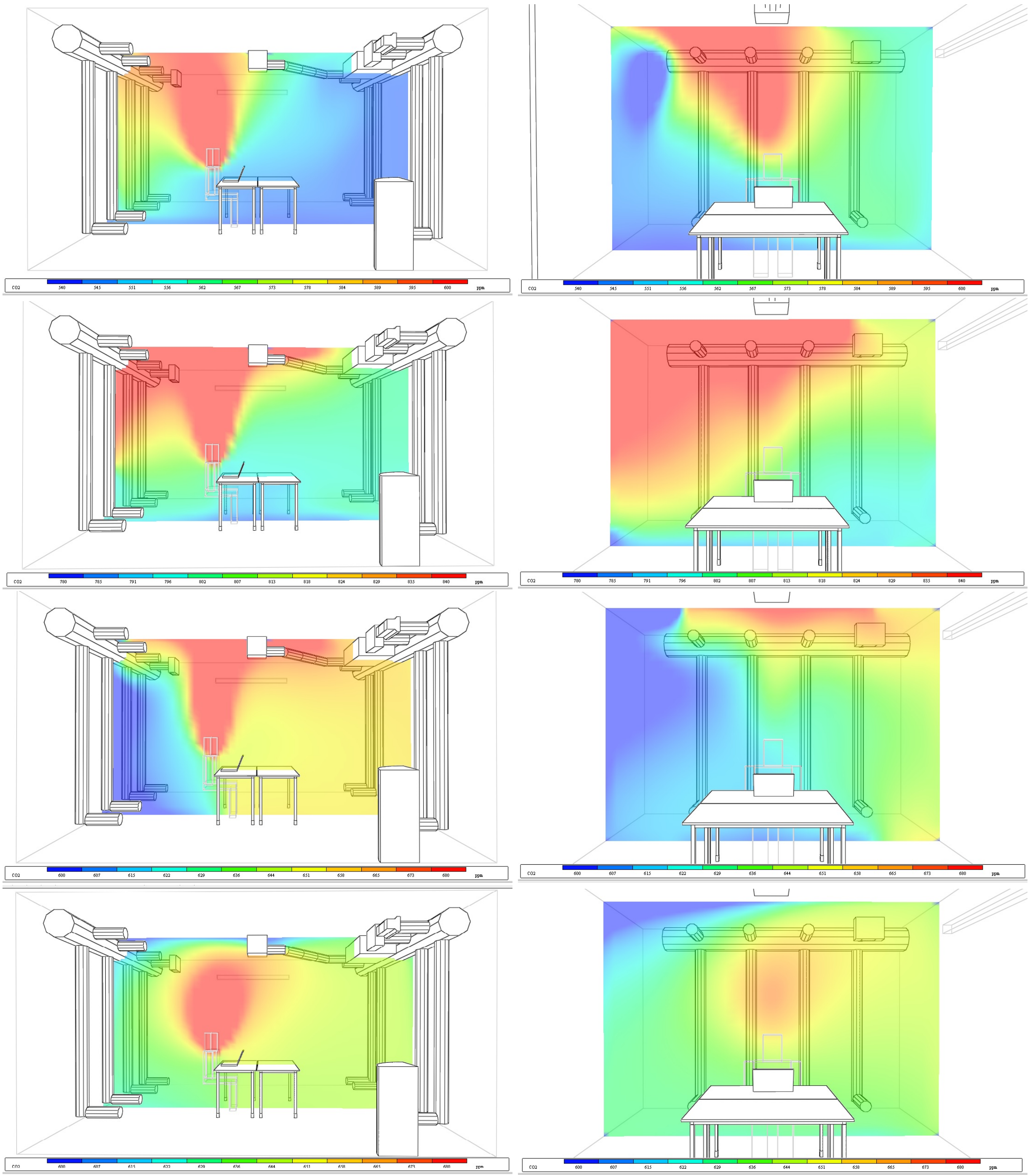

Figure B.2: Comparison between the different schemes using the horizontal grille. From top to bottom: scheme 1, scheme2, scheme 6 and scheme 7. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

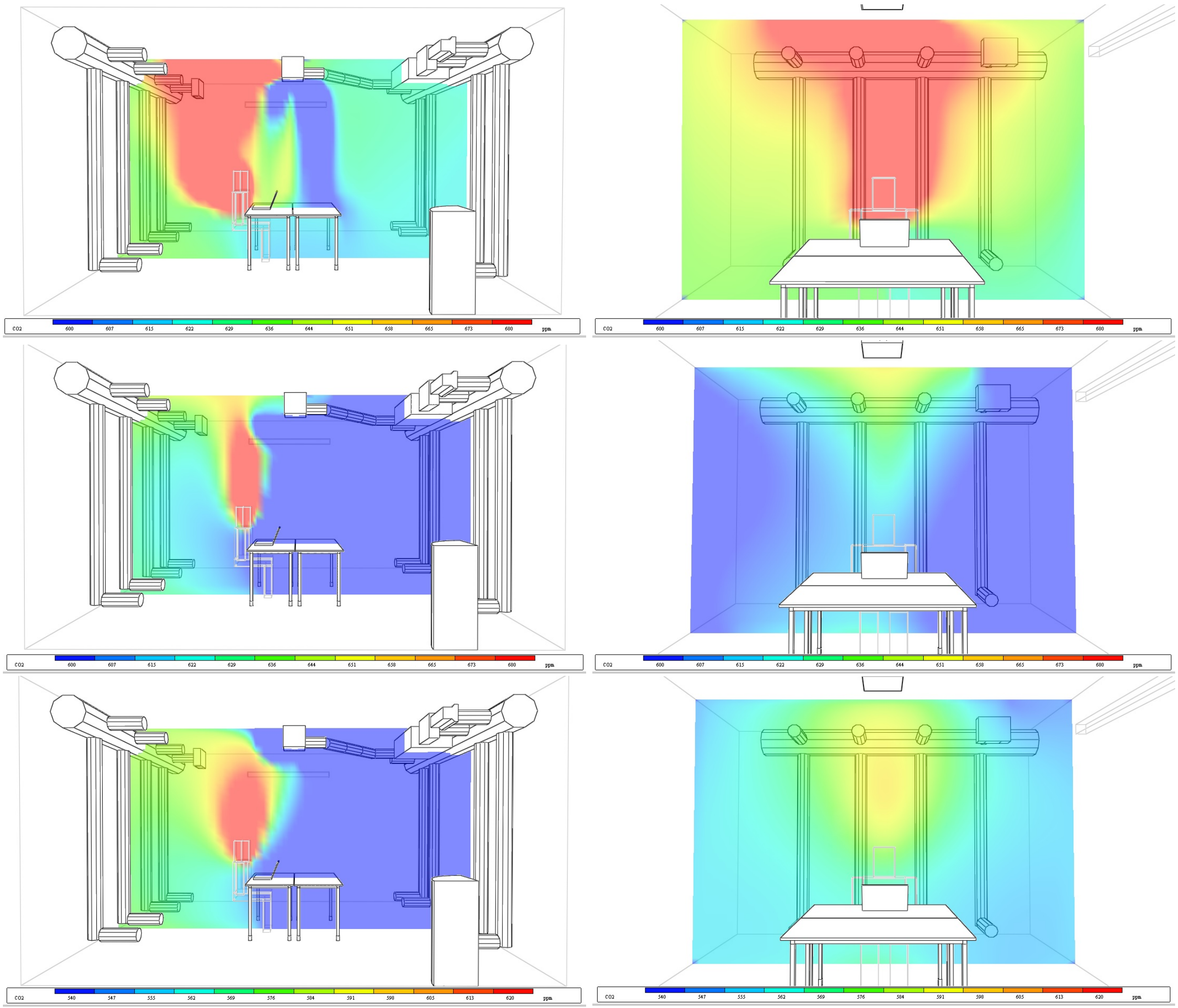

Figure B.3: Comparison between the different schemes using the central diffuser. From top to bottom: scheme 4, scheme 9 and scheme 11. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

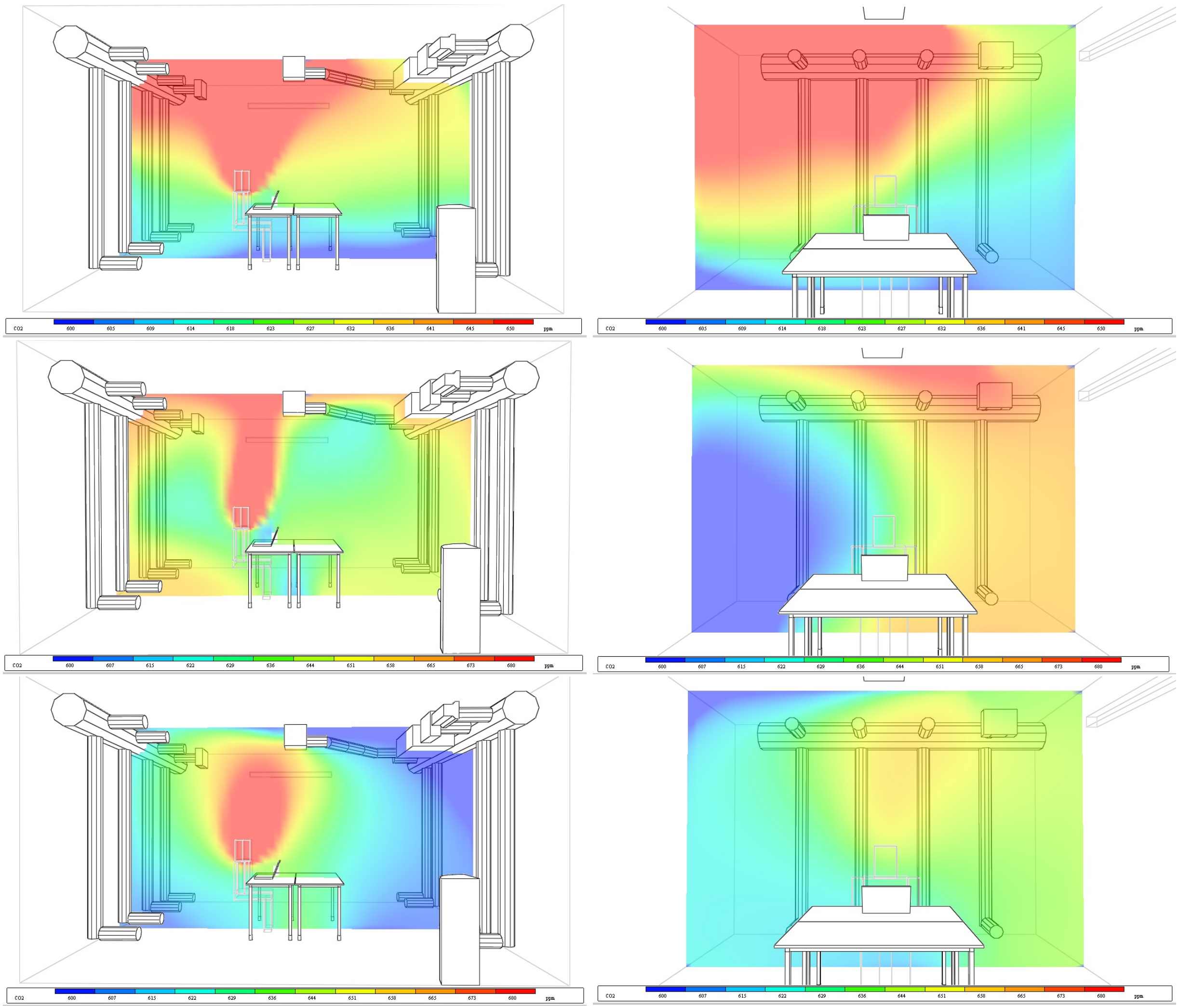

Figure B.4: Comparison between the different schemes using the displacement diffuser. From top to bottom: scheme 5, scheme 12 and scheme 13. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

Appendix C

ACE CFD images

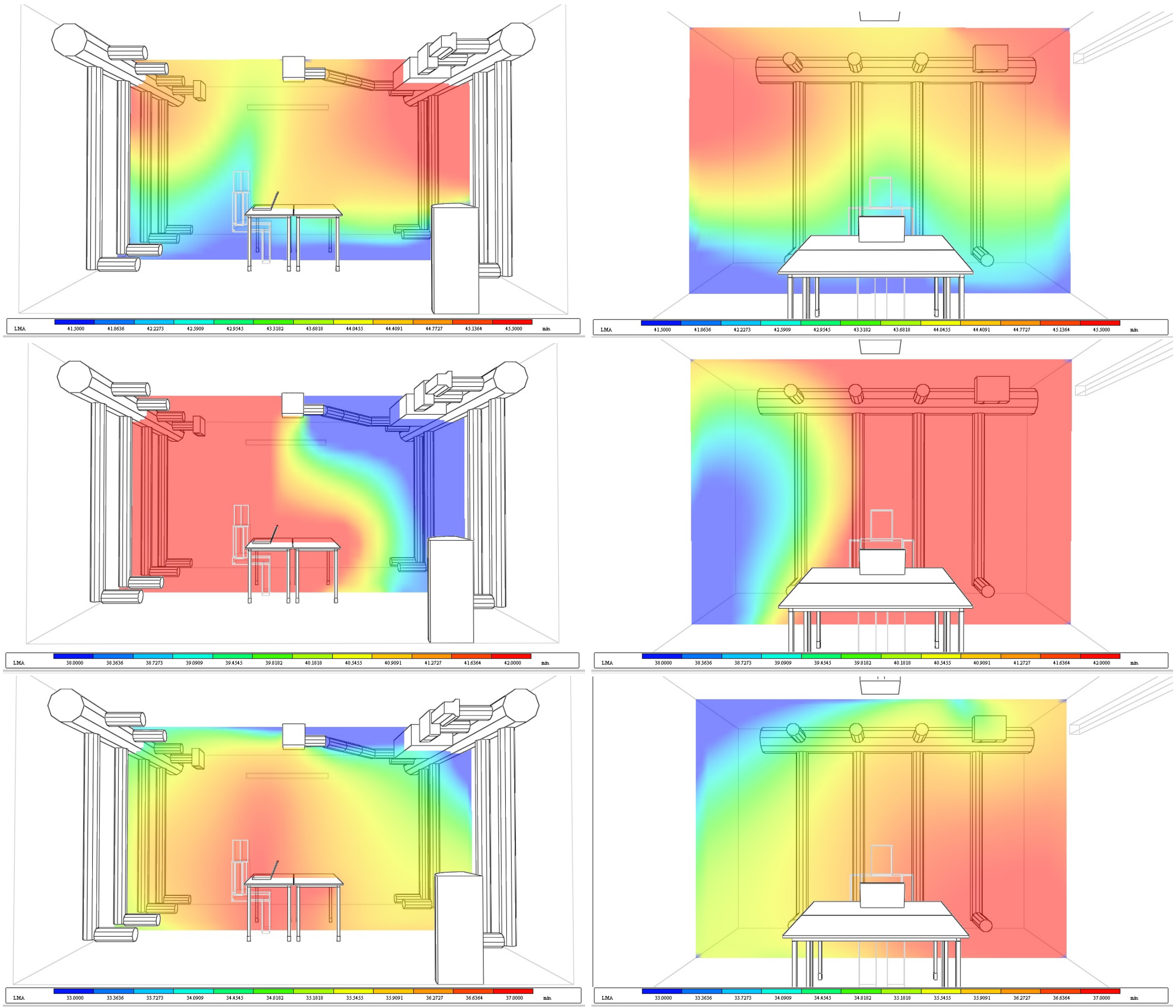

Figure C.1: Comparison between the different schemes using the vertical grille. From top to bottom: scheme 3, scheme 8 and scheme 10. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

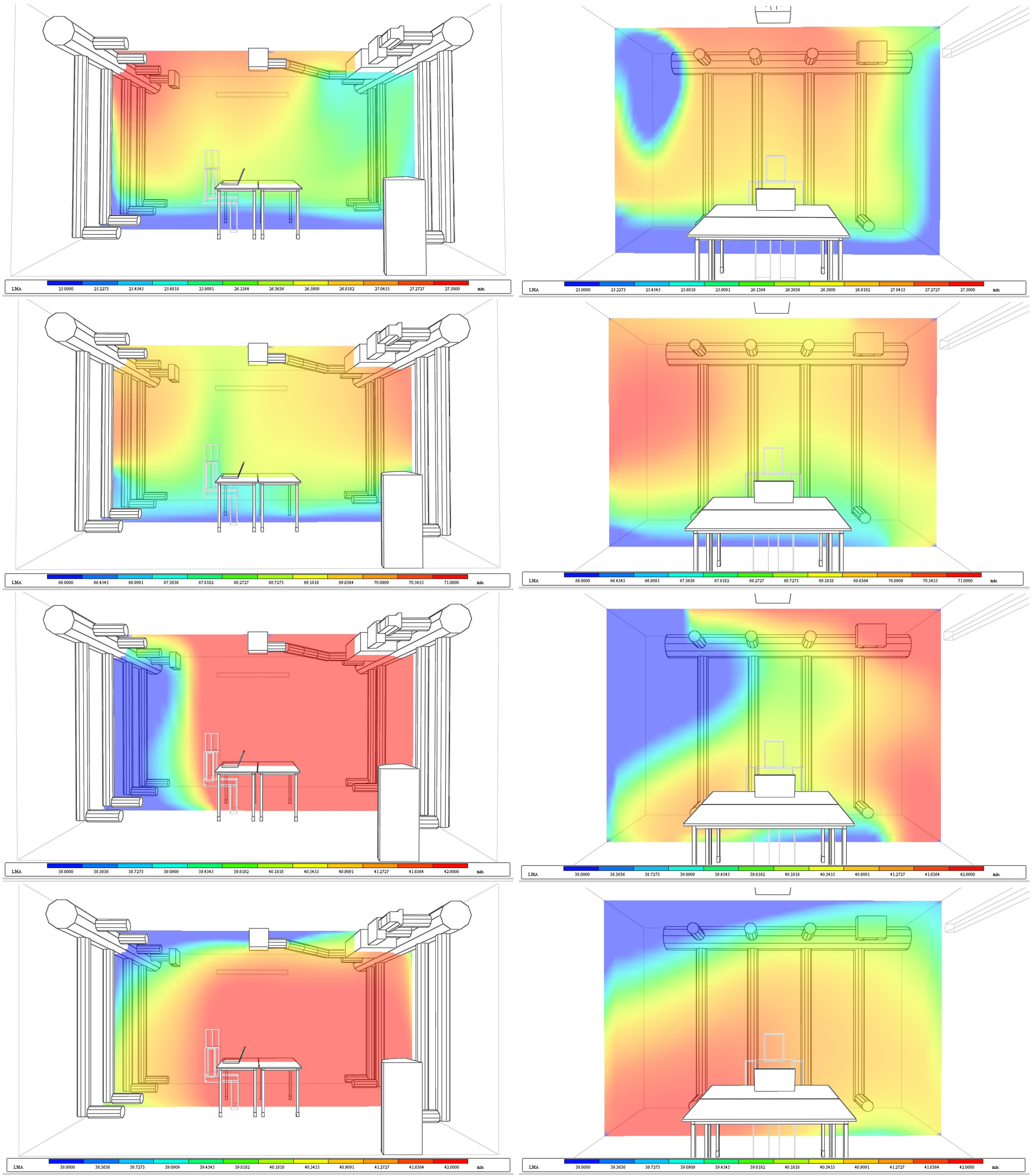

Figure C.2: Comparison between the different schemes using the horizontal grille. From top to bottom: scheme 1, scheme2, scheme 6 and scheme 7. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

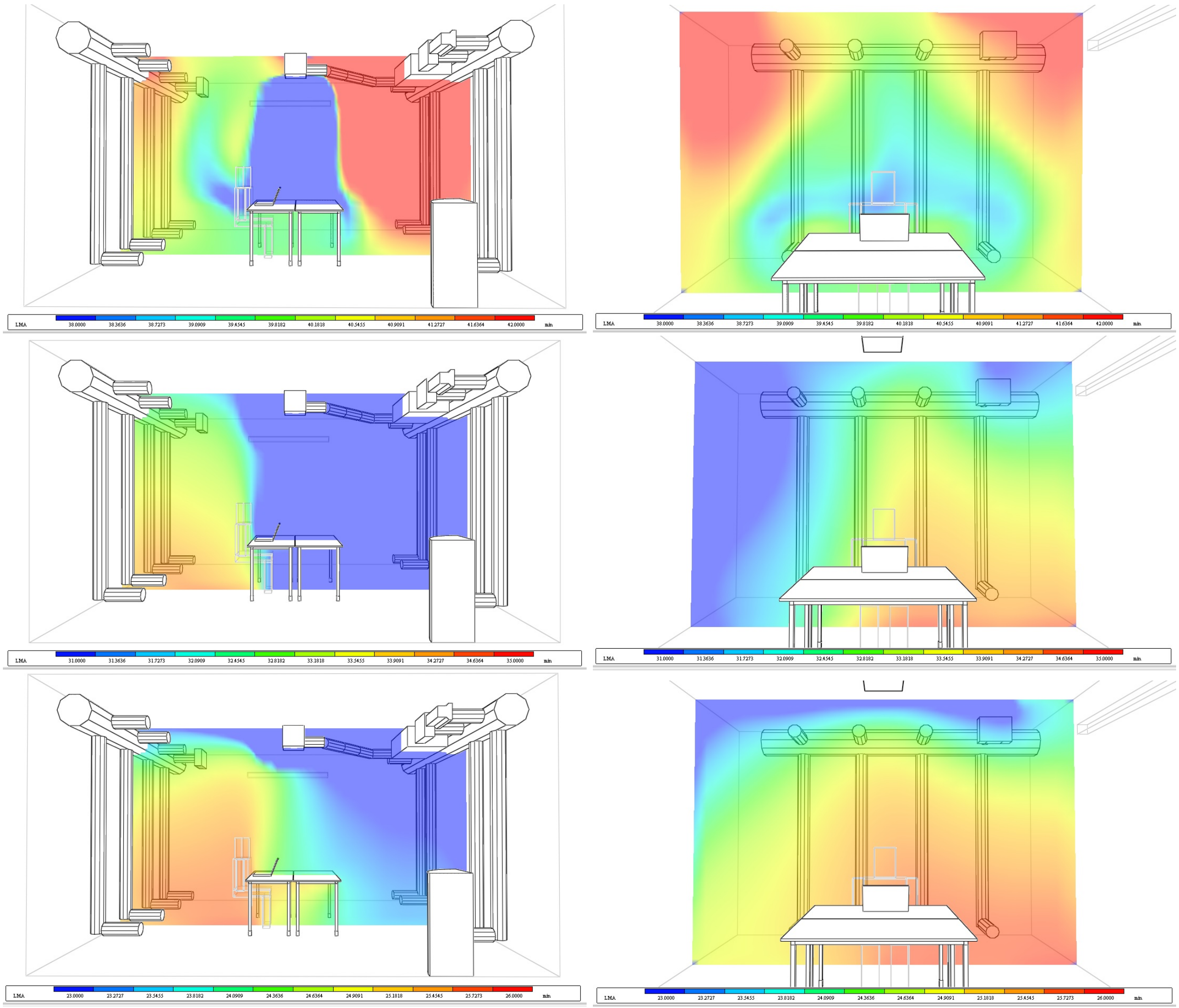

Figure C.3: Comparison between the different schemes using the central diffuser. From top to bottom: scheme 4, scheme 9 and scheme 11. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.

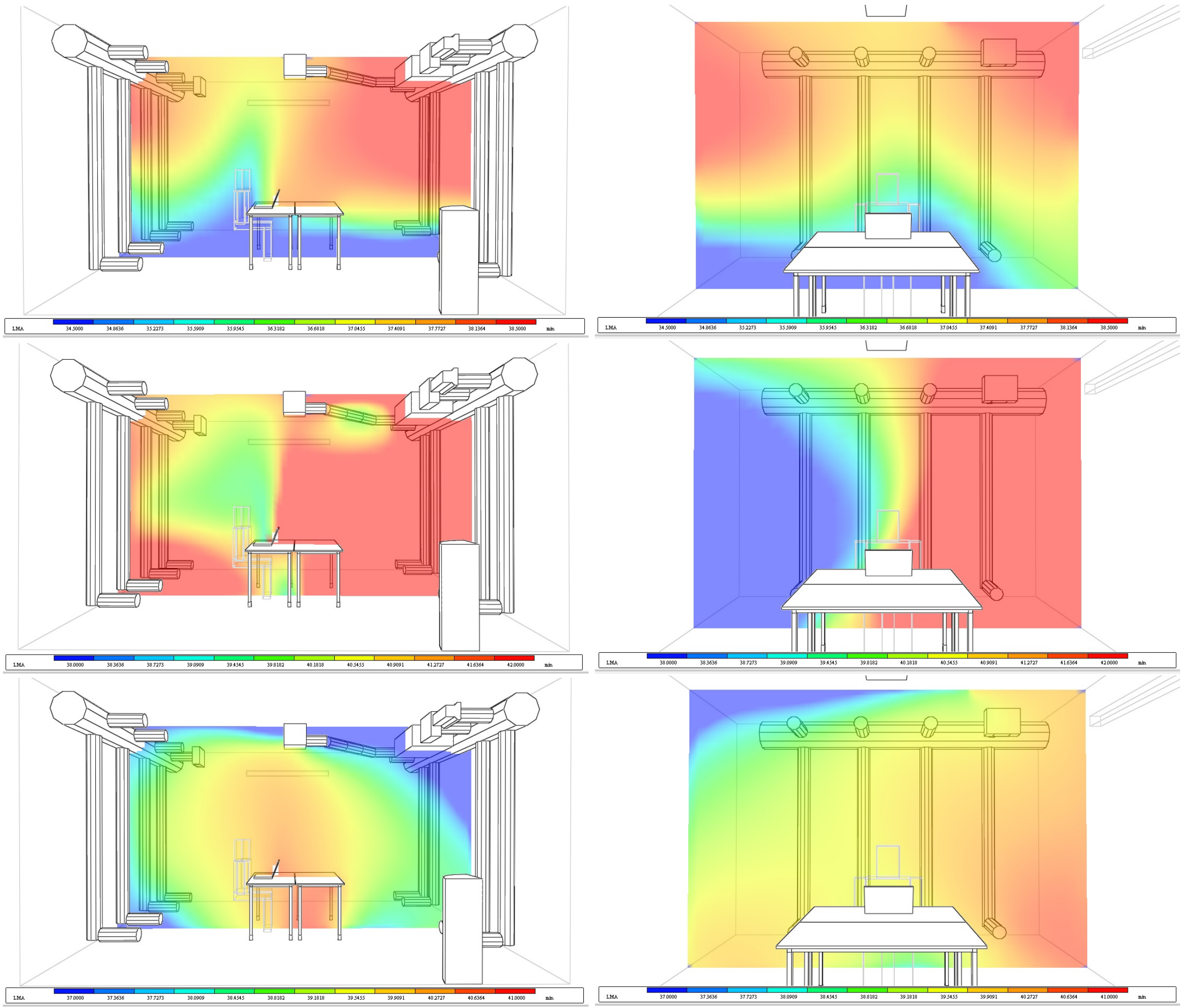

Figure C.4: Comparison between the different schemes using the displacement diffuser. From top to bottom: scheme 5, scheme 12 and scheme 13. The figures on the left are for the plane x=2m and the figures on the right are related to the plane y=3,63m.