Ventilation efficiency measurements

A comparison between three supply air methods

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Abstract

This study compares the two traditional ventilation methods; mixed ventilation and displacement ventilation, with a new type of diffuser which attempts to solve the problem with blockage of the diffuser for the displacement ventilation. This new diffuser called Airshower is a product developed by Airson engineering AB whom is also the company where all simulations and tests for this study were performed in a built office space. The study compares in term of ventilation efficiency, ventilation index, air exchange efficiency and mean age of air.

The Airshower diffuser did prove to be the most efficient ventilation solution in comparison to the other systems when measuring with constant heat loads but the worst one with variable loads. The uncertainty of the results is estimated to be of consequence since the built office space had some flaws such as leakage, issues with the adjustable floor and control system.

In ventilation index terms however, the mixed air system is the only system which yields any expected results, this fraction between room- and exhaust air once again lacks confidence due to the unsealed laboratory. The displacement system would do a lot better if the mean age of air in the exhaust air would be older.

The mean age of air does not differ much from system to system, it is in the interval of 20-30 minutes for all the systems with the exception for one point in the room for both of the displacement system with the probable cause of turbulence causing air currents reducing the efficiency of the system.

This study is not enough to conclude that any system is significantly better than any other but both the results and the conclusion of this study could change a lot for the displacement systems should the floor and walls be sealed.

Keywords: Ventilation efficiency, air exchange efficiency, local ventilation index, mean age of air, comparison, mixed ventilation, displacement ventilation

Preface

The authors of this investigation would like to extend their thanks to Airson Engineering AB for allowing free access to their laboratory, equipment and competence. Special thanks to Dan Kristensson CEO of Airson Engineering and Andreas Dahl, engineer at Airson Engineering, for their guidance throughout this study.

Lund University also deserves gratitude for providing the measuring equipment which this study is entirely dependent on.

Finally, we would like to thank Dennis Johansson for his great assistance during this entire process, his patience and pedagogic way of explaining was an important factor especially for the development of the distribution system and for the making of this written report. And Hans Bagge for his valuable input and critically review of this written report.

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Table of content

Air exchange efficiency
A measure of how efficiently air is changed in a room.

Homogeneous emission
The injection rate per unit volume is equal in all parts of a ventilated system with this strategy to inject tracer gas.

Internal gains / Heat loads
Objects inside a room releasing heat.

Local ventilation index
A measure of how efficiently pollutions are removed in a specific point or a smaller part of a room.

Mean age of air
A statistical concept based on the age distribution of different air molecules. The age is counted from when the molecules enter the building. The air in a specific point is composed of molecules with different ages.

Pumped sampling
Air is pumped through sorbent tubes and is one of the most versatile sampling methods. The pump is located in the read of the sorbent tube so that the air is drawn onto the front (sampling/grooved) end. (Ecaservice)

Temperature efficiency
A relationship of the conditions of air in terms of temperature in the exhaust, supply and room air.

Ventilation air flow
Volume flow of "clean" air being supplied to a room to dilute and transport pollutions away from the room.

Ventilation efficiency
An average measure of how efficiently pollutions are removed from a whole room.

Ventilation process
A process in which pollutants are diluted with air and transferred away from a ventilated space.

Symbols, definitions and units

Symbol	Definition	Unit
τ_n	Nominal time constant	Minutes
τ_p	Local mean age of air	Minutes
$\Delta\tau$	Measuring interval	Minutes
τ_i	Time since start	Minutes
C_i	Concentration at given time	g/cm^3
τ_M	Time to equilibrium	Minutes
C_M	Concentration at equilibrium	g/cm^3
λ_{exp}	End-curve gradient	-
ε_a	Air exchange efficiency	%
$\bar{\tau}_r$	Air exchange time	Minutes
$\langle\varepsilon\rangle^C$	Ventilation efficiency	%
ε_p	Local ventilation index	Decimal
C_f	Concentration of tracer gas in exhaust diffuser	g/cm^3
C_p	Concentration of tracer gas in point measured	g/cm^3
τ_f^C	Mean age of room air	Minutes

1 Introduction

1.1 Background and problem motivation

To establish an indoor environment in office buildings suitable for human habitants some kind of system is required to control the air. This system is commonly a mixed air ventilation system in today's offices. The downside off using a mixed air distribution as a way to deliver fresh air is that this fresh air mixes with the old and a mixture is ventilated out as exhaust air. To some extent the exhaust air then contains energy such as heat or chill that was meant to heat or cool the room to a desired temperature. To reach this desired temperature a mixed air system then needs to cool or heat more to maintain the temperature preferred. Another flaw of using a mixed air system is that pollutants such as carbon dioxide and odors will be mixed with the old air and remain in the room.

A method for solving this problem is to use a displacement ventilation system. This system is essentially trying to push the old air out simultaneously as the new air enters the space. The reason the system might work is due to the way hot and cool air acts. As the air in an office heats up due to habitants and equipment, it rises and as new cool air is delivered, the old warm air is pushed out through an exhaust vent at the ceiling. A common problem with displacement ventilation systems is the way the diffuser is placed. Usually it is integrated into the wall at floor level which then might be blocked by furniture or machinery. Also there will be an area close to the diffuser with air speeds causing draught. This study investigates a solution to this problem using a newly developed displacement diffuser from the company Airson Engineering AB, located in Ängelholm, Sweden.

1.2 Aim

The aim with this study was to analyze and verify whether the displacement principle with vertical wall placed supply air diffusers near the ceiling would maintain a high level of indoor climate in office environment regarding ventilation efficiency. This principle was also compared against two more traditional principles to determine which principle was preferable. The principles compared with were the traditional mixed air ventilation method and a displacement ventilation method with the diffuser on floor level.

1.3 Scope

This study will be based on measured values in a mockup cell office to represent a real case with distribution of tracer gas to act like a pollution. Focus will be on tracer gas concentrations to calculate mean age of air, ventilation index and ventilation efficiency. Temperatures will also be studied for comparison while computer simulations are a matter of future research. To create a real case, heat loads for lighting, computers and external sun will be used together with time schedules to measure one case with constant heat loads and one case with variable heat loads.

Three different types of ventilation techniques will be studied in this mockup office space for air exchange efficiency;

- Displacement principle with vertically wall placed supply air diffuser near the ceiling
- Displacement principle with traditional supply air diffuser near floor level
- Mixed air principle with ceiling mounted diffuser

2 Theory and litterature review

Air quality itself expressed in pollutant concentrations can depend on different things and in this study tracer gas was used as a “pollution” and focus was on (Blomqvist et. al, 1983).

- Pollutant distribution
- Ventilation flow, q_v

The important factors considered for distribution of pollutants (gas) and air was (Blomqvist et. al, 1983).

- Characteristics
- Supply air temperature
- Placement of diffusers
- Placement of heat sources
- Qualities of diffusers

2.1 Pollution sources

Indoor pollutions consist of fragmentation from different sources like furniture, building materials, humans, chemical processes and tobacco smoke. Since humans are the central part of the context it was our own personal perceptions and demands that decides the air quality. In addition to pollutions humans also produce heat as well as affecting the way air moves. Air streams close to the body have a high air flow, around 150 m^3/h above the head (0.7 m) according to *Blomqvist et. al, (1983)*. Since air streams start at floor level they have a large impact regarding pollutant levels for inhaled air. This attribute displacement ventilation makes use of (Blomqvist et. al, 1983).

2.2 Ventilation principles

There are in general two main types of systems to supply outdoor air to a space, natural ventilation and fan controlled ventilation where the last will be studied in this report. Starting with air flow and air distribution there are mainly three types of ventilation principles that can be utilized with fan controlled systems. Displacement principle, mixed air principle and piston flow where piston flow is the most expensive due to high air flow. Short circuit is a fourth principle where supply air flows directly to the exhaust but this is not acceptable and will not be studied. The main difference between mixed air and displacement is that with mixed air it is the ventilation air that controls the air currents in the room while for displacement it is the heat sources (Blomqvist et. al, 1983).

2.2.1 Displacement ventilation

Displacement principle is a cheaper method to create piston-like flow without a high air flow by utilizing the features of convection flows generated by heat sources with stratified currents. Stratified currents mean that cold supply air which is heavier than warm room air enters with low velocity and distributes across the whole floor. By thermal forces the air rises (one-way flow) when it comes in contact with warm bodies, bringing pollutions up towards the ceiling where the air and pollutions are being extracted.

Convection flows have two factors to consider;

- Air flow increases with the distance from the heat source
- Velocity decreases with the distance from heat sources

In the end, by using displacement ventilation this results in two zones in the room, one lower zone with piston flow and one higher with mixed air. This is explained with the convection flows that always have an up-going air current. When the height increases (the distance from heat source) the flow also increases resulting in higher drawn surrounded air for the convection flows. The lower zone works as piston flow up to the height where the convection flows current, q_p, is equal to the ventilation flow, q_v.

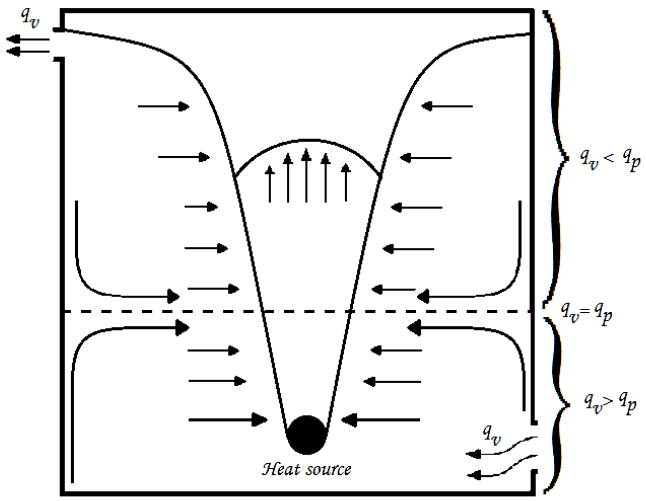

Figure 1 – Explanation of how air flow increases with distance from a heat source and how velocity is decrease with distance from heat source.

To calculate the air flow one need to know the power output from the heat sources. Since the air flow is proportional against the power output raised to a third. The air flow is relatively low depending on the output from the heat source. It is explained by *Blomqvist et. al, (1983)* that around 17 l/s (61.2 m^3/h) and person is enough to raise the arbitration level above inhalation height creating good air quality. It is also explained that this is not always necessary to maintain good air quality since convection flows around a person gather “clean” air from the lower zone which displaces the arbitration level upwards.

The human benefits of this principle are that transportation of pollutions from the higher zone to the lower is prevented and the air currents close to a person lowers the concentration of pollutions in the inhalation air. A problem with this principle might be sound traveling through the supply air diffusers due to its larger opening area for achieving low velocities. Another problem is that warm air cannot be supplied (Warfvinge & Dahlblom, 2010; Blomqvist et. al, 1983).

2.2.2 Mixed air

The main reason with mixed air is to obtain uniformity across the whole room with equal temperatures and concentrations of pollutions. This results in same conditions in the exhaust air. With this principle supply air diffuser are placed outside the “working space” to reduce draft while air is supplied with relative high speed. They can be located towards the rear, front or ceiling of a room or below the window. Extract air diffusers are often placed in ceiling level but can be placed anywhere as long as short circuit is prevented.

Mixed air ventilation can be achieved with two different methods; supply of air with jet beams (high-speed technology) or supply of chilled air at ceiling level with rather high speed (thermally controlled air flow). In this report the later method is used with chilled air which descends to the floor creating only one zone.

Mixed air is based on the properties of turbulent jets which follows as;

- Flow increases with the distance from the supply diffuser
- Width and thickness of the beam increases with the distance from the supply diffuser
- Speed decreases with the distance from the supply diffuser
- Speed fluctuations (turbulence) is created

It is mostly the first property who creates air currents which then creates mixed air (Warfvinge & Dahlblom, 2010; Blomqvist et. al, 1983).

2.2.3 Piston flow

For special applications with higher demands on air quality a more expensive and exclusive principle can be used. Here the supply air moves like a piston through the room, pushing the room air out. The room could be seen as a continuation of the supply air duct. To achieve this together with thermal air movements from heat sources the supply air flow and speed has to be high, around 40 - 50 cm/s (Warfvinge & Dahlblom, 2010).

Table 1 shows the intervals for air exchange efficiency for each of the ventilation principles. Air exchange efficiency is a measure of how efficiently air is changed in a room.

Table 1 – Air exchange efficiency for different ventilation principles

Ventilation principle	Air quality	Air exchange efficiency
Piston flow	Supply ratio	100%
Displacement	Supply ratio lower part of room	>50%
Mixed air	Extract ratio	~50%
Short circuit	Worse	<50%

3 Method

This part of the study presents two methods for measuring and calculating how efficiently a specific supply air system exchanges air and transports pollutants away in an office room for one person. To evaluate ventilation efficiency both calculations and measurements can be used. The complexity of air however means that computational fluid dynamics (CFD) must be used in the case of calculations. Real measurements on the other hand has less uncertainties but require a laboratory in order to ensure repeatability. Actual measurements in a laboratory were preferred since mere calculation of how the air would act in such a scenario might be imprecise and unrealistic. It was considered to compare the measured values with simulated ones using a program called FloVENT, however this was concluded as future studies.

3.1 Diminishing concentration method

This method delivers three different types of results which is of interest in this study:

- Mean age of room air
- Air exchange efficiency
- Ventilation efficiency

Mean age of room air is a measure of the average age of all of the air in the room. Air exchange efficiency is a measure of how efficiently the air inside the room is exchanged with new air. Ventilation efficiency is a measure of how efficiently a pollutant is transferred away from the room.

To set up such a measurement, there needs to be an even concentration of a tracer gas throughout the entire room as an initial condition. The investigated system is then switched on and the concentration of tracer gas is measured in the exhaust air diffuser over time. This information is than plotted in a graph as shown in Figure 2.

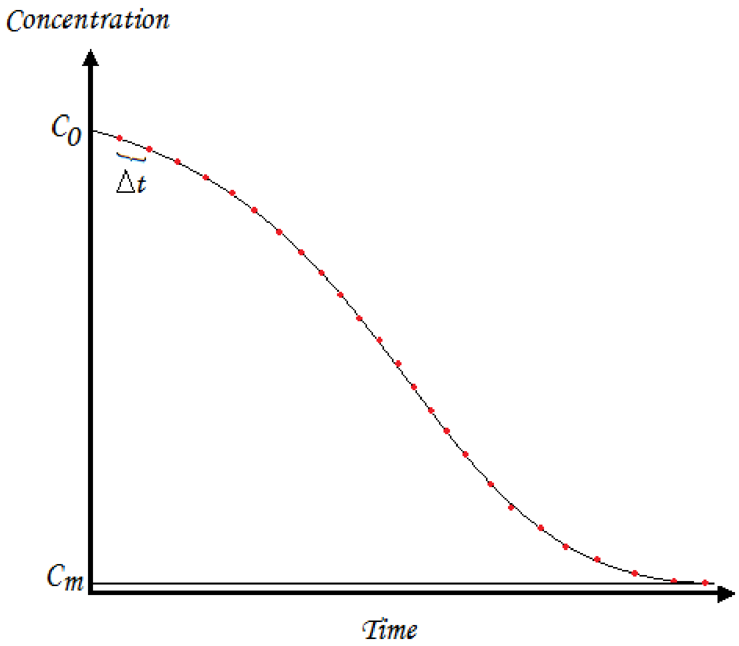

Figure 2 - Concentration of tracer gas in exhaust diffuser over time

C_0 is the initial concentration in the exhaust air diffuser, C_m is the concentration of tracer gas after the system has reached its equilibrium and $\Delta\tau$ is the time between each measurement.

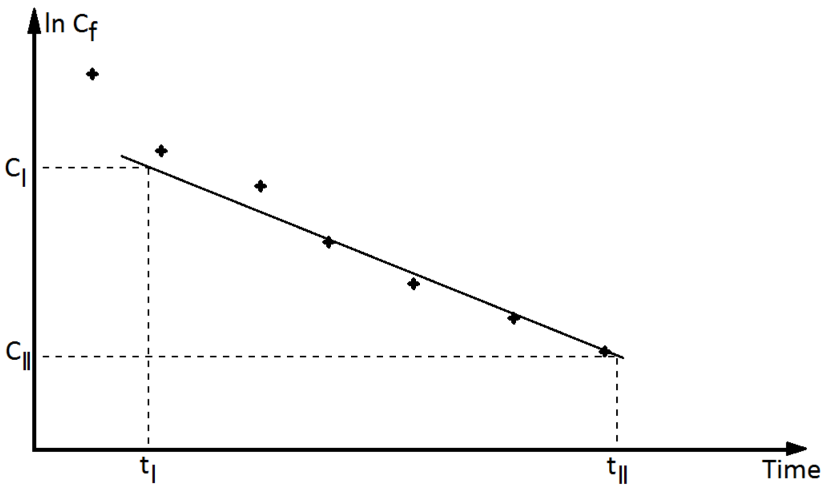

Figure 3 – Lin-Log diagram of exhaust air diffuser measured tracer gas concentration

Figure 3 shows an example of a Lin-Log diagram of the end part of the concentration graph (Figure 2). This end-curve gradient is useful in the calculations following the formula.

The air exchange efficiency can be calculated using the following equations.

$$Mean\ age\ of\ room\ air = \frac{Element_{Calculated} + Element_{remaining}}{Area_{calculated} + Area_{remaining}} \tag{1}$$

$$Element_{Calculated} = \Delta\tau \cdot \sum_{i=1}^{M-1} \tau_i \cdot C_i + \frac{1}{8} \cdot \Delta\tau^2 \cdot C_0 + \frac{1}{2} \cdot \Delta\tau \cdot \tau_M \cdot C_M \tag{2}$$

$$Element_{remaining} = \frac{C_M}{\lambda_{exp}} \cdot \left(\tau_M + \frac{1}{\lambda_{exp}}\right) \tag{3}$$

$$\lambda_{exp} = \frac{\ln C_{II} - \ln C_I}{\tau_I - \tau_{II}} \tag{4}$$

$$Area_{Calculated} = \Delta\tau \cdot \sum_{i=1}^{M-1} C_i + \frac{\Delta\tau}{2} \cdot (C_0 + C_M) \tag{5}$$

$$Area_{remaining} = \frac{C_M}{\lambda_{exp}} \tag{6}$$

$$Air\ exchange\ efficiency = \frac{\bar{\tau}_p}{2 \cdot \langle\bar{\tau}\rangle} \tag{7}$$

$$\langle\bar{\tau}\rangle = \frac{\sum \tau_i C_i \Delta\tau}{\sum C_i \Delta\tau} \tag{8}$$

$$\bar{\tau}_p = \frac{Area_{measured} + Area_{measured\ remaining}}{C_0} \tag{9}$$

$$Area_{measured} = \Delta\tau \cdot \sum_{i=1}^{M-1} C_i + \frac{\Delta\tau}{2} \cdot (C_0 + C_M) \tag{10}$$

$$Area_{measured\ remaining} = \frac{C_M}{\lambda_{exp}} \tag{11}$$

$$Ventilation\ efficiency = \frac{\bar{\tau}_p}{\langle\bar{\tau}\rangle} \tag{12}$$

3.2 Homogeneous emission method

The result this method is able to produce, that is of interest to this study, is as follows:

- Local mean age of air
- Local ventilation index
- Air exchange efficiency
- Ventilation efficiency

Local mean age of air is the average age of air (age is 0 when air is supplied to the room) in a point in the room, Local ventilation index is a measure of how efficiently a pollutant is transferred away from a specific point.

To be able to measure according to this standard it is essential to distribute gas evenly throughout the room. This means that each outlet of tracer gas should cover an equal volume of indoor air in comparison to any other outlet in the room. Not only must the distribution of outlet points be evenly distributed, the tracer gas flow must be even as well. To ensure that the distribution system actually does follow these requirements further analysis is discussed in the chapter 4.3 Gas distribution and gas measuring equipment.

The measuring points can be placed anywhere in the room but it is preferred to have points in areas of interest such as the occupancy zone or different air layers by height. One of the points is required to be placed in the exhaust air diffuser, otherwise it is impossible to investigate the room as a whole.

After setting up the investigated room the output information from the measuring equipment needs to be modified in order to draw any comparable conclusions. This can be calculated using the following formulas.

$$\boldsymbol{\tau_p} = \frac{\boldsymbol{C_p}}{\frac{\boldsymbol{E}}{\boldsymbol{V}}} \tag{11}$$

Where the local mean age of air (τ_p), is the age of the air since it entered the room through the supply air diffuser. The steady state concentration of tracer gas, C_p, E is the flow of tracer gas and V is the volume of the room.

$$\boldsymbol{\varepsilon_a} = \frac{\boldsymbol{\tau_n}}{\boldsymbol{\bar{\tau}_r}} \tag{12}$$

Where $\boldsymbol{\varepsilon_a}$ is the air exchange efficiency, τ_n is the nominal time constant and $\bar{\tau}_r$ is the air exchange time.

$$\tau_n = \frac{V}{q} \tag{13}$$

Where q is the supply air flow.

$$\bar{\tau}_r = 2 \cdot \tau_f^C \tag{14}$$

Where τ_f^C is the mean age of the room air which is assumed to be the same as the local mean age of air in the exhaust diffuser.

$$\langle\varepsilon\rangle^C = \frac{\tau_n}{\tau_f^C} \tag{15}$$

Where $\langle\varepsilon\rangle^C$ is the ventilation efficiency.

$$\varepsilon_p = \frac{C_f}{C_p} \tag{16}$$

Where ε_p is the local ventilation index, C_f is the concentration of tracer gas in the exhaust air diffuser, C_p is the concentration of tracer gas in the point measured.

There are a few standards applicable when using this method, one of these are the ISO 16000-8:2007. ISO 16000 explains indoor air while part 8 determines the local mean age of air in a building with the use of a single tracer gas and characterizing ventilation conditions. This standard is available for purchase if more information is desired, it was not used in this study.

Another standard is Nordtest, a freely available standard. This standard specifies how homogeneous tracer gas distribution should be achieved in order to determine local mean age of air in a room or building. The standard requires a constant flow of tracer gas distributed in a way so that each node of deliverance responds to equal volume to any other node in the same system. Apart from the distribution system the standard also requires a tracer gas sampling and analyzing equipment which has adsorption samplers (NT VVS 118).

3.3 Choice of method

The method chosen for this study was the one of homogeneous emission. This method is superior in terms of conclusions one can draw from it. In contrast to the diminishing concentration method this method is available to say something about the quality of the air in specific points in the room which is desired in this case since it is the air in the occupancy zone which is of interest. This method is however more complicated when it comes to the distribution of the gas but the output of the method makes up for this. Another pro of this method is that it allows for continuous measurement, meaning that you do not have to interrupt the measurement for refill of gas. The standard chosen for this method was the Nordtest standard for the simple reason it was available free of charge and is very alike the ISO standard.

3.4 Temperature efficiency

In order to compare the different systems in terms of cooling a term called temperature efficiency is used. Temperature efficiency is a measure of the conditions in the exhaust air, supply air and the temperature of a point in the room according to formula 17.

$$\varepsilon_T = \frac{T_f - T_t}{T_a - T_t} \tag{17}$$

Where ε_T is the temperature efficiency, T_f is the temperature in the exhaust air diffuser, T_t is the temperature in the supply air diffuser and T_a is the temperature in the point measured.

3.5 Requirements and limitations

These methods/standards with homogeneous emissions can be applied to all ventilation principles, any building and regardless of the use of the building. During the sampling period average values of mean ages of air is determined. For short term averages pumped sampling is better while integrating diffusive sampling is better for long term averages (Ecaservice).

4 Laboratory setup

The chosen method for measuring air exchange efficiency requires an experimental setup. Since the foremost intended use of the ventilation technique investigated is office buildings it is important that this experiment area mimics the properties of such a room. Such a room was available to the study and is specified in this section of the report.

4.1 Test room

The test room was built in 2012 inside a larger surrounding room like a box as a typical cell office for one person. It measures 2.4 m width, 5.1 m length and 2.7 m height, see Figure 4. The floor height is adjustable which creates a small area under the floor.

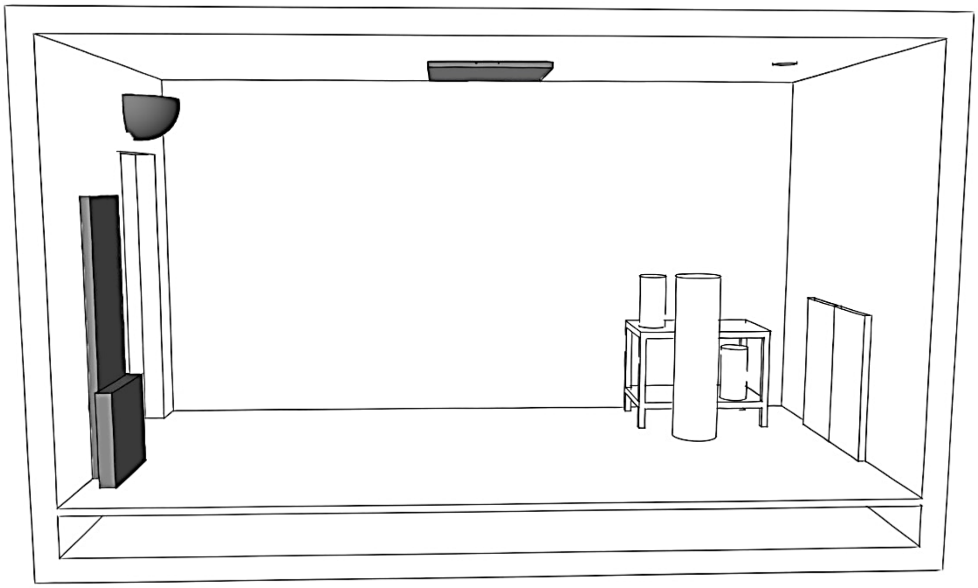

Figure 4 – Model showing test room layout

The external surfaces have been coated with 150 mm extruded polystyrene foam with $\lambda = 0.037$ W/(m·K). For natural light without compromising heat transmission it has four 3-glass windows, two on each long side measuring 0.2 m width and 2.0 m height. The U-value for the walls have been calculated to 0.31 W/(m²·K). The surrounding room has large windows that has been shaded with vertical venetian blinds (Erlandsson & Glyré, 2014).

Supply air into the office space was constant at 18 °C with an average tolerance of ±0.5 °C. The relative humidity was assumed to be constant. The air velocity outside the test room was assumed to be low (≤0.03 m/s) from last year's simulations. All heat loads are gathered with short distance to each other on the other side of the entrance door, this was assumed to be normal.

4.2 Internal gains

The simulation room has been equipped with a computer with monitor, one ceiling light, a window simulator and a heat load representing a working person. Simulation have been tested with both constant heat loads and variable heat loads. For constant heat loads max power was used according to Table 2, sun simulator was not active. For variable heat loads Figure 5 shows the schedule for how the effect varies.

Table 2 – Constant heat loads

Internal gain	*Labeled power*	*Measured power*
Computer	90 W	82 W
Lighting	72 W	93 W
Human heat load	100 W	90 W
∑	262 W	265 W

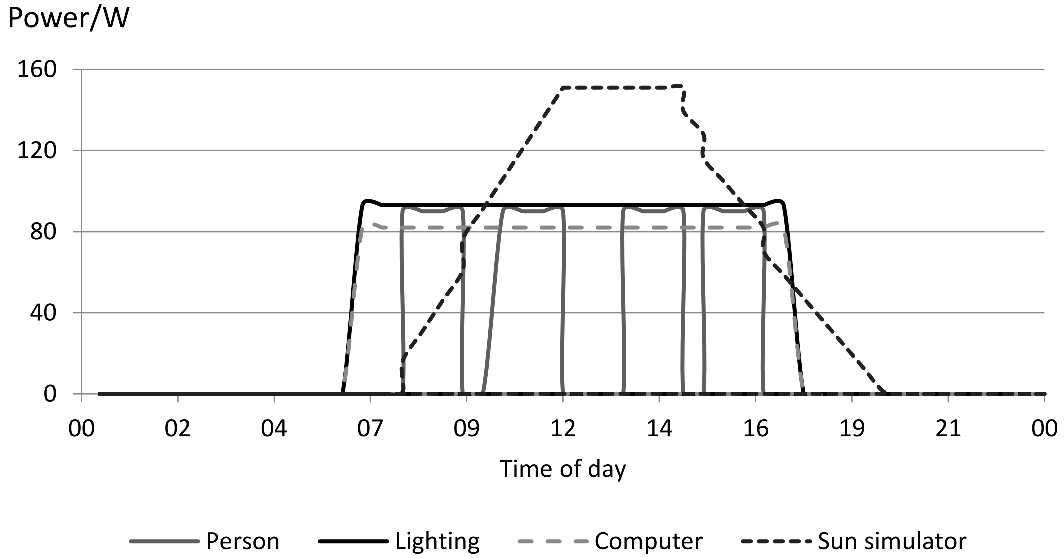

Figure 5 – Variable heat loads schedule where computer and lighting is on all the working day, person and sun simulation is varied.

4.2.1 Person simulator

To simulate a person a closed cylinder made of cardboard with a diameter of 0.32 m, height of 1.20 m with a total surface area of 1.85 m^2 was built. Inside this a smaller cylinder made of metal was built with a diameter of 0.16 m, height of 0.40 m. A Eurofarm red IR light bulb 100 W PAR (E27) was installed in the bottom of the metal cylinder to prevent high surface temperatures of the cardboard cylinder. This light bulb is special made to correspond to human heat transfer. During this experiment only sensible heating is considered which is around 75 - 95 W with a surrounding temperature at 20 - 25 °C (Erlandsson & Glyré, 2014).

The schedule for a working person was based on a study of an office for during which hours the office was occupied (Johansson, 2005).

Table 3 – Simulator schedule for a working person

Time of day	***Activity***
08:00 – 09:30	Occupied
09:30 – 10:00	Break
10:00 - 12:00	Occupied
12:00 – 13:30	Break
13:30 – 15:00	Occupied
15:00 – 15:30	Break
15:30 – 17:00	Occupied

4.2.2 Computer equipment

The computer was simulated using two metal cylinders measuring 0.20 m diameter and 0.40 m height. For the simulated monitor a light bulb of 30 W was used while a 60 W light bulb was used for the computer itself. Computer was turned on from 08:00 – 17:00 which was considered a normal working day.

4.2.3 Lighting equipment

One ceiling light source located behind working space was used. Location was considered normal in a cell office and the light source consisted of two Philips 827 TL-D 36 W fluorescent lamps, a total of 72 W. Lighting was turned on from 07:00 – 18:00 during variable heat load simulations. For constant heat loads it was turned on all the time. When the lighting was in use a power of 93 W was verified with a multimeter.

4.2.4 Sun simulator

In order to maintain a reliable and realistic test the heating effect of the sun was simulated. In older studies a sun simulator built with two 0.013 m concrete boards (0.9 m x 2.3 m) with an electric heating cable of 235 W was used. The problem of this solution was heat accumulation which removed the effect of heat peaks which was the main intention of the simulator. A new model was built using two Ebeco mirror heaters 1004x524 mm^2 with a total effect of 200 W, each agglomerated to a metal sheet. After testing the system, it was found that the voltage varied slightly during the time it was operable. The voltage and power was measured using a multimeter in order to determine what actual values the two varied between.

The sun simulator was connected to the computer that manages all of the heat loads in the room. The computer was not able to directly reduce the voltage of a power socket but could instead choose whether it should be turned on or off. The computer used input information of what the peak effect of a heat load in combination with information regarding the desired efficacy of said heat load. This information was used to know during how long intervals the power socket should be active in order to deliver the desired load. The computer was programmed so that it divided simulated time into packets of 90 seconds each. The reason for dividing the simulation time into smaller packets is to prevent a heat load from being on for hours and later shut off for hours to get the right delivered power. A simulation lasting less than 90 seconds will not yield any result that matches the input data due to the “on-off cycle not being complete.

Example: The sun simulator is able to deliver 200 W of power as a heat load however only 151 W is desired. This means that during a timeframe of 90 seconds the socket needs to be active $\frac{151}{200} = 75.5$ % *of the timeframe.*
This means 67.95 s active and 22.05 s switched off.

The simulator is then connected through the computer program with a time schedule to simulate the sun path and when the sun hits the window etc. The old sun simulator had a peak value of 151 W during spring/autumn which gives a factor of around 63 % from the max output of the system. With the newer system a new factor had to be calculated to 88 % to produce the same effect of 151 W during its peak. These values can be seen below in *Table **4***.

Table 4 – Schedule over solar gains

Schedule Sun	
Simulation time	12 h
Peak load	151 W
Time at peak power	180 min

4.3 Gas distribution and gas measuring equipment

A system was needed to be developed to distribute and measure a tracer gas throughout the room according to homogeneous emission method. A tracer gas also had to be chosen to best fit the properties of air.

4.3.1 Tracer gas analyzing

For this study a 1303 Multipoint Sampler & Doser with 6 sampling channels and a 1312 Photoacoustic Multi-Gas Monitor was used. For reliability a repair and calibration was carried out by LumaSense Technologies in Denmark for both instruments. The 1312 was calibrated with a UA0984 optical filter (Dinitrogen oxide) which gives a linear measurement range of 0.5 – 250 PPM.

The 6 channels for sampling were distributed according to Figure 6 in a center line of the room so that one channel measured in the exhaust diffuser, two channels measured the working place and the last three measured outside the working area to get a larger view of how the tracer gas moved.

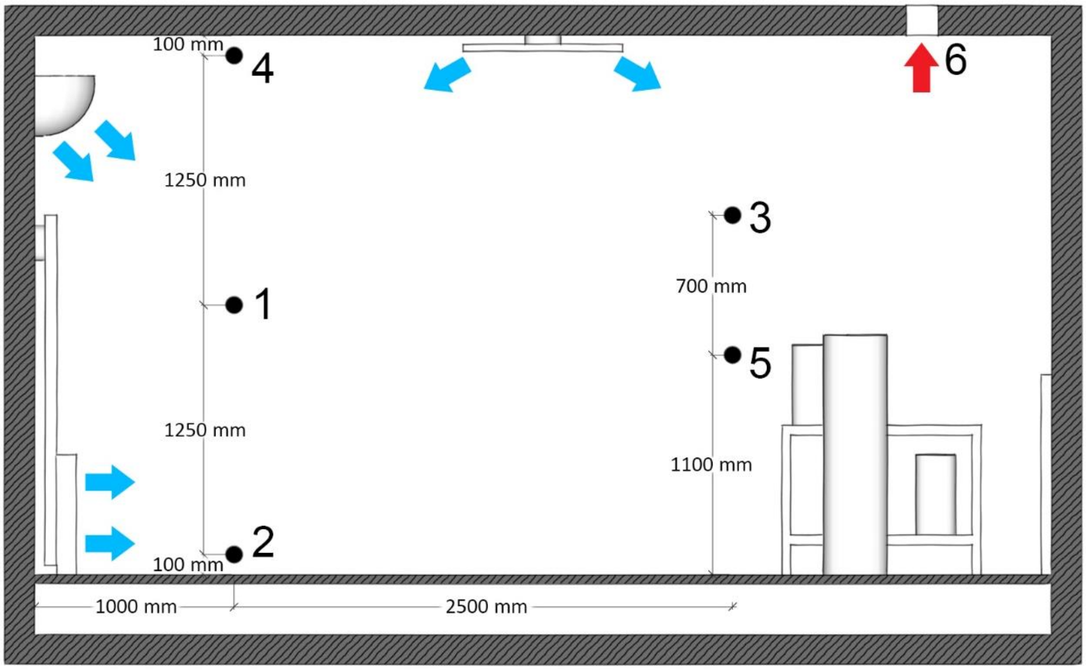

Figure 6 – Tracer gas measure points in center of the test room

The 1303 offers the ability to perform multi-point monitoring and multipoint air-exchange analyses. In this report only the sampling from its 6 inlet channels was used. Air samples from the 6 channels are then directed to the 1312 for analysis (Sampler).

The 1312 is a highly accurate, stable and reliable gas monitor with room for up to 5 optical filters. It measures photoacoustic infra-red light meaning it can measure almost any gas. Thanks to the ability of compensating for other gases, water-vapor and temperature fluctuations the results are very accurate while self-tests are performed automatically to ensure reliability of measurements (Ierents 1).

The gas monitor works by drawing air with a pump from the sampling area, in this case the air samples gathered with the 1303. These air samples are then hermetically sealed in an analysis cell. With a mirror, infra-red lighting from a light source is then reflected through a chopper wheel which oscillates the light. The oscillating light is then transmitted through the optical filter where it afterwards is selectively absorbed by the sealed gas. Due to the oscillation the temperature of the gas increases and decreases which causes an equivalent behavior in the gas pressure, an acoustic signal. This signal is then measured by two microphones in the analysis cell to get the concentration of the gas since the concentration is directly proportional to the pressure behavior (Ierents 1). To export and evaluate the data collected from the gas monitor a program called 7620 (v.5.02) was used together with Microsoft Excel.

4.3.2 7620 Tracer gas program

To measure data and export values the program 7620 was used. It was connected to a computer with RS-232 communication. To connect the computer with the multi-gas monitor properly to avoid an error called “Communication error with Gas-Monitor” all preferences was set according to Table 5 for both the Gas-Monitor, 7260 software and in preferences for COM1 port in windows hardware monitor. Units were also set to match.

Table 5 – RS-232 Communication parameters

Baud Rate	*9600*
Stop Bits	1
Data Bits	7
Parity	Even
Hardwire Mode	Switched-Line
Handshake Type	Hard-wired

Another important setup was to measure both dinitrogen monoxide and carbon dioxide otherwise the program did not receive any values at all. For the results only measured values from dinitrogen monoxide was evaluated.

4.3.3 Tracer gas choice

The choice of tracer gas was between two commonly used tracer gases; sulfur hexafluoride (SF_6) and dinitrogen monoxide (N_2O). The density of the gases was one of the major factors in determining which gas should be used, the sulfur hexafluoride and the dinitrogen monoxide has a density of 6.26 kg/m^3 and 1.87 kg/m^3 respectively. Since the tracer gas is meant to mimic normal indoor air which has a density of roughly 1.26 kg/m^3 the most suitable choice was dinitrogen monoxide (Airliquide 1).

4.3.4 Tracer gas distribution

For distribution of tracer gas a system had to be custom built. The requirement of this system was to be able to deliver very low flows of tracer gas in outlets evenly distributed throughout the room with low velocity. The flow had to be low enough in the system to release gas with a concentration in the range of 0.5 – 250 PPM. To be able to reduce the pressure to a wanted level a regulator was ordered from Air Liquid with an input pressure range of 0 – 315 bar and an output pressure range of 0.01 – 0.02 bar. For even further pressure control a RGC1260 rotameter with integrated needle valve and visual fluid monitoring was ordered from Rotameter-shop. This type of rotameter was designed for measurements of gas and liquid flows and had a min and max flow range of 0.2 l/h – 1.9 l/h (Rotameter). Pneumatic tubes and distributor blocks to connect everything were ordered from *Festo AB* to build this system.

Figure 7 presents the diagram over the gas distribution. A gas tube with a directly mounted regulator (A) was connected to a rotameter (B). From this rotameter a L-shaped 3-way distributor (C) was directly connected with each of its outlets going to its own distributor block (D) with 8 outlets each, making it a total of 24 outlets.

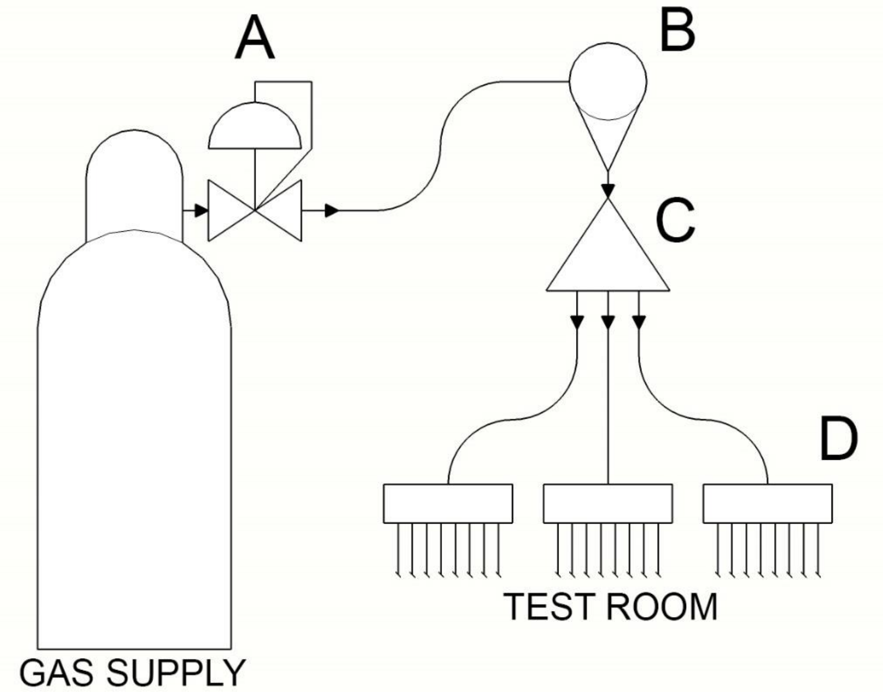

Figure 7 – Pneumatic diagram for gas distribution

Table 6 – Connection type and length for tracer gas distributions system

Number	***Type***	***Length***	***Inside Ø***	***Outside Ø***
A	Tank pressure regulator	-	-	-
AB	Plastic tubing (PUN-H-4X0,75-SW)	1 m	2.6 mm	4 mm
B	Rotameter	-	-	-
C	Multiple distributor (QSLV3-G1/8-4)	-	-	-
CD	Plastic tubing (PUN-H-2X0,4-SW)	0.4 m	2.6 mm	4 mm
D	Distributor block (FR-9-M3-B)	-	-	-
D-Testr.	Plastic tubing (PUN-H-2X0,4-SW)	10 m	1.2 mm	2 mm

To be able to meet the requirements of homogeneous emissions the gas had to be evenly distributed across the whole room in a 3D grid. This grid was made using symmetry lines dividing the floor area into 8 equally big squares which was discussed to be enough, visualized in Figure 8. The same method was then used when adding the height dimension to the grid resulting in an equal distance between each node of tracer gas supply tube creating a distribution system with a total of 24 outlets on heights of 45 cm, 135 cm and 225 cm.

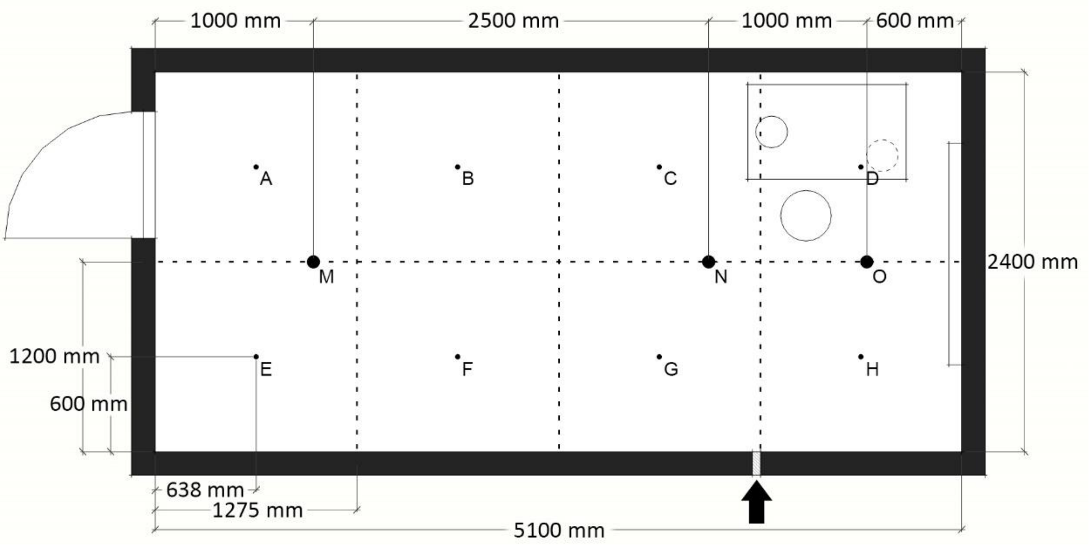

Figure 8 – This figure shows distribution points for gas supply (A-H) and temperature measure points (M-O)

4.3.5 Tracer gas distribution system pressure drop

To ensure equal gas distribution between each of the 24 outlets a pressure drop calculation was made using an online calculation tool (Pressure 1). It was considered that if the pressure drop in each distribution block was considerably lower than the pressure drop in the outlet tubes it would not make an impact and therefore the flow out of each outlet could be considered equal.

From the distribution diagram in Figure 7 there was two distribution blocks to calculate. Section C, Multiple distributor (QSLV3-G1/8-4) and section D, Distributor block (FR-9-M3-B). In Table 7 all flows are presented which were needed for the calculation.

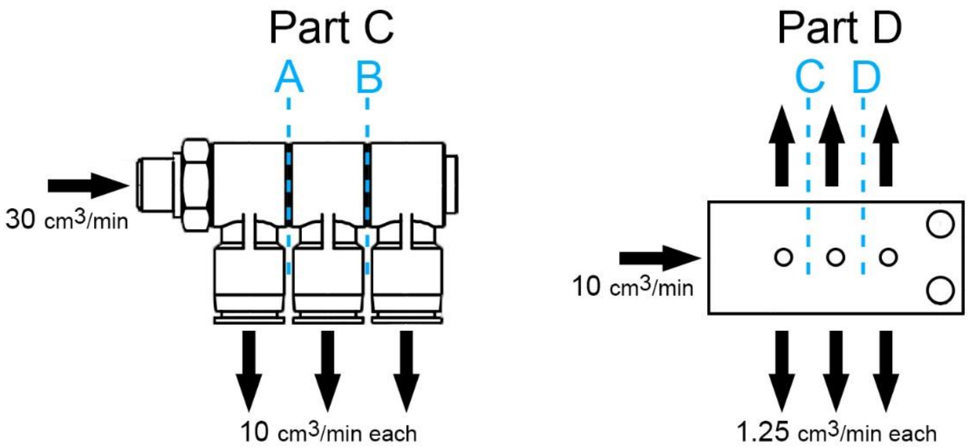

Figure 9 - Flow chart for each distribution block that was calculated

Table 7 – Pressure drop calculation for tracer gas distribution system

Part	Section	Flow rate (q)	Length (L)	Pipe (Ø)	Pressure drop (Δp)
C	Line A	20 cm^3/min	8.5 mm	8.8 mm	2.83E-4 Pa
C	Line B	10 cm^3/min	8.5 mm	8.8 mm	1.41E-4 Pa
C-D	Tubing	10 cm^3/min	400 mm	2.6 mm	8.74E-1 Pa
D	Line C	6.25 cm^3/min	2 mm	8.8 mm	2.08E-5 Pa
D	Line D	2.5 cm^3/min	2 mm	8.8 mm	8.32E-6 Pa
D-Testr.	Tubing	1.25 cm^3/min	10 000 mm	1.2 mm	60.2 Pa

From this result it can be seen that the pressure drop is so low in each distribution block compared to the outlet tubing that it will not have any impact on the gas distribution in each outlet. This means that the same flow of dinitrogen monoxide will be supplied from each of the 24 outlet.

4.4 Temperature sensors

For this study eleven PT1000 temperature sensors were used. One in the inlet and one in the outlet while the other nine measure room temperature three different heights of 10 cm, 110 cm and 180 cm according to last year's study to be able to compare them. The sensors were calibrated by lowering all of them together in water and then calibrating each sensor so they show the same temperature with the largest difference of 0.3 ^{0}C. All temperatures were measured in a five second interval by a control unit called Bastech BAS2 XE16 – COM.

4.5 Supply air diffusers

Mixed air was supplied with a Swegon Colibri CC diffuser connected to an ALSd 125-200. It is placed in the middle of the room with its nozzles about 100 mm down from ceiling level oriented for radial diffusion.

Figure 10 – Swegon colibri mixed air diffuser

Airson Airshower is one of the displacement ventilation diffusers. A vertical wall placed supply air diffuser located 200 mm above the entrance door. This placement has minimal impact in the occupied zone and it contributes to easier duct connection for collection ducts outside. A benefit with the diffusers high location is that when the air is falling to the floor to create displacement ventilation without mixing with room air it heats up reducing risk of draft.

Figure 11 – Airson Airshower displacement ventilation diffuser

Another displacement ventilation diffuser used is the Swegon DIR-c-600 connected to a DIRT 4-600. This is a standard displacement diffuser and it is located to the right of the entrance door at floor level.

Figure 12 - Traditional displacement diffuser at floor level

5 Results

5.1 Calibration of equipment and room

Pressure, temperature, traces of tracer gas and ventilation system was tested before any simulation was started to ensure test lab was working as intended. The ventilation system was tested by starting a simulation with a static supply of tracer gas and constant heat loads which should after some time come to equilibrium. The results in Figure 13 showed that the concentration increased a lot during the night and this was due to the ventilation system automatically turning off, this problem was corrected.

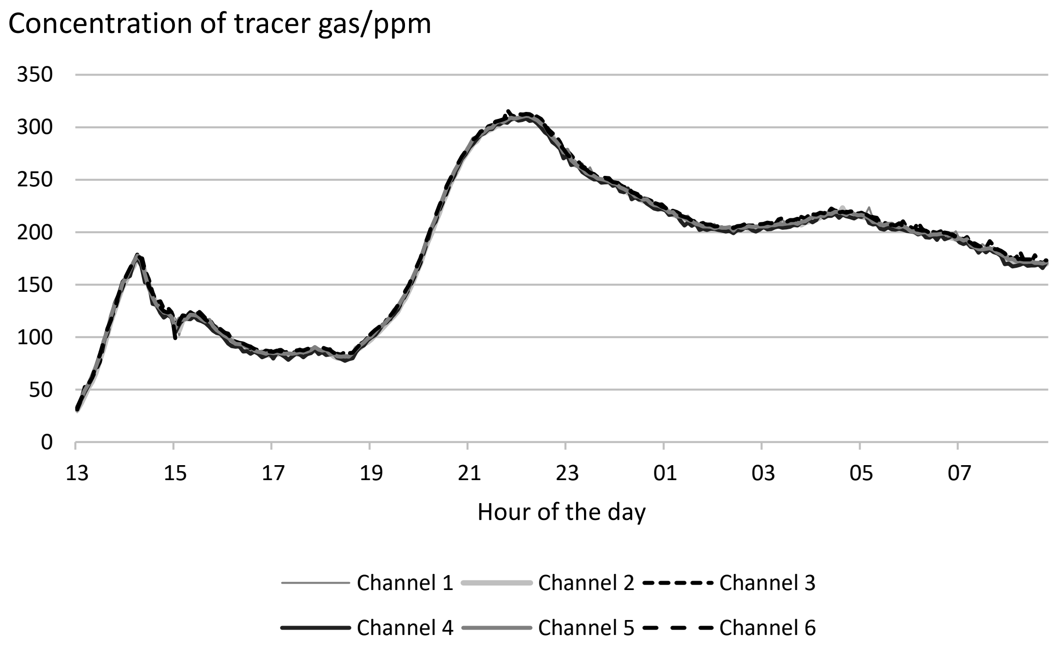

Figure 13 – Ventilation test

Investigation of tracer gas was conducted to determine if there was any natural presence of tracer gas without supplying it. In Figure 14 it can be seen that there was a very small amount of tracer gas left from a previous with starting levels of 2.5 – 3 ppm. Without supplying tracer gas, it decreased as expected.

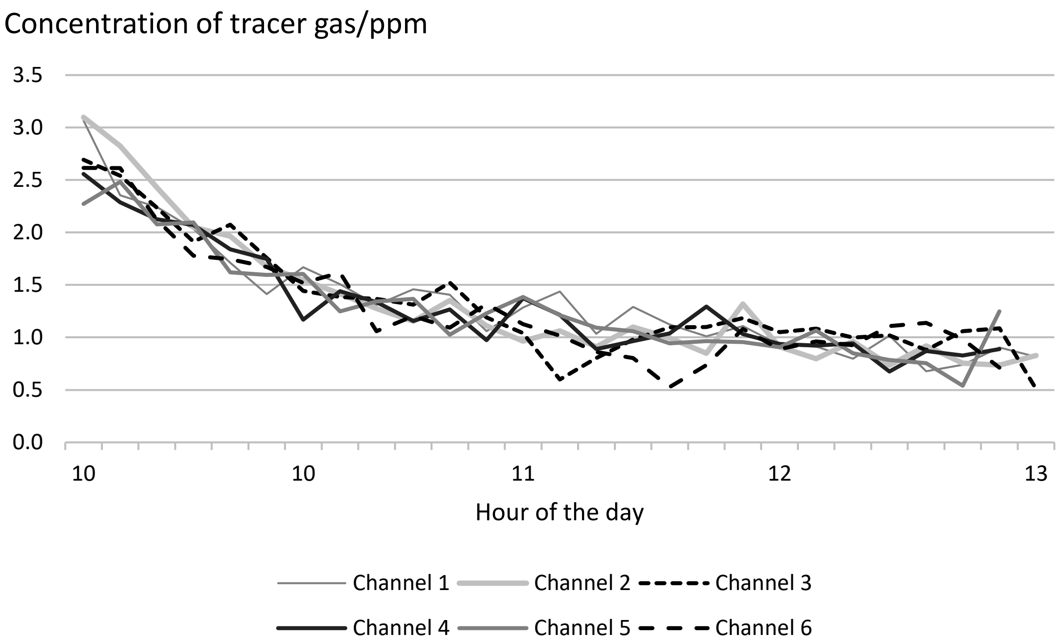

Figure 14 – Tracer gas test

Temperature sensors were checked without heat loads. All sensors showed correct values within the Δ0.3 °C. It can be seen in Figure 15 that GT2:6 had a higher temperature than the rest of the sensors both with and without fans running. This was probably due to a hole in the wall made for cables. This hole was enclosed to prevent heat transfer.

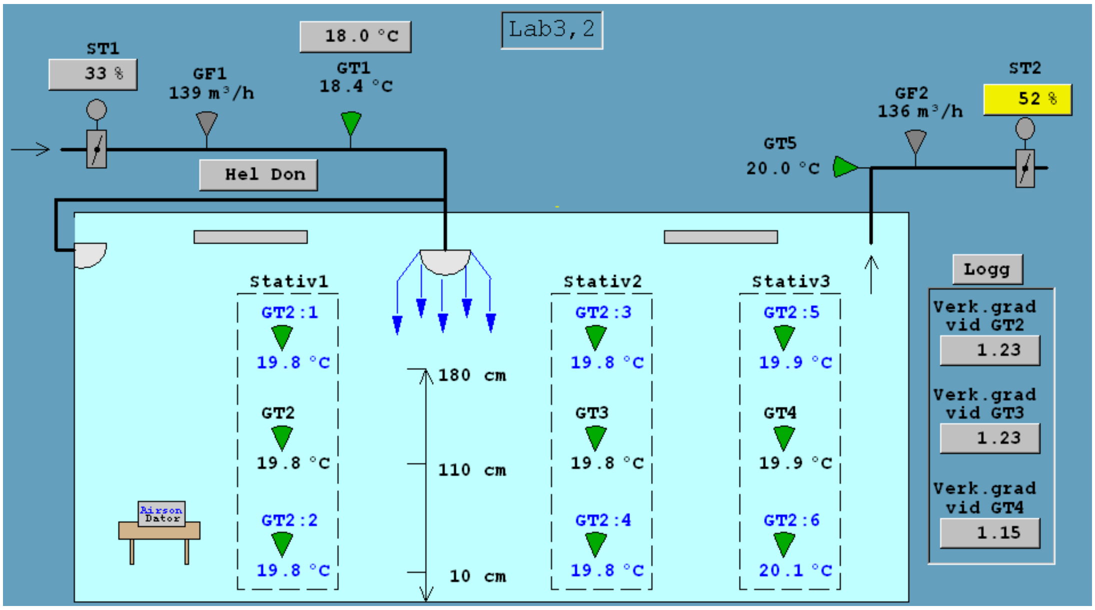

Figure 15 – Temperature check of sensors

Pressure was checked before every simulation between test room and surrounding room to be in an interval of 1 – 10 Pa.

5.1.1 Control of voltage and power for maximum sun load

Due to uncertainties in delivered voltage and power through the power socket these values were measured to calculate average values. The average values were then compared to the old sun simulator which delivered 151 W to determine the tolerance and the reduction factor. Table 8 shows the measured values for the voltage and power using a multimeter connected to the power socket of the sun load.

Table 8 – Voltage and power for sun simulated in chronological order of measurements

Voltage V	Power W
221	178
224	182
217	170

$$Average\ voltage = \frac{221 + 224 + 217}{3} = 220.6\ V$$

$$Average\ power = \frac{178 + 182 + 170}{3} = 176.6\ W$$

5.2 Mixed air ventilation

5.2.1 Constant heat loads

During this test the mixed air ventilation was running with constant heat loads from computer, person and lighting and without sun. Tracer gas flow was set to constant 30 cm^3/min but it slightly rose during the simulation to around 35 cm^3/min throughout the simulation period. Ventilation was turned on all the time.

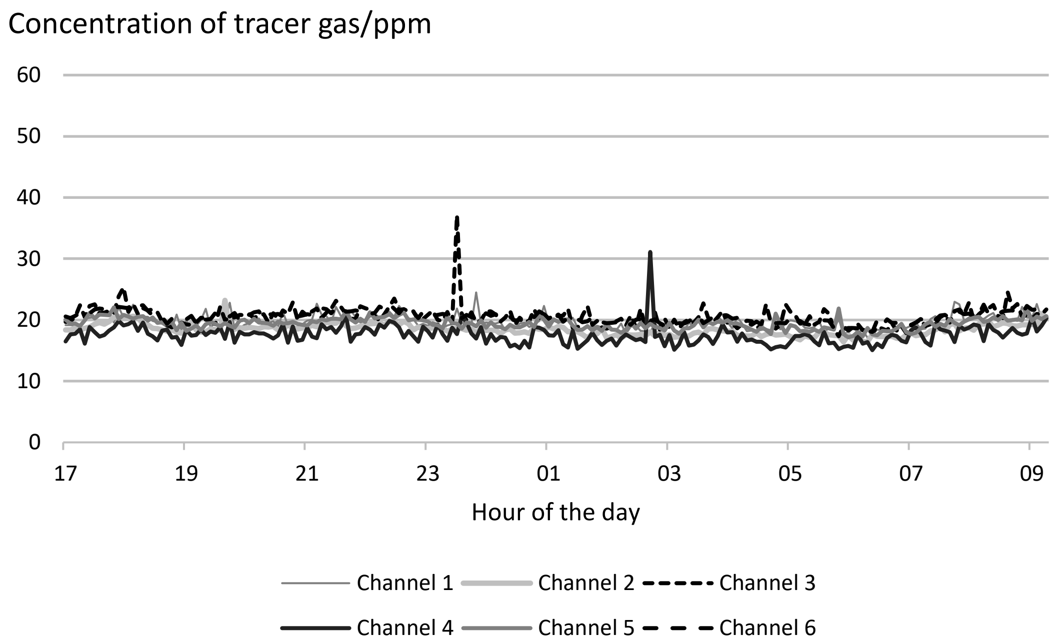

Figure 16 – Tracer gas concentration for mixed air with constant heat loads

Figure 16 shows the concentration of dinitrogen monoxide from the start value of 20 ppm in the room. The concentration varied slightly during the length of the simulation but was at its equilibrium during the entirety of the measurement. It can be seen that with mixed air the highest amounts of concentration occurs in the exhaust air (Channel 6).

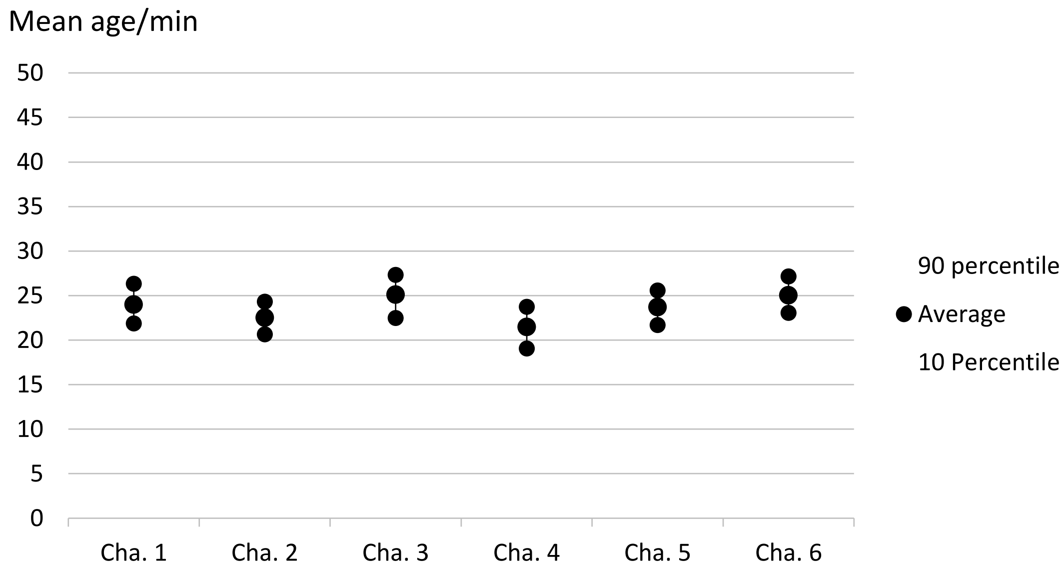

Figure 17 – Mean age distribution for mixed air with constant heat loads with average value, the 10th percentile and the 90th percentile

Figure 17 shows how the mean age of air is distributed for each channel measured. The figure shows 80 % of all the values in order to remove the short duration peaks which otherwise would twist the result. The three dots on each line represents the limits and mean value of each channel. Channel 2 for example has 90 % of its values above 20 minutes of age, a mean value of 23 minutes and 90 % of its values below 25 minutes of age.

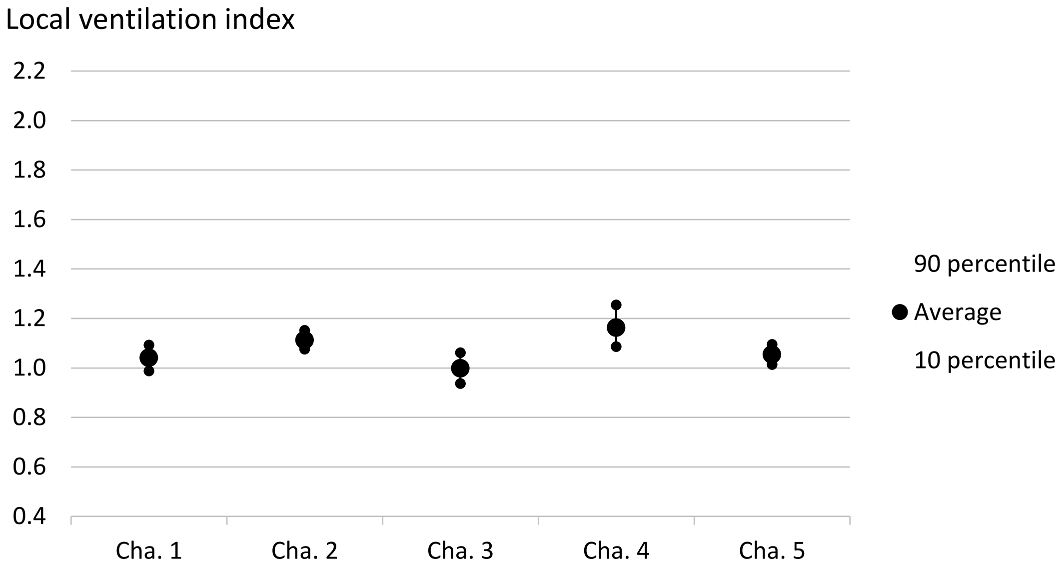

Figure 18 – Local ventilation index for mixed air with constant heat loads with average value, the 10th percentile and the 90th percentile

Figure 18 shows the ventilation index for the mixed ventilation system. Each point has an upper and lower limit which shows what 80 % of the values are spanning.

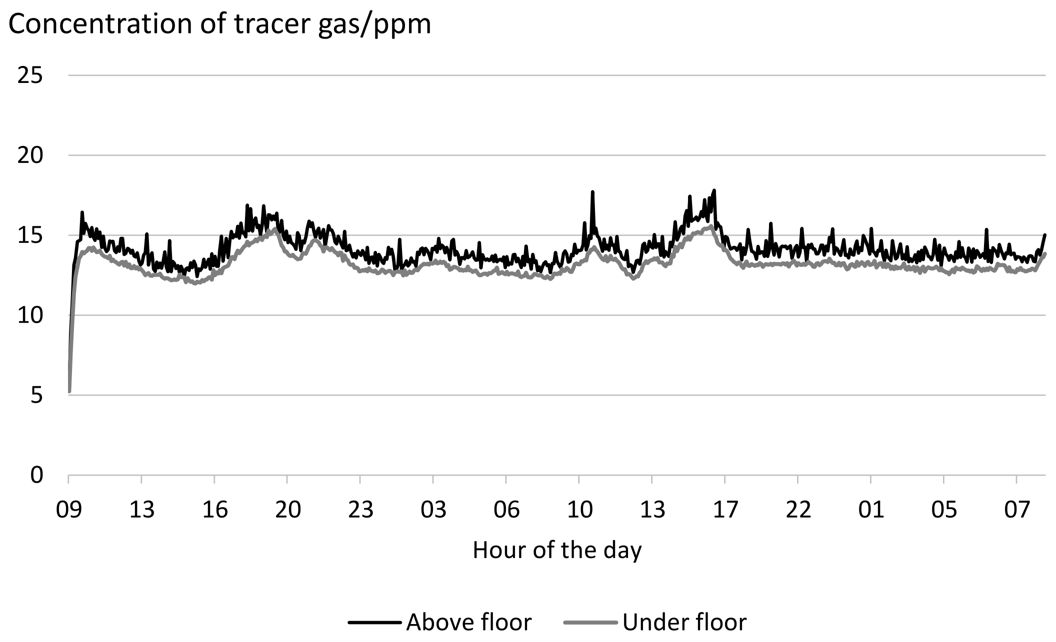

Figure 19 – Concentration of tracer gas for mixed air with constant heat loads above the adjustable floor and under the adjustable floor

Figure 19 displays how the concentration of dinitrogen monoxide changed during the measure time. Average values have been calculated above the floor and under the floor to visualise that the concentration of tracer gas is rather even in both cases. The mixed ventilation system managed to keep the concentration at an intervall between 12 - 16 ppm.

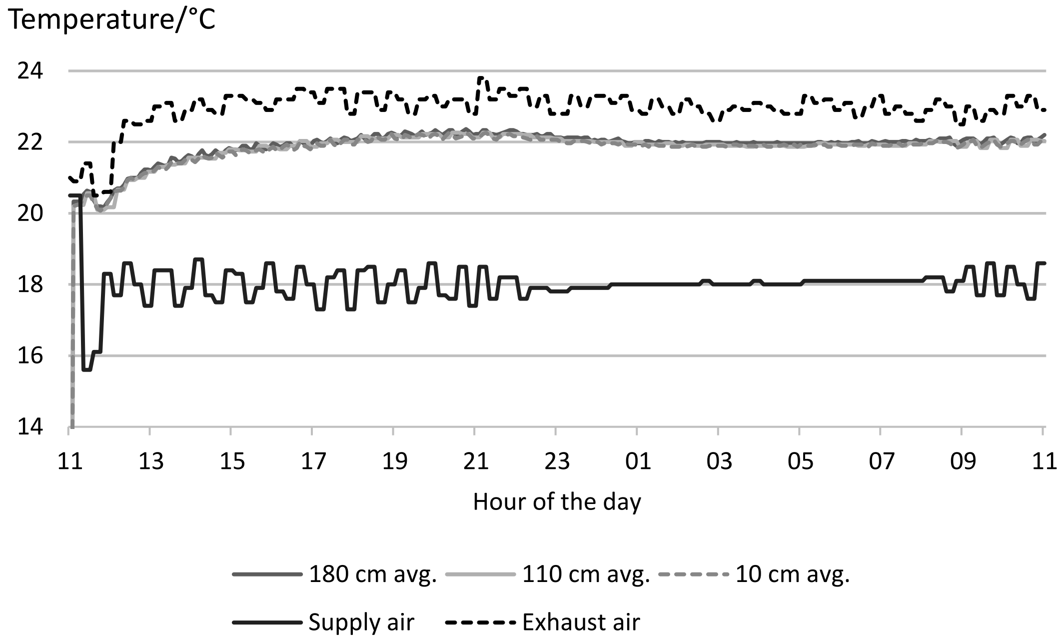

Figure 20 – Average temperatures for mixed air with constant heat loads

Figure 20 shows the temperature of different points and heights in the room. The room temperature was somewhat constant at 22 °C during the experiment for each measure point in the room. What could be seen was that 3.5 m away from the heat source the temperature was on average 0.3 °C lower at 180 cm and 0.7 °C lower at 10 cm.

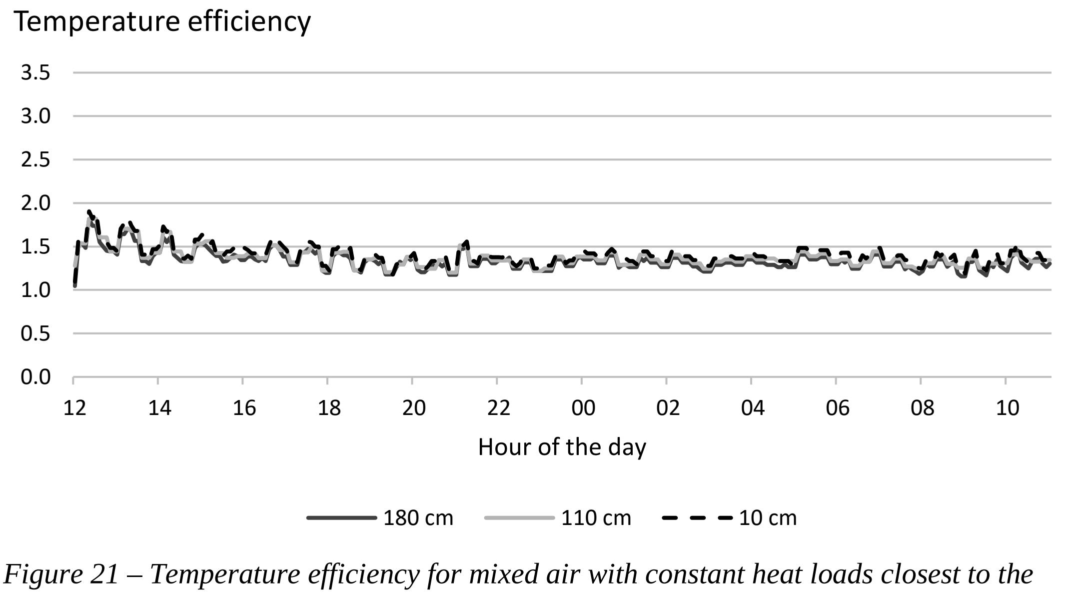

Figure 21 – Temperature efficiency for mixed air with constant heat loads closest to the working space for three different heights of 180 cm, 110 cm and 10 cm

By calculating the temperature efficiency with *formula 17* one could see an efficiency around 1.2 – 1.5 for constant heat loads with mixed air ventilation.

5.2.2 Variable heat loads

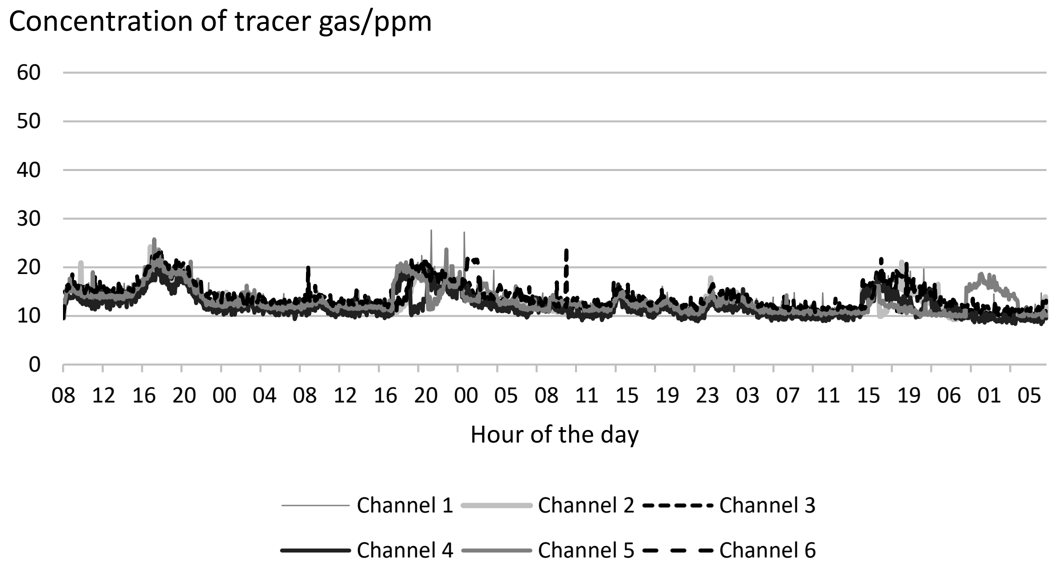

Figure 22 - Tracer gas concentrations for mixed air with variable heat loads

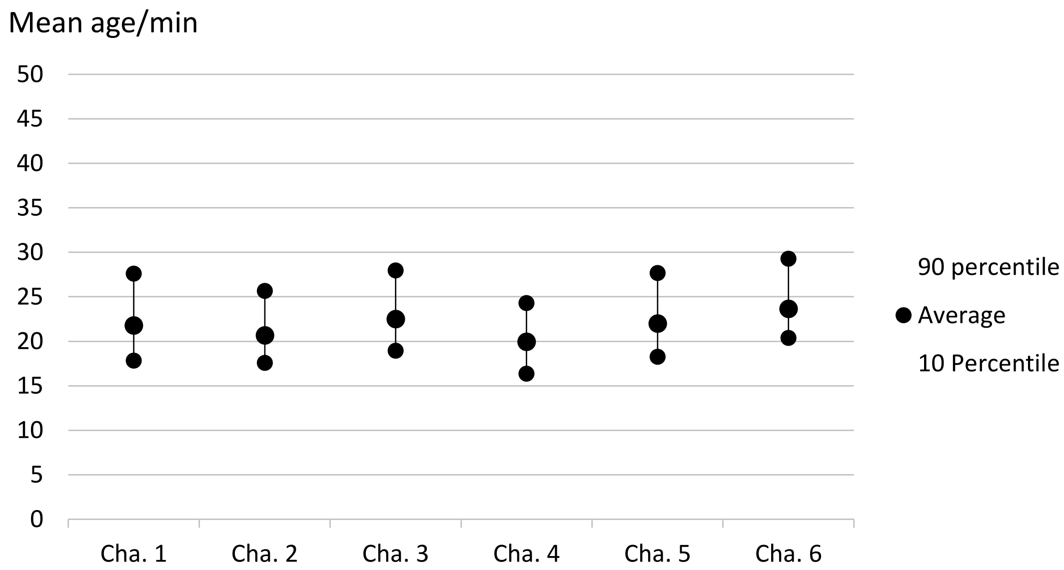

Figure 23 - Mean age distribution for mixed air with variable heat loads with average value, the 10th percentile and the 90th percentile

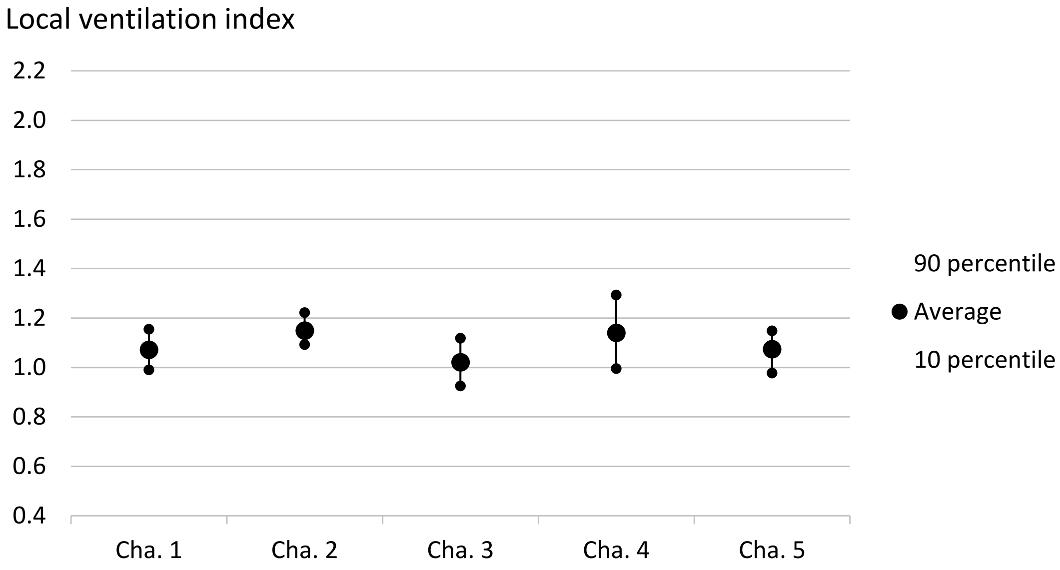

Figure 24 - Local ventilation index for mixed air with variable heat loads with average value, the 10th percentile and the 90th percentile

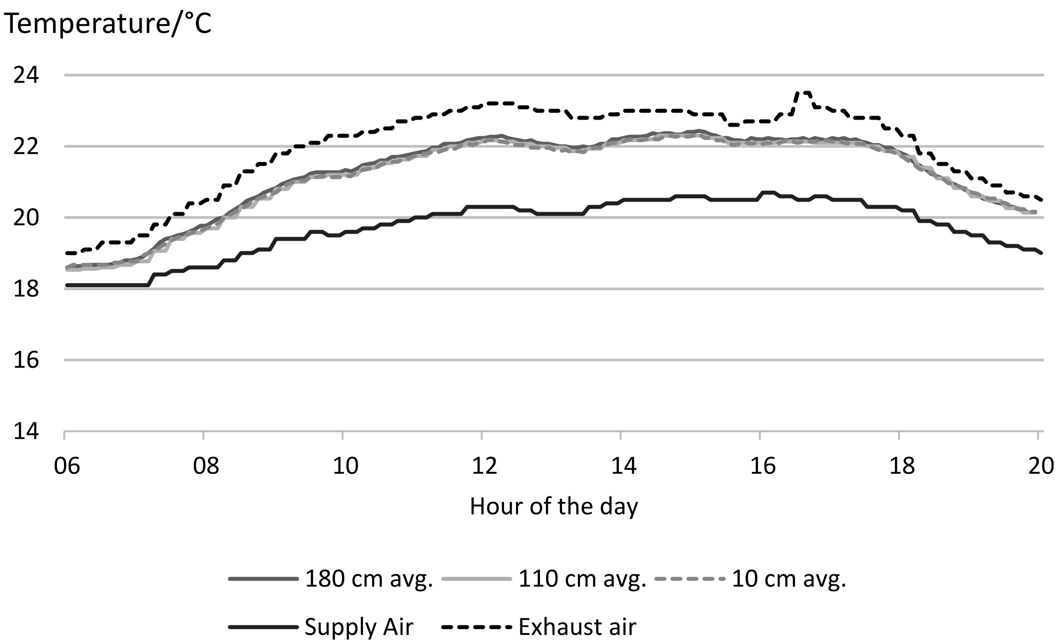

Figure 25 - Average temperatures for mixed air with variable heat loads

Unfortunately, there were some problems with the ventilation system which made the inlet temperature higher than the preferred 18 °C.

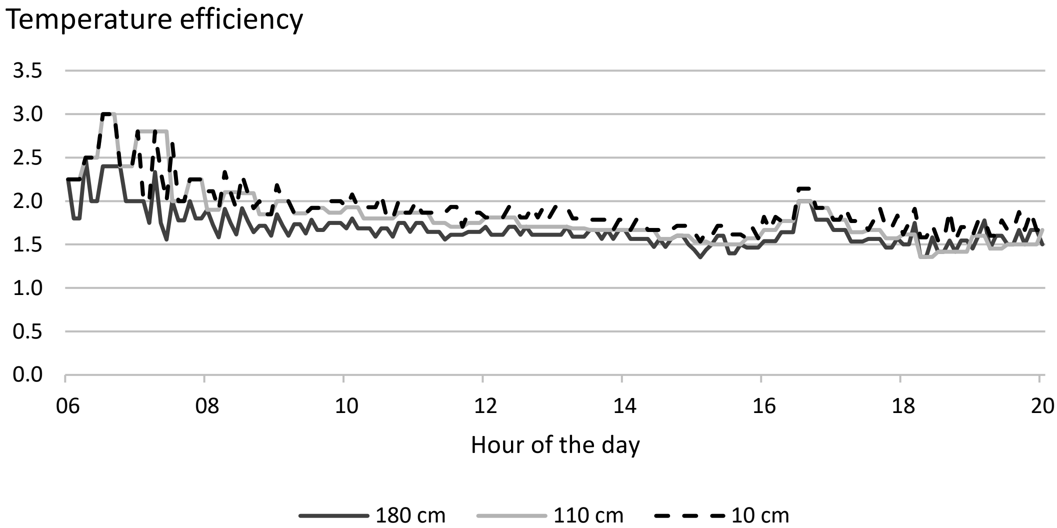

Figure 26 - Temperature efficiency for mixed air with variable heat loads closest to the working space for three different heights of 180 cm, 110 cm and 10 cm

By calculating the temperature efficiency with *formula 17* one could see an efficiency around 1.5 – 2.0 for variable heat loads with mixed air ventilation.

5.3 Airson Airshower

5.3.1 Constant heat loads

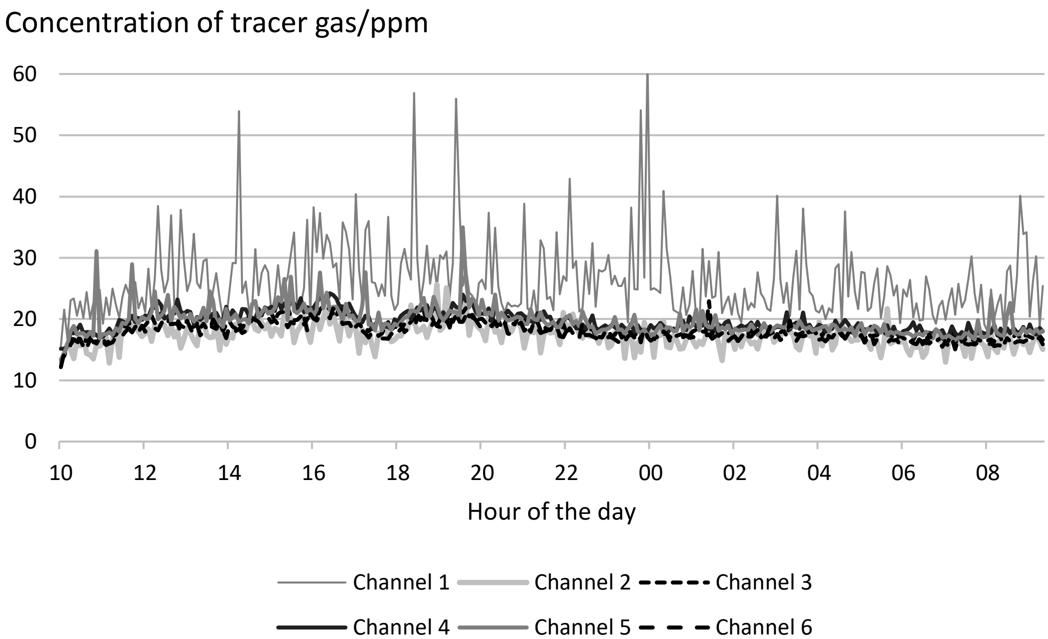

Figure 27 - Tracer gas concentrations for Airshower with constant heat loads

Figure 27 shows the concentration of dinitrogen monoxide where the airshower system reaches its equilibrium. All of the channels showed a value of below 20 ppm where as channel 1 (the one closest to the diffuser and in the middle of the room height wise) showed a higher concentration than the rest of the points. The rest of the room concentration varied between 15 - 20 ppm during the measured time.

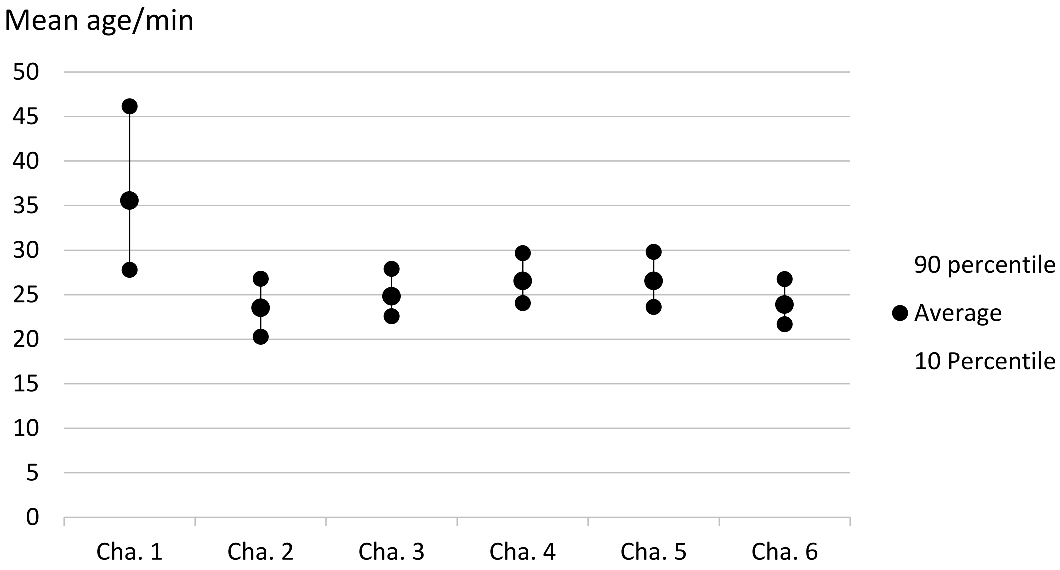

Figure 28 - Mean age distribution for Airshower with constant heat loads with average value, the 10th percentile and the 90th percentile

The mean age distribution is more interesting than the concentration itself since mean age depends on the measured concentration and the supplied amount of tracer gas. Here it can be seen that channel 2 and channel 6 had lowest mean age. Since channel 6 is the exhaust diffuser a problem arises due to a lower concentration being measured than in the rest of the room.

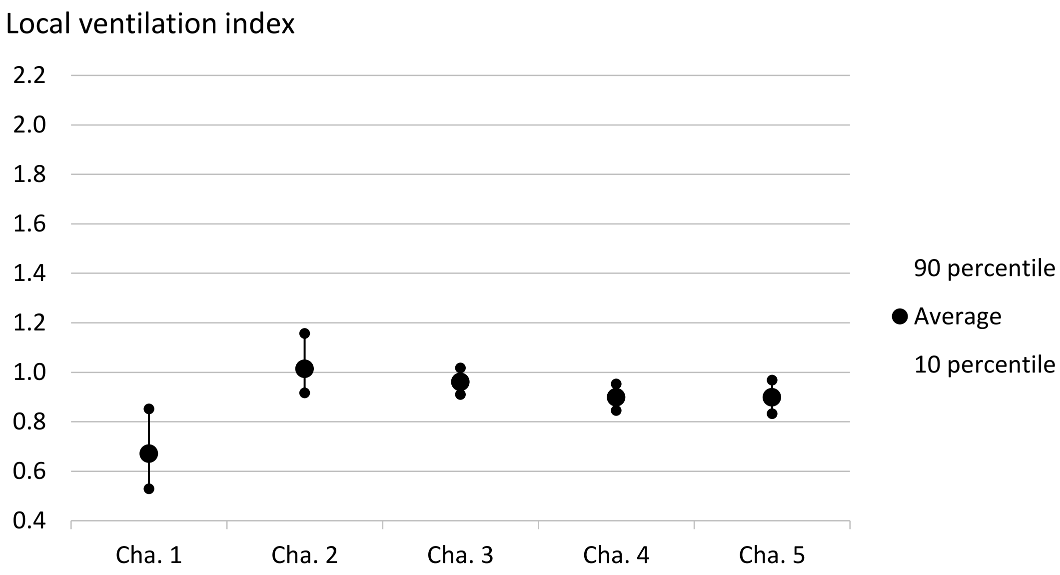

Figure 29 – Local ventilation index for Airshower with constant heat loads with average value, the 10th percentile and the 90th percentile

Figure 29 shows the local ventilation index for each measured point of the airshower diffuser. The low values is a direct consequence of the exhaust air concentration. This figure shows 80 % of the values to avoid sudden brief peaks of the system that would otherwise alter the result unrealisticly. For better understanding, a simulation were made where two channels were located under the floor to see any tracer gas accumulation.

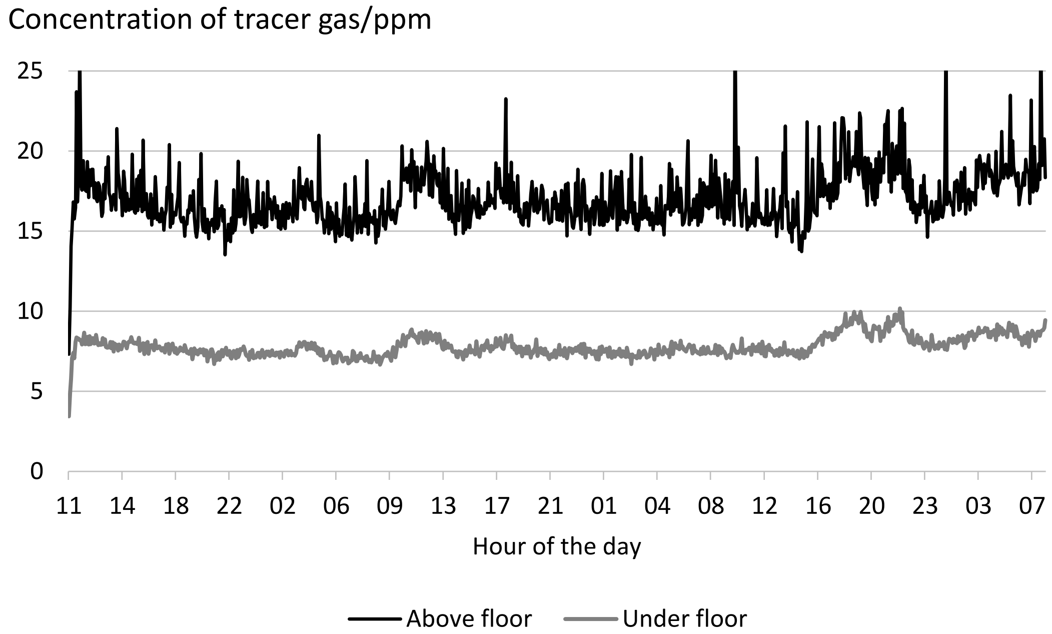

Figure 30 – Concentration of tracer gas for Airshower with constant heat loads above the adjustable floor and under the adjustable floor

Figure 30 shows how the airshower diffuser handles three days of constant use. During this measurement two channels (channel 3 & 5) was placed in the fake floor to measure if there's any accumulation of dinitrogen monoxide in the floor. Both channels looks constant and the rest of the measured points looks as they've done during the other measurements. This shows that there was some tracer gas in the fake floor but it was not increasing nor decreasing which makes accumulation impossible.

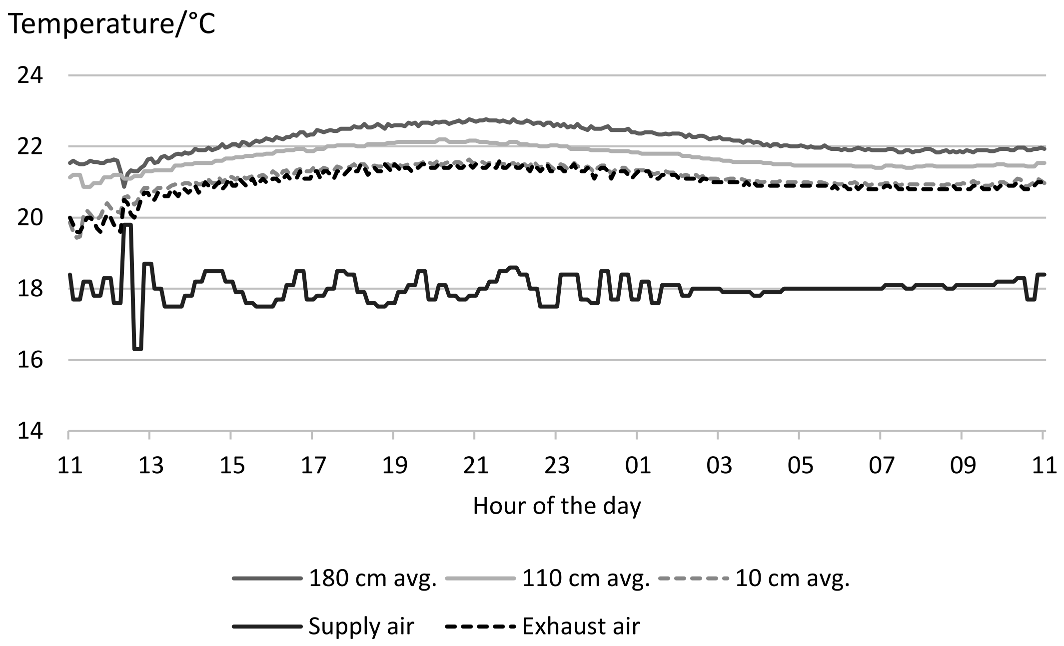

Figure 31 - Average temperatures for Airshower with constant heat loads

Figure 31 shows the temperatures which were measured during the simulation. There were three distinct layers of air in the room with different temperature which is the base on which the displacement techonology is built. All of the temperature measurements were more or less constant during the simulated time. What could be seen was that 3.5 m away from the heat source the temperature was on average 0.3 °C lower at 180 cm and 0.8 °C lower at 10 cm.

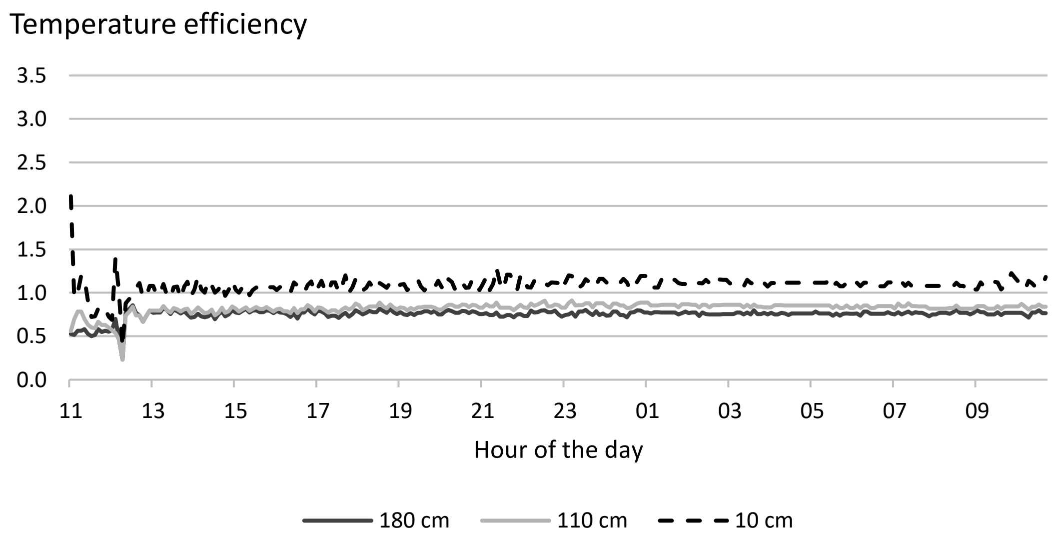

Figure 32 - Temperature efficiency for Airshower with constant heat loads closest to the working space for three different heights of 180 cm, 110 cm and 10 cm

By calculating the temperature efficiency with *formula 17* one could see an efficiency around 0.8 – 1.1 for constant heat loads with Airson Airshower.

5.3.2 Variable heat loads

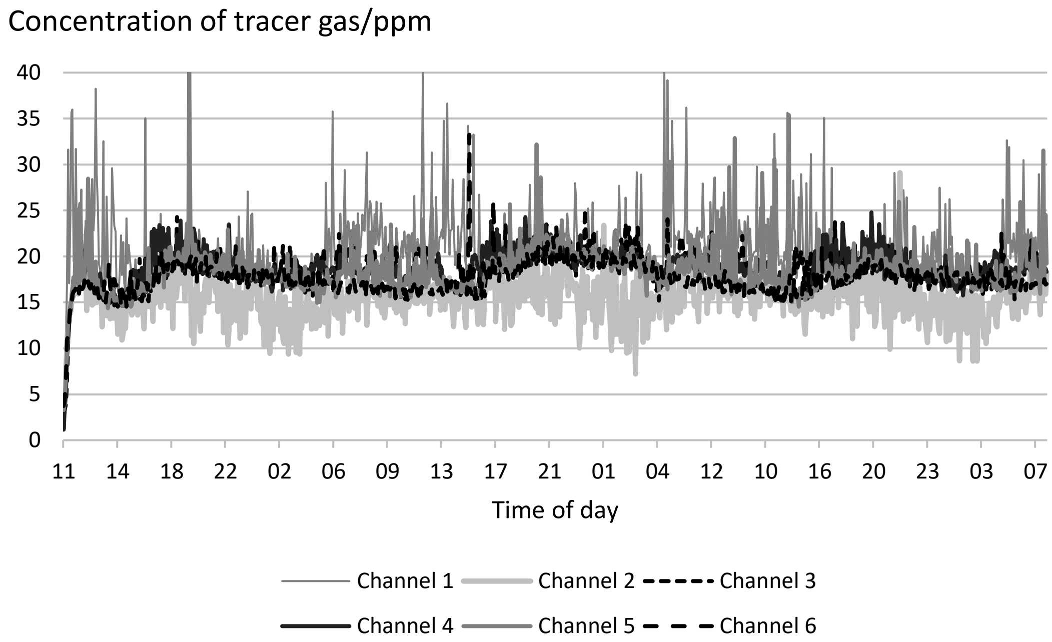

Figure 33 - Tracer gas concentrations for Airshower with variable heat loads

Figure 33 shows the concentration of tracer gas which was measured during the variable heat load simulation for the Airshower diffuser. There was no major difference from this case to the one with constant heat loads.

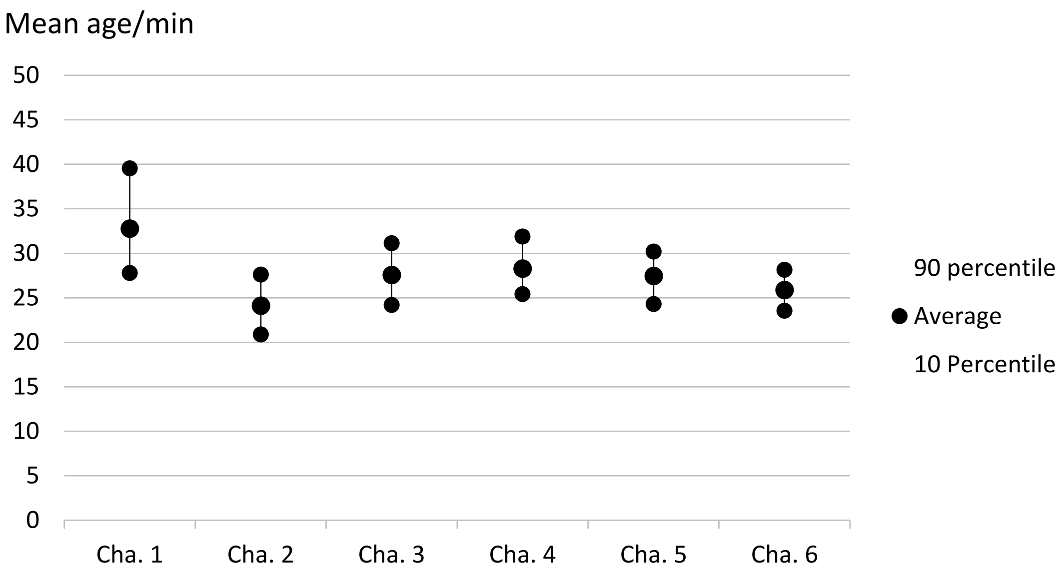

Figure 34 - Mean age distribution for Airshower with variable heat loads with average value, the 10th percentile and the 90th percentile

Figure 34 shows the mean age of the air in the variable heat load simulation. Just like the constant heat load simulation, channel 1 exhibited a high mean age in comparison to the other channels. The exhaust air was also very low when comparing with other channels.

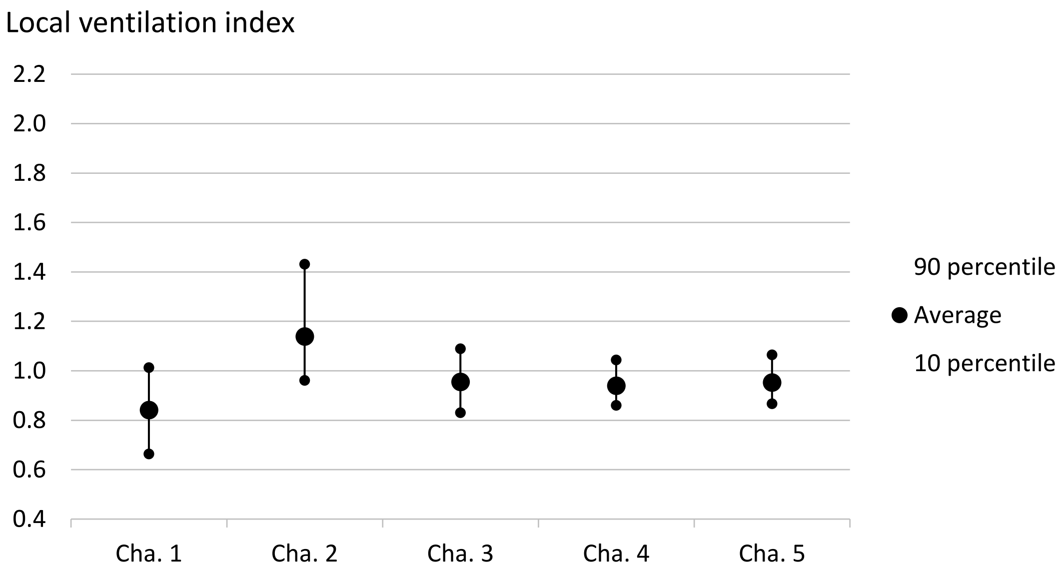

Figure 35 – Local ventilation index for Airshower with variable heat loads with average value, the 10th percentile and the 90th percentile

Figure 35 shows the ventilation index of each individual point measured in the room. However, due to the young age of the exhaust air these values is lower than expected just as the constant heat load case was.

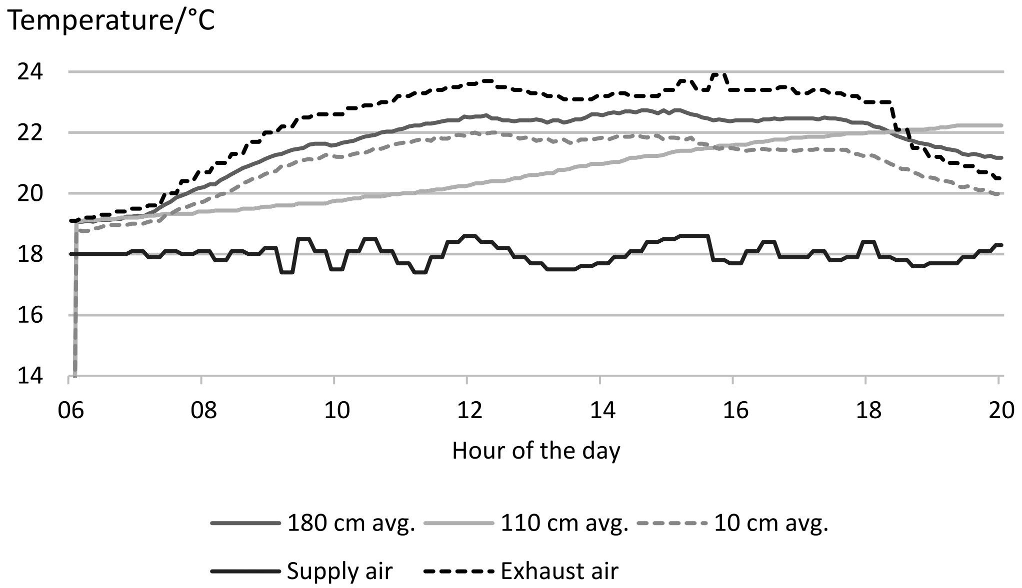

Figure 36 – Average temperatures for Airshower with variable heat loads

The temperature variation in this figure can be a bit misleading, since the 110 cm average values never stabilize.

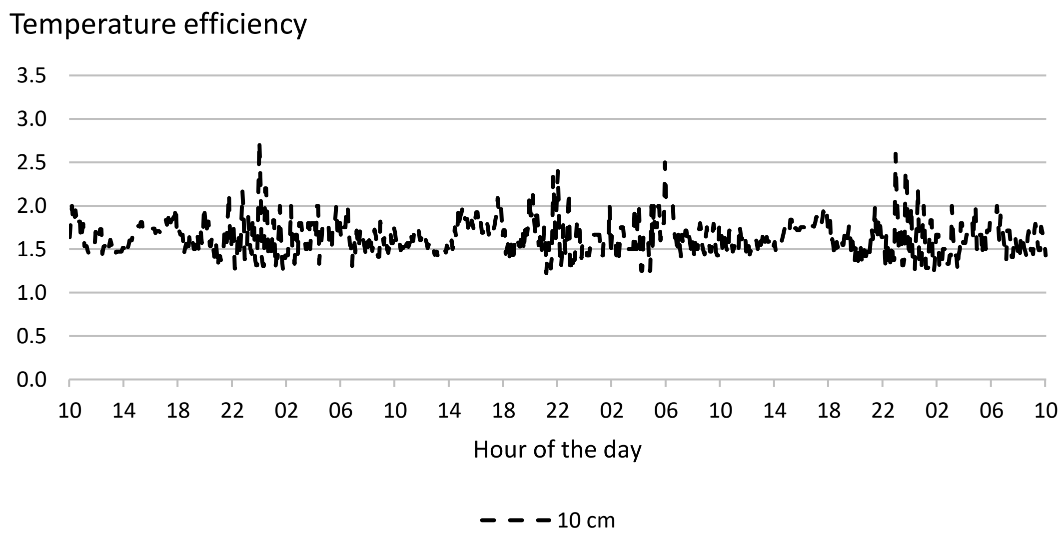

Figure 37 - Temperature efficiency for Airshower with variable heat loads closest to the working space for only 10 cm height due to errors

By calculating the temperature efficiency with *formula 17* one could see an efficiency around 1.4 – 1.8 for variable heat loads with Airson Airshower.

5.4 Displacement with floor diffuser

5.4.1 Constant heat loads

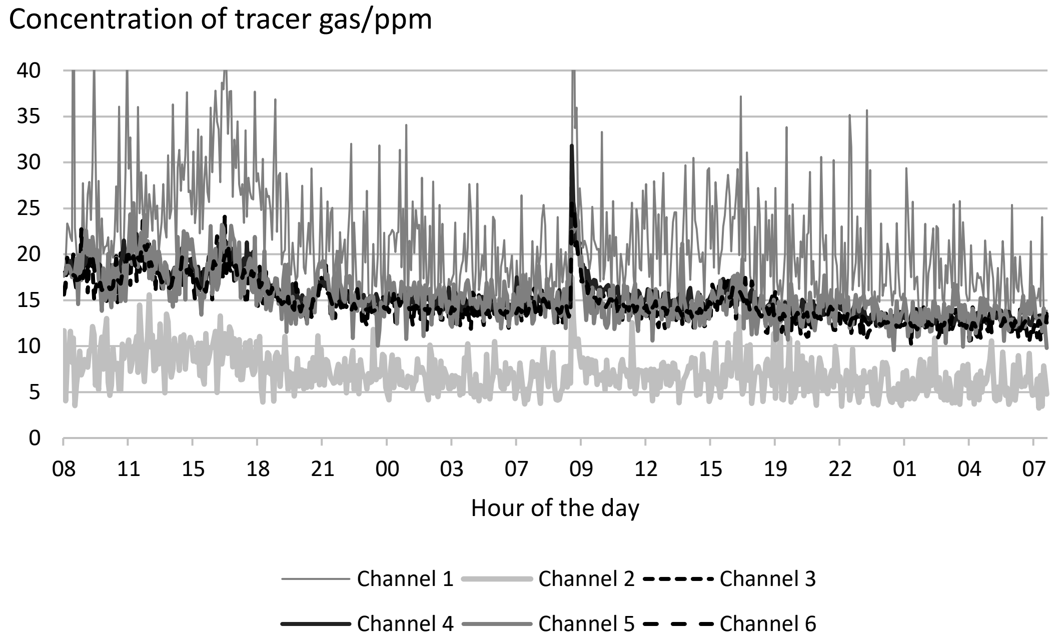

Figure 38 – Tracer gas concentration for traditional displacement with constant heat loads

Figure 38 shows the concentration thoughout the measuring period using a traditional displacement ventilation system. Channel 2 is the measurement of dinitrogen monoxide located direcly infront of the diffuser. Channel 1 that was located above channel 2 and the rest was distributed in the room. Results show all channels except 1 and 2 were constant at 10 - 20 ppm.

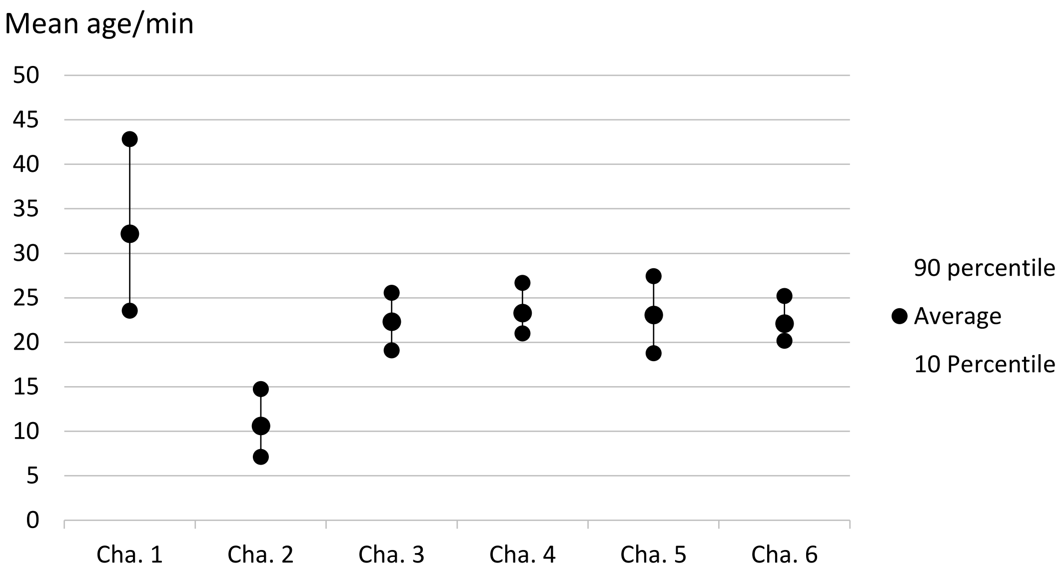

Figure 39 – Mean age distribution for traditional displacement with constant heat loads with average value, the 10th percentile and the 90th percentile

Figure 39 shows the mean age distribution of the displacement ventilation with floor diffuser. Channel 2 should be neglected since this measuring point was directly in front of the diffuser and channel 1 was as much of an accumulation zone as it is with all the other displacing system simulations.

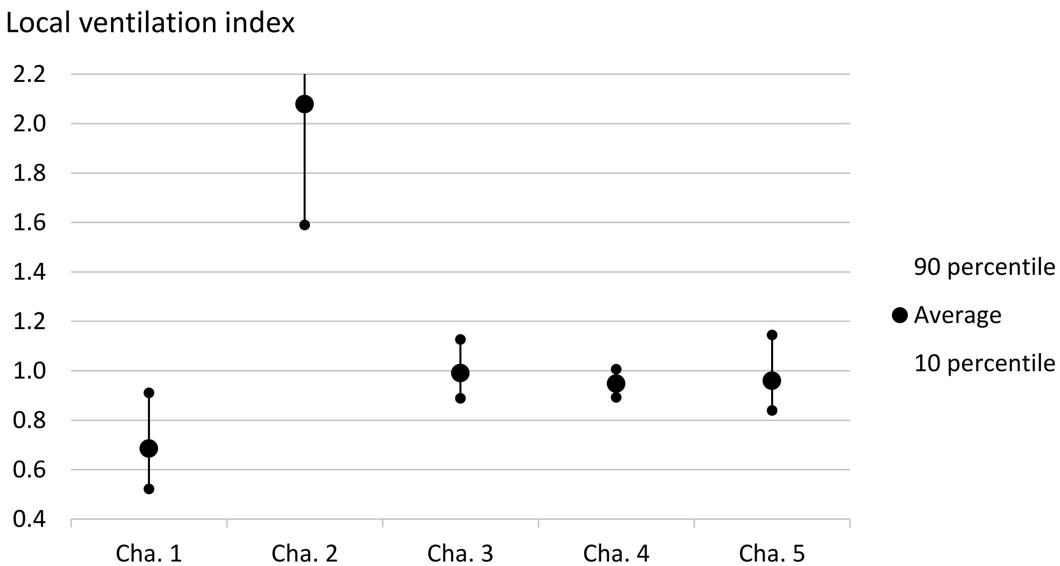

Figure 40 – Local ventilation index for traditional displacement with constant heat loads with average value, the 10th percentile and the 90th percentile

Figure 40 shows the ventilation index for the traditional displacement diffuser which was very close to the values expected when simulating a mixed air system.

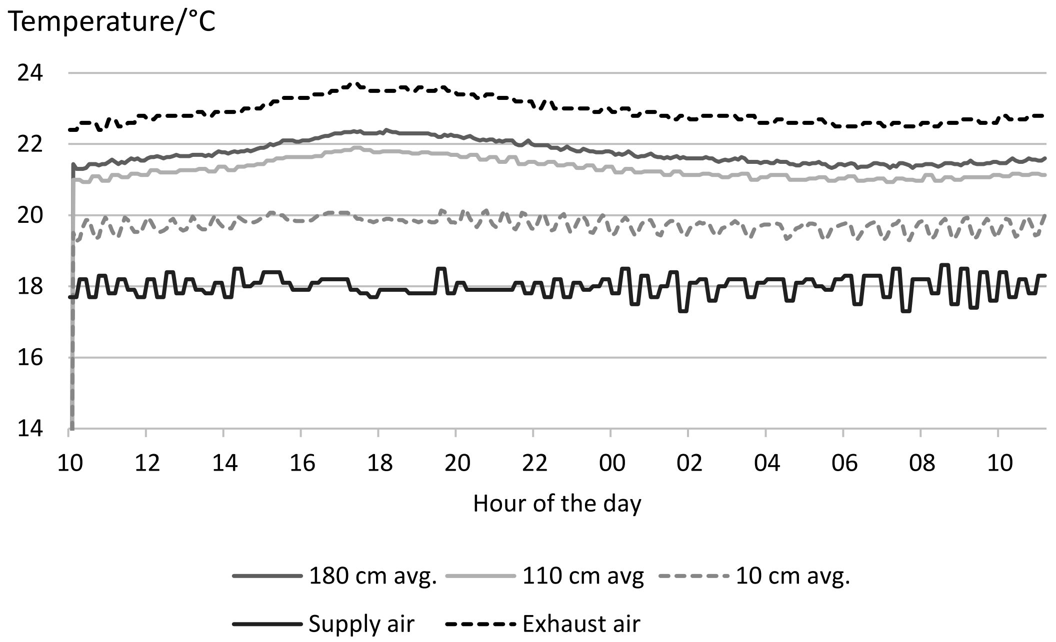

Figure 41 – Average temperatures for traditional displacement with constant heat loads

Figure 41 shows temperatures of different layers in the room during a time period of 1 day. What could be seen was that 3.5 m away from the heat source the temperature was on average 0.4 °C lower at 180 cm and 1.6 °C lower at 10 cm. The reason for such a low temperature at floor height is due the location of the supply diffuser.

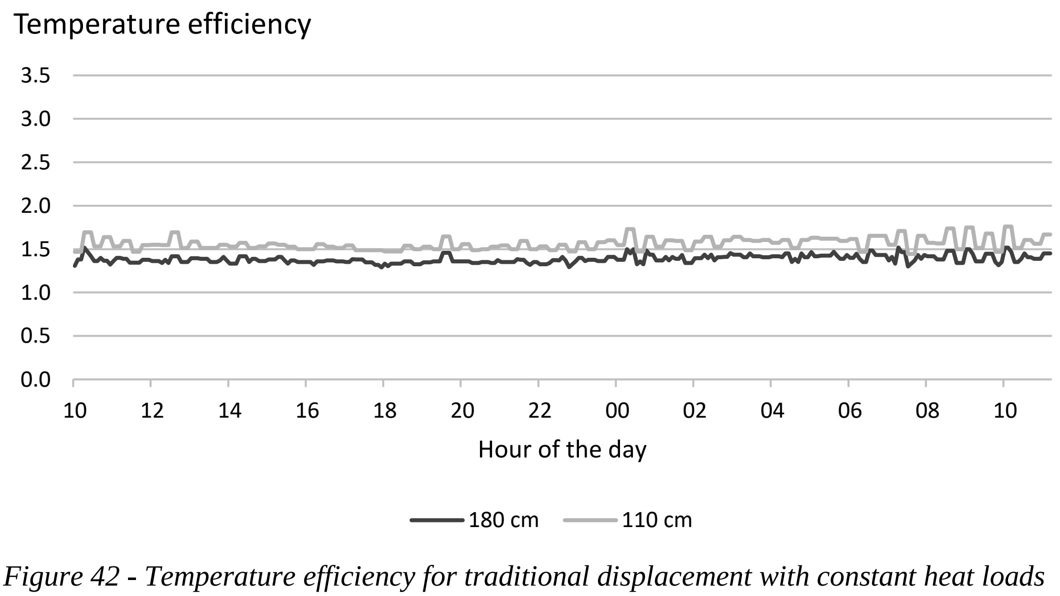

Figure 42 - Temperature efficiency for traditional displacement with constant heat loads closest to the working space for two different heights of 180 cm and 110 cm

By calculating the temperature efficiency with *formula 17* one could see an efficiency around 1.3 – 1.6 for constant heat loads with traditional displacement princple. 10 cm is not shown since the temperature efficiency was around 6 – 10 due to the high supply air flow from the floor diffuser.

5.4.2 Variable heat loads

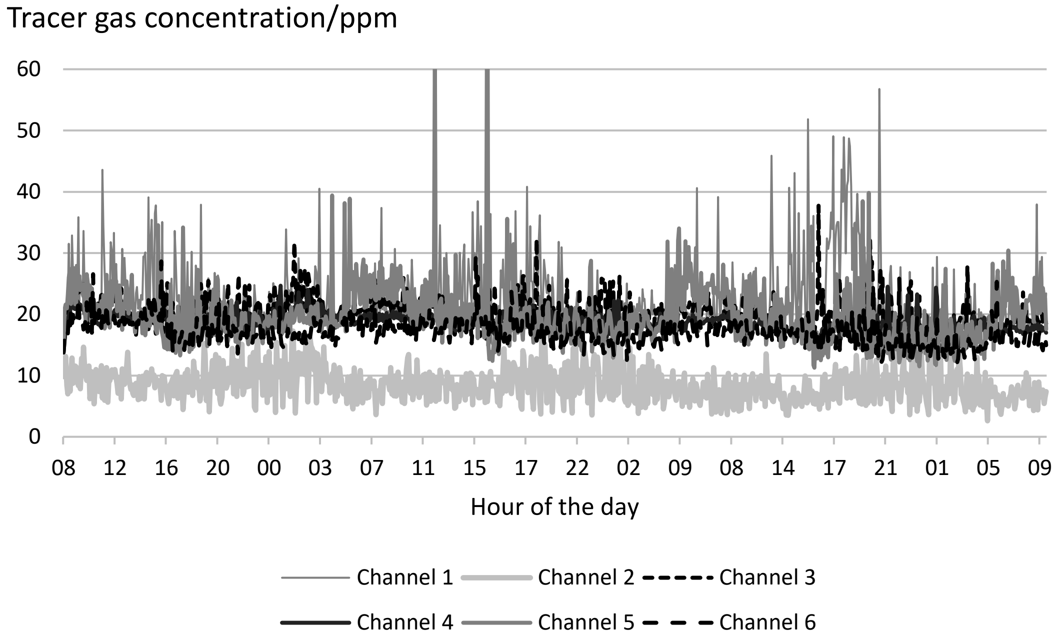

Figure 43 - Tracer gas concentrations for traditional displacement with variable heat loads

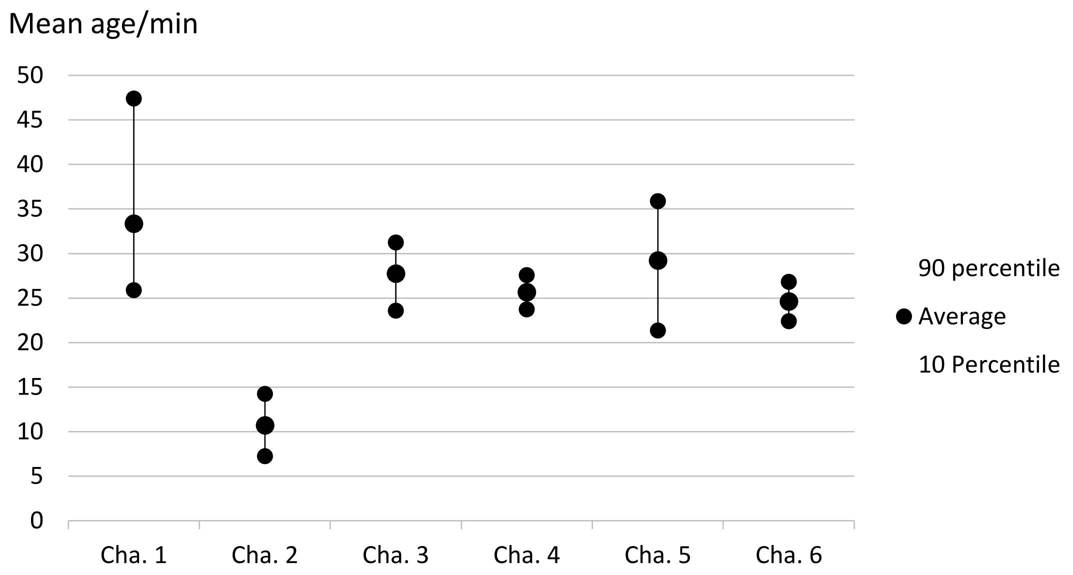

Figure 44 - Mean age distribution for traditional displacement with variable heat loads with average value, the 10th percentile and the 90th percentile

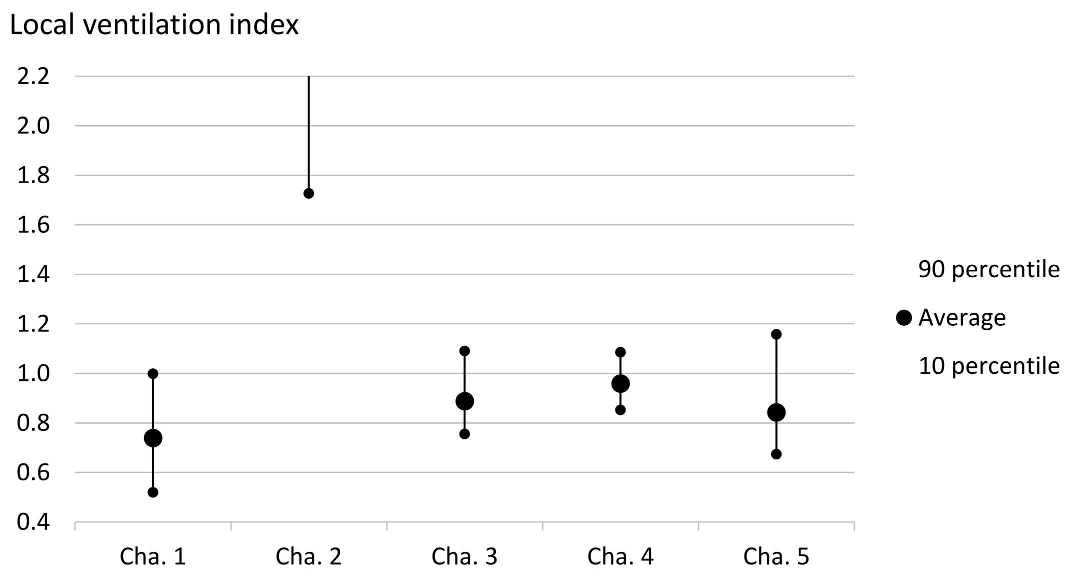

Figure 45 - Local ventilation index for traditional displacement with variable heat loads with average value, the 10th percentile and the 90th percentile. Channel 2 has a very high value for displacement which is not of use

5.5 Comparison between all three systems

To be able to compare the three systems to each other their average values have been examined.

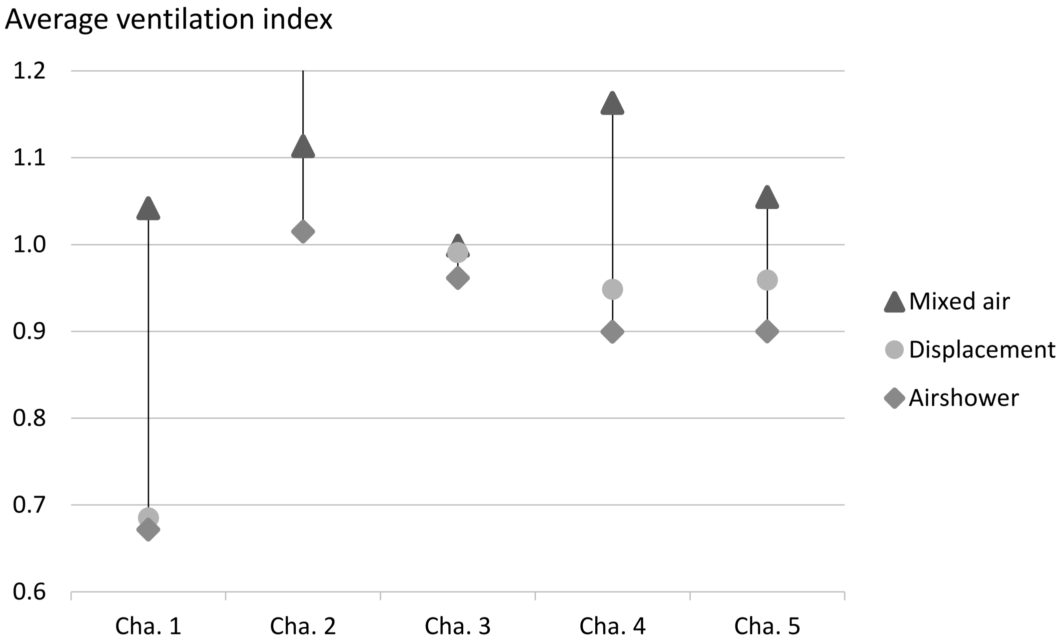

Figure 46 – Average local ventilation index with constant heat loads for all three systems. Channel 2 has a very high value for displacement which is not of use

Figure 46 presents the values in each channel for local ventilation index which is calculated from formula 9. A high percentage means that the concentration in the measured point is lower than the measured value in the exhaust diffuser. The figure shows that mixed air had highest ventilation index for all channels except in channel 2. This is because that measure point was in front of the displacement floor diffuser. It also shows that Airshower had the lowest values. This has everything to do with the exhaust air concentration which is discussed in the upcoming chapter.

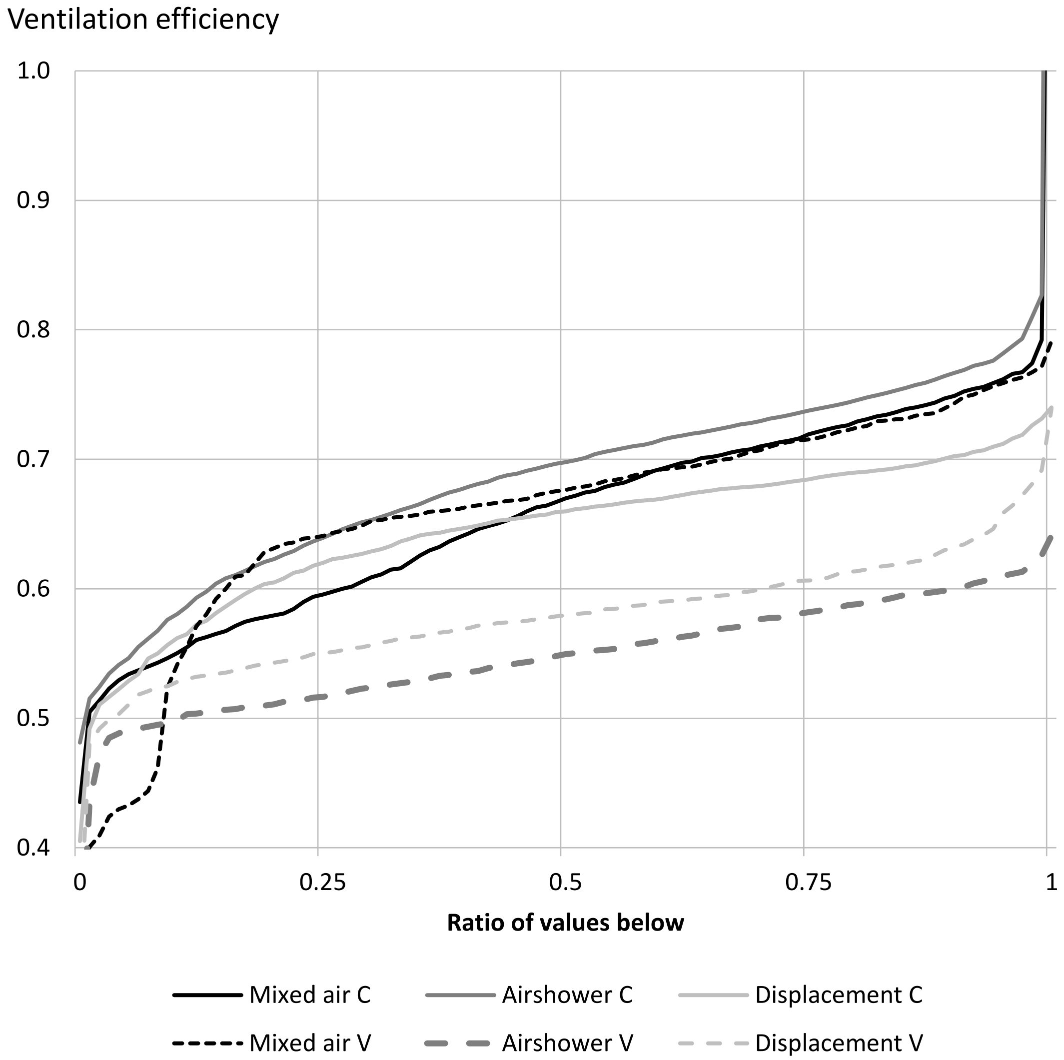

Figure 47 – Ventilation efficiency for all three systems with constant heat loads and variable heat loads. For variable heat loads all values between 8-17 are presented while for constant heat loads start and end values with increasing/decreasing gas has been removed

In Figure 47 ventilation efficency is presented for all three systems. By looking at this figure it can be seen that all systems have roughly 65 % efficiency at the 0.5 percentile which is a very low number since usual values tends to follow the table below.

Table 9. Typical values for ventilation efficiency according to Blomqvist et. al, (1983)

Ventilation system type	*Ventilation efficiency (%)*
Mixed air ventilation	≈ 100 %
Displacing ventilation	> 100 %

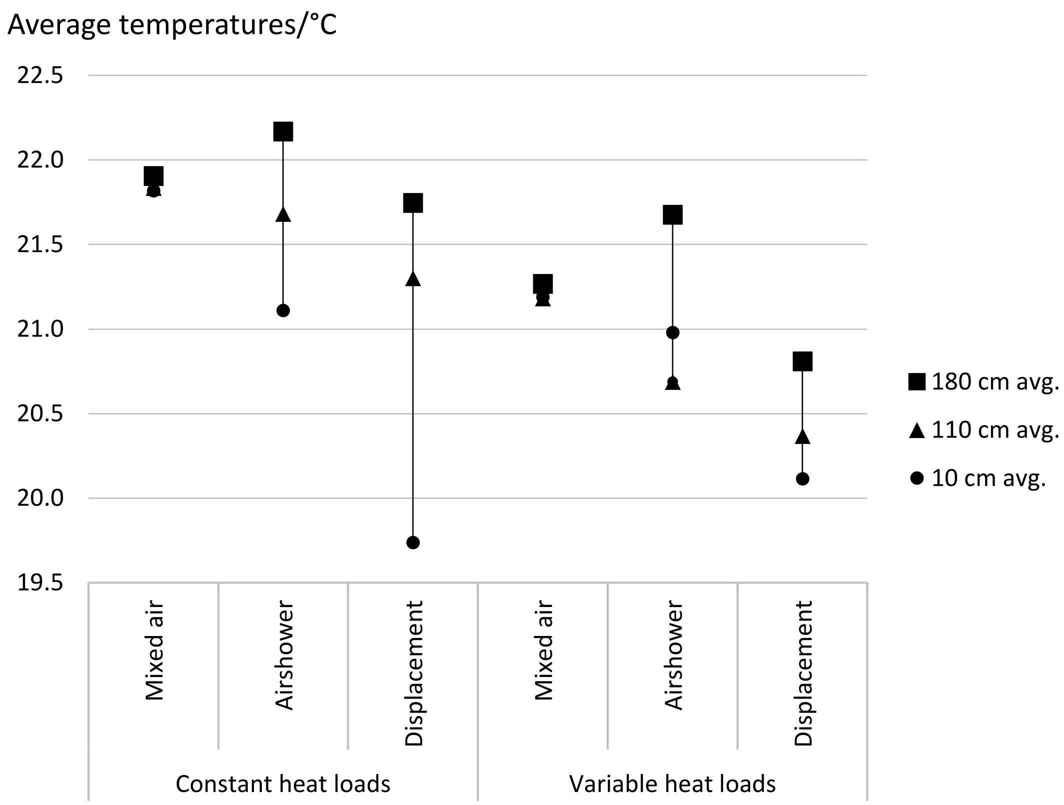

Figure 48 – Average temperature for each system on different heights of 180 cm, 110 cm and 10 cm with constant heat loads and variable heat loads

Figure 48 shows the different temperatures in each of the three air layers measured (180, 110 and 10 cm from floor level). As can be seen from the figure, the traditional displacement system was the one with the lowest temperature in both the case of constant heat load and in the case of variable heat loads. The Airson diffuser was better at keeping cold temperatures than the traditional mixed air system as long as the height being measured at was at or below 110 cm measured from the floor.

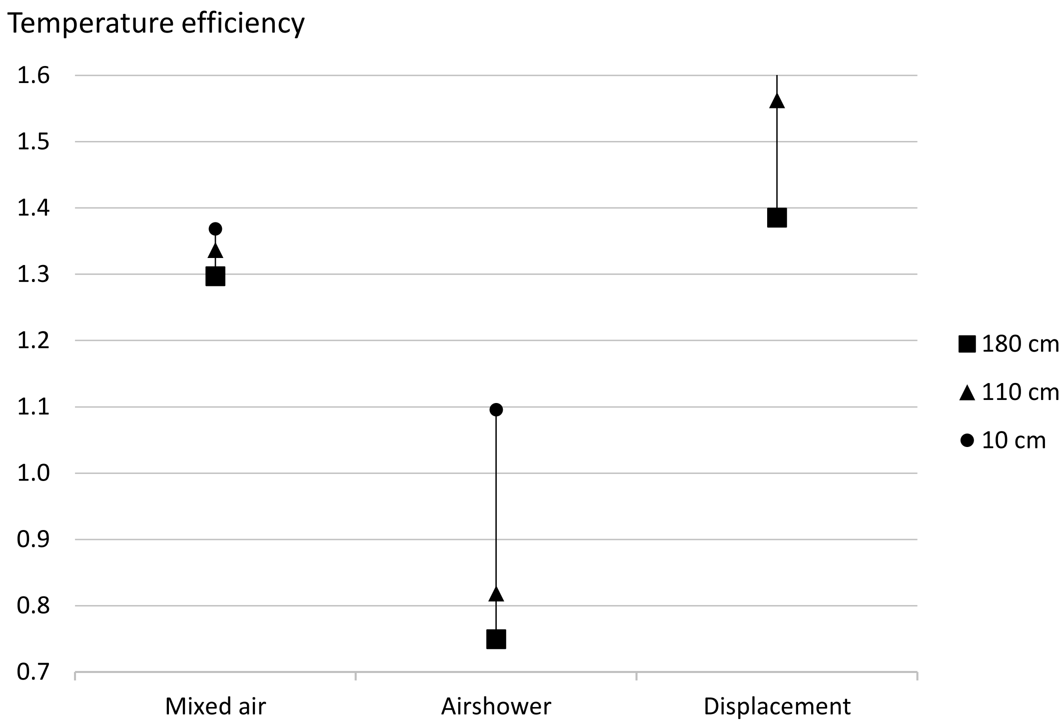

Figure 49 – Temperature efficiency for the three different systems and three different heights closest to working space. Displacement system has an average value of 6.4 at the height of 10 cm which is outside the displayed interval

Figure 49 shows a comparison between the three systems for three different air layers (180, 110 and 10 cm measured from room floor). The mixed air system had a steady temperature efficiency at around 130-138 % while the wall mounted displacement system had a much lower efficiency at around 75-83% for the habitable zone. The traditional displacement system on the other hand was preforming admirable with very high efficiencies in all points.

6 Discussion

To compare the different systems with each other, calculations have been made to determine common properties that describe how well a system is functioning. These calculations are as follows;

- Mean age of air
- Local ventilation index
- Ventilation efficiency
- Air exchange efficiency

With these four properties, a comparison is made between the three discussed systems.

6.1 Mean age of air

The mean age of air for the different systems is found individually in each subheading for the three ventilation systems.

6.1.1 Mixed ventilation

As can be seen in Figure 17 the mean age of air for the mixed ventilation varies quite little in each point which is due to the predictability ventilation method. This is not unexpected since the mixing of the air is strives to have equal concentration and temperatures throughout the room. The hypothesis for the way mixed air acts is confirmed by the results found.

6.1.2 Airshower

The mean age of air for the Airshower is very even comparing points to each other and within a point itself. This is true for all point except channel 1, which is the point located close to the diffuser in middle height of the room. The suspected reason for this is the consequence of turbulence in that area. A cool air stream meets the warm air of the room in the point measured and this could cause a loop of gas that takes longer to replace. Another issue with Figure 28 is the age of the exhaust air. This air should be the oldest air in order to get a well working system. Since the age of the air in the rest of the room is rather low, the conclusion is that some of the gas must be stuck in the area controlled by channel 1. It could also be a consequence of the gas being too heavy, accumulation in a point not measured, or leakage from the room.

6.1.3 Displacement floor diffuser

This system is very similar to the Airshower system and that becomes even more obvious when comparing the two graphs for mean age distribution. What happens in Figure 39 is in many ways a reflection of the Airshower measurements with one exception. Channel 2 is showing very low ages of air and this is due to the fact that the point is right in front of the diffuser. It is not surprising that the results are so low when considering supply air being pumped out right on the point.

6.1.4 Comparison

When comparing these three systems with each other it is hard to say that there is any significant difference. Both of the systems based on displacement ventilation show an accumulation in channel 1 but the rest of the points are very similar to every other system. This comparison is true even when the variable loads are activated.

6.2 Local ventilation index

To compare the different systems according to local ventilation index means to compare the concentration levels of gas in a specific point in the room with the concentration levels in the exhaust air diffuser. This fraction is considered good if the exhaust air diffuser shows a high amount of gas compared to the point.

6.2.1 Mixed ventilation

With mixed air ventilation the air quality in each measured point presented in Figure 18 for constant heat loads and Figure 24 for variable heat loads is over 100 % in all points. The only noticeable difference between the two loads was that with constant heat loads the distribution of local ventilation index was smaller than with variable heat loads. This may be due to higher room temperatures with variable loads since the inlet temperature was above 18 °C.

6.2.2 Airshower

Figure 29 shows the quality of the air in each point in the room in terms of ventilation index. The air in the habitat zone (Channel 3 and 5) is the air that is most interesting in the study since this more or less reflects the conditions in the zone used in a room. Since the ventilation index is a fraction between the concentration in the exhaust air and the concentration in a chosen point, it is very important to have a high concentration in the exhaust air. This is, as discussed earlier, not the case and therefore the values do not reflect the system fairly. A system with displacement ventilation should be > 100 % yet the results does not show this. The lack of contaminant in the exhaust air is the cause.

6.2.3 Displacement floor diffuser

The displacement with a floor diffuser is much the same as the Airshower diffuser with the same exception as discussed earlier with the channel 2 being in the airstream of the supply air.

6.2.4 Comparison

When comparing the three systems in terms of local ventilation index it is easy to see that the mixed ventilation principle is superior. This is unexpected since the displacement principles should provide a low concentration rate in the comfort zone and the gas ventilated away should accumulate in the ceiling and in the exhaust air diffuser, however this is not what is seen in the results. A possible explanation for this is that the comfort zone is lowered as a consequence of having an adjustable floor leaving an empty space for the cool supply air to accumulate. This will result in a sort of mixed air condition in the zone measured.

6.3 Ventilation efficiency

The ventilation efficiency for the three systems shown in Figure 47 shows which of the systems that is best at transporting away a pollutant in general. It depends on the concentration in the exhaust air being low. As a rule, the mixed ventilation should be at around 100 % in ventilation efficiency and displacing should be even higher than that. The reason for not finding this as a result is discussed in the chapter Sources of uncertainties.

6.4 Temperatures

Since the temperature measured with variable heat loads showed some errors resulting in no values at all for displacement principle when looking at ventilation efficiency only the constant heat loads were considered for this. From Figure 48 showing average temperatures it can be seen that it was warmer with constant heat loads than variable which is reasonable. Take note that there is a significant difference between the two displacement systems when looking at the temperatures for different air layers. It was harder to distinguish between the two systems when comparing ventilation efficiency and air exchange efficiency. The reason for this behaviour may be due to the air currents created by the wall mounted displacement system as the cool air is forced to mix with some of the warmer air layers on its way down toward the floor.

When comparing temperature efficiency in each of this nodes it can be seen that the wall mounted displacement system if far inferior to the other two systems even though it looked like the system would perform well if one only has temperature of the nodes in mind. The cause of this is the lower temperature in the exhaust air (in comparison to the other systems). This lower temperature is suspected to be connected with the low concentration of tracer gas in the exhaust and the possible explanations for this is assumed to be connected as well.

6.5 Sources of uncertainties

This part of the discussion will state known errors made during the study, what caused the error and proposals for correcting the mistake.

6.5.1 Adjustable floor height

The adjustable floor is likely a main reason for inaccuracies, when the laboratory was build it might have seemed like a good idea to be able to vary the floor height in order to simulate different types of room volumes. In theory this is indeed a property which makes simulations more flexible but due to the fact that the floor is not air tight some issues arises. A leaky floor will alter the actual volume of the room, which is used when calculating both the nominal time constant (formula 2) and the mean age of air (formula 5).

$$\boldsymbol{\tau_n} = \frac{\boldsymbol{V}}{\boldsymbol{q}}\ [\boldsymbol{s}] \qquad (1) \qquad \boldsymbol{\tau_p} = \frac{\boldsymbol{C_p}}{\frac{\boldsymbol{E}}{\boldsymbol{V}}}\ [\boldsymbol{s}]\ (2)$$

This will increase the mean age of air as a result however it will not affect any of the other calculations concerning ventilation efficiency or air exchange efficiency since both of these also have the nominal time constant in their equations and the volumes cancel each other out.

To prevent such a flaw of the laboratory in the future, a possible solution could be laying a plastic film over the floor and making sure it is airtight in the wall to floor connections. This was not done during the study due to the late discovery of the issue.

6.5.2 Purity of gas

The medical dinitrogen monoxide which was used for the constant heat load cases was the purest gas used in the study. When this gas ran out it had to be exchanged for a gas with lesser purity. The problem this might have caused is connected with the formula concerning the mean age of air (formula 5). Changing the purity will cause E (the amount of dinitrogen monoxide released) to be somewhat less than it is said to be. This will increase the mean age of the air slightly. This will in turn change both the ventilation efficiency and the air exchange efficiency to a lower value.

This problem can be solved by using a more expensive and purer gas, though the factor of which the result might change is very small and might not be noticeable at all.

6.5.3 Leakage

The lab is constructed in a way so that all of the measuring and control equipment is located outside of the lab itself. This is of course a must if it should be possible to check the equipment without disturbing an ongoing simulation, however it has a downside as well. The tubes and cables running from the equipment to the gas distribution, tracer gas tracking and power for heat loads must pierce the wall somewhere. These holes are an obvious source of leakage and all of these gaps are located on the lower side of the walls to enter at floor-level in the room. This might not do much to sabotage for the mixed ventilation system but for the displacement it could be a whole different story. This error in combination with the adjustable floor might further cause trouble for the displacement techniques.

A possible solution would be an air seal in the form of foam which expands to fill the gap left from the cables and tubes. It might also be an idea to protect the rest of the walls from linear leakage with a vapor barrier to be on the safe side that air doesn't travel through the isolation.

6.5.4 Measuring equipment errors

Even though the gas tracer equipment is very reliable and precise, there are parts of the system that might not be entirely accurate. This parts involve the control unit for the ventilation flow for the supply and exhaust air, the rotameter which controls the supply of tracer gas and the temperature sensors. The error which each of these parts may contribute with should be small but even so it could have some influence. A solution for this is hard to make cost effective since precise measuring equipment is expensive and the results might not change notably at all if changes are made.

6.6 Future studies

For future studies, a re-measurement of the tracer gas concentrations would be preferred but this time after handling the stated sources of uncertainties. This re-measuring should than be compared with CFD calculations with a suitable simulator for example FloVENT.

Next step could be to investigate how these systems compare in real applications during a longer timespan.

7 Conclusion

This study is not enough to conclude that any system is significantly better than any other due to the uncertainties. Both the results and the conclusion of this study could change a lot for the displacement systems if the floor and walls were to be sealed.

When measuring with constant heat loads the Airshower diffuser did prove to be the most efficient ventilation solution in comparison to the other systems but the worst one with variable loads. Due to leakages, issues with the adjustable floor and control system the results show some uncertainty.

In ventilation index terms however, the mixed air system is the only system which yields any expected results, this fraction between room- and exhaust air once again lacks confidence due to the unsealed laboratory. The displacement system would do a lot better if the mean age of air in the exhaust air would be older. The mean age of air could be seen to not differ much from system to system, it is in the interval of 20-30 minutes for all the systems with the exception for one point in the room for both of the displacement system with the probable cause of turbulence causing air currents reducing the efficiency of the system.

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