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MASTER'S THESIS

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ABSTRACT:

In Norway 40 % of the energy used is related to buildings. The fitness industry has experienced a solid growth. From 30 fitness centres in 1960 to close to 1200 in 2018. The last ten years the numbers of fitness centres have doubled and over a million Norwegians are members of a fitness centre. Fitness centres and sport halls are buildings with high metabolic activity and demands large air volumes and cooling capacity for ensuring good air quality and comfortable temperatures. The need for large air volumes and cooling capacities is again leading to a high energy consumption in these buildings. As the governing recommendation is 250 m^3/h for rooms with high activity in fitness centres, like spinning rooms, this thesis will look at the opportunity to lower the recommended air volumes and yet keep the air quality at an acceptable level. This thesis will also apply radiant cooled ceiling as an alternative to fan coils as cooling.

KEYWORDS (one per line):
Metabolism
CFD
Radiant cooled ceiling

Preface

3 years as a fulltime student and 3 years as a part time student at Oslomet – storbyuniversitetet is approaching the end of the last semester with the submission of this master thesis. The idea of the master thesis came into being with work on a rehabilitation project including a fitness centre where the recommended air volumes for spinning rooms was - in my opinion, considered conservative and that there was room for improvements when it comes to the recommended air volumes. Feedback from the users of the spinning contained complaints about draught caused by the installed fan coils which led this thesis to explore air volumes and cooling in a spinning room.

I wish to thank my supervisor professor Moon Keun Kim for helping me with methods and guiding me towards CFD simulations when the corona pandemic was preventing me from attaining measures from existing spinning rooms, and for suggesting radiant cooled ceiling as an alternative to fan coils as a cooling system. I also wish to thank my colleagues for answering all my questions I encountered during the work on this thesis. Finally, I want to thank my family. My significant other for letting me dedicate every evening and weekend (and some nights) so I could finish this thesis and my daughter for helping me keep perspective at life during frustrations and exhaustions.

Oslo, 09 Juni 2021

Terje Prestby Tørholen

Abstract

In Norway 40 % of the energy used is related to buildings. [1] The fitness industry has experienced a solid growth. From 30 fitness centers in 1960 to close to 1200 in 2018. The last ten years the numbers of fitness centers have doubled and over a million Norwegians are members of a fitness centre. [2] Fitness centres and sport halls are buildings with high metabolic activity and demands large air volumes and cooling capacity for ensuring good air quality and comfortable temperatures. The need for large air volumes and cooling capacities is again leading to a high energy consumption in these buildings. The aim of this thesis was to examine the possibilities to reduce the air volumes recommended in a spinning room and to design a room which is experienced as comfortable for the spinning athletes. As reduced air volumes leads to a decreased cooling capacity by air, this has to be compensated by an increased cooling system. The existing cooling units are fan coils, and this was compared to a radiant cooled ceiling as there are concerns about noise and draught problems caused by the fan coil units.

CFD simulations showed that draught is a major risk in the scenario with fan coils and that the placement of the fan coils supply terminal needs careful assessments to prevent the users of the spinning room to be exposed to high air velocities and draught. The results of the radiant cooled ceiling showed that control over the moisture production caused by the spinning activity in relation to the temperature of the ceiling is a necessity. The scenario with the reduced air volumes showed that the air volumes are sufficient to ensure good air quality in the room. The exception happens when the intensity of the activity is at its highest and there is a risk of elevated levels of CO_2 in the breathing zone of the athletes. The simulations also showed that the thermal stratification is maintained despite reduced air volumes and a relatively large cooled ceiling area.

There were uncertainties in the simulations which lead to a proposal of a new, further simplified simulation to see if the same results are replicable. The simulations were never validated by physical measurements and this also leads to a proposal of validating the simulations. Both proposals could be completed as either a bachelor- or master thesis.

Even though there are uncertainties about the simulations the results are considered plausible. Based on the results it is concluded that the recommended air volumes for a spinning room should be in the region of 150 m^3/h, reduced from the governing recommendation by 100 m^3/h.

Sammendrag

I Norge er 40 % av energibruken knyttet til drift av bygninger. [1] Treningsindustrien har gjennomgått en solid vekst. Fra 30 treningssentre i 1960 til tilnærmet 1200 i 2018. De siste 10 årene har antall treningssentre blir fordoblet og over en million nordmenn er medlemmer av et treningssenter. [2] Treningssentre og idrettshaller er bygninger med høye metabolske aktiviteter som krever store luftmengder og kjølekapasitet for å sikre god luftkvalitet og komfortable temperaturer. Dette igjen fører til et stort energiforbruk knyttet til disse bygningene. Målet med denne master oppgaven var å undersøke mulighetene for å senke de anbefalte luftmengdene for spinningrom og allikevel designe et rom som oppleves med god luftkvalitet og er komfortabelt for spinning atletene. Siden reduserte luftmengder leder til en redusert kjølekapasitet ved lufttilførsel, måtte dette kompenseres ved å øke kjølekapasiteten til kjølesystemet. Eksisterende kjøling er basert på fan coils, og disse ble sammenlignet med radierende kjølt himling, siden det har blitt ytret bekymringer rundt lyd- og trekkproblematikk ved bruk av fan coils.

CFD simulasjoner viste at det var stor fare for trekk i scenarioet som inkluderer fan coils og at plassering av fan coil uniten må adresseres med omhu for å unngå at brukere av spinningrommet blir utsatt for høye lufthastigheter og trekk. Resultatene av kjølt himling viste at kontroll over fuktbelastningen i rommet er viktig. Scenarioet med reduserte luftmengder viste at luftmengdene er tilstrekkelige for å opprettholde god luftkvalitet i rommet. Unntaket er når intensiteten av aktiviteten er ved maks da det vil være en risiko for økte CO_2 mengder i oppholdssonen. Simuleringen viste at den termiske sjiktingen er ivaretatt selv ved lavere luftmengder og økt areal på den kjølte himlingen.

Det ble avdekket usikkerheter ved simuleringen som fører til forslag om at det utføres en ny simulering som er ytterligere forenklet i forhold til denne for å se om de samme resultatene er replikerbare. Simuleringen ble heller aldri verifisert gjennom fysiske målinger og dette fører også til et forslag om å gjennomføre fysiske målinger for å verifisere simuleringen.

Selv om det ble oppdaget usikkerheter knyttet til simuleringen ble resultatene vurdert som sannsynlige. Basert på resultatene blir det foreslått å anbefale en luftmengde på 150 m^3/h person i spinningrom, som reduserer dagens anbefalinger med 100 m^3/h.

Contents

Nomenclature

Abbreviations:

HVAC – Heating, ventilation and air conditioning

CFD – Computational fluid dynamics

PPM – Parts per million

PMV – Predicted mean vote

PPD – Predicted percent of dissatisfied

Met – Metabolic equivalent of task

CAD – Computer assisted drawing

FCU – Fan coil unit

Mathematical symbols:

S	*Source emission rate*	mg/h
c_a	*Ambient concentration*	mg/m^3
n	*Air change per hour*	h^{-1}
V	*Volume*	m^3
K	*Decay rate*	h^{-1}
C	*Indoor concentration*	mg/m^3
Q_L	*Latent heat flow*	W
$h_{e,vap}$	*Enthalpy of Vaporization*	Kj/Kg
ϱ	*Density air*	Kg/m^3
q_{air}	*Air flow*	m^3/s
Δx	*Change in humidity ratio*	kg_{H_2O}/kg_{air}

Figures

1 Introduction

The building and building-construction sectors combined are responsible for over one-third of global final energy consumption and nearly 40 % of total direct and indirect CO_2 emissions in the world. [3] Norway is no exception and 40 % of the energy used in Norway is related to buildings. [1] The fitness industry has experienced a solid growth. From 30 fitness centres in 1960 to close to 1200 in 2018. The last ten years the numbers of fitness centres have doubled and over a million Norwegians are members of a fitness centre. [2] Fitness centres and sport halls are buildings with high metabolic activity that demand large air volumes and cooling capacity for ensuring good air quality and comfortable temperatures. The need for large air volumes and cooling capacities is again leading to a high energy consumption in these buildings.

1.1 Background

Ventilation of rooms with activity levels such as spinning rooms have high demands to the HVAC design. Despite this, there are few existing guidelines for design of these rooms. The Norwegian ministry of culture have a guidance for sport halls, named Idrettshaller - planlegging og bygging [4] which only states that the air volume for high intensity rooms such as spinning should have an air volume of 250 m^3/h per person. This value is calculated by using the formula given by Sintef Byggforsk guidance 421.503: [5]

$$q_p = 15000M \cdot \frac{T_s}{T_i(C_i - C_o)} \cdot \frac{1}{\epsilon_v} \tag{1}$$

M is the metabolic rate
q_p is the air volume
T_s is the supply air temperature in Kelvin
T_i is the room temperature in Kelvin
C_i is the indoor concentration of CO_2 in ppm
C_o is the outdoor concentration of CO_2 in ppm
ϵ_v is the ventilation efficiency
$q_p = 250 m^3/h$, provided that M=10, C_i=1000 ppm, C_o=400 ppm, $T_i = T_s$ and $\epsilon_v = 1$

Apart from this there are up to the engineer to design a system that is in accordance with the demands given by the regulations presented in the next chapter.

With work on a project relating to rehabilitating a sport hall and a fitness centre, the attention to the air volumes recommended in spinning rooms arose. The room is described in chapter 3. The size of the spinning room and the amount of people able to participate in a single spinning session is leading to large air volumes and big dimensions of the canals for the ventilation system. As the calculated air volumes for the existing spinning room leads to an air change of the room of approximately 26 h^{-1} there is a concern that the air volumes are being over dimensioned with the recommendations given by guidance 421.503 and the guidance from the ministry of culture.

1.2 Aim of the thesis

The aim of this thesis is to examine the possibilities to reduce the air volumes recommended in a spinning room and to design a room which is experienced as comfortable, regarding temperature, for the spinning athletes. As reduced air volumes leads to a decreased cooling capacity by air, this has to be compensated with increased cooling by the cooling system. The existing cooling units are fan coils, and this will be compared to a radiant cooled ceiling as there are concerns about noise and draught problems caused by the fan coil units. Simulations done by a CFD software will be conducted to create a basis for analysis and comparisons for the different scenarios.

The main goal of the thesis:

- To reduce the air volumes in the spinning room and:
    - Ensure that the spinning room has good air quality.
    - Ensure that the demands and regulations are met.
- Examine any problematic areas regarding the use of fan coils as cooling.
- Examine if a radiant cooled ceiling is applicable in a spinning room.
- Optimize the design of the spinning room in relation of the different scenarios.

2 Theory

In this chapter the governing regulations and demands will be presented along with necessary theory for understanding the thesis.

2.1 Regulations and demands.

In this chapter the relevant regulations will be presented and discussed. The goal is to highlight those demands, relevant for this thesis, that needs to be met. Byggteknisk forskrift, henceforth referred to as TEK17, [6] are the regulations on technical requirements for construction works. Chapter 13 addresses indoor climate and health.

§ 13-1 General requirements for ventilation

1. Buildings shall have ventilation that ensures satisfactory air quality through:
    a) ventilation adapted to the rooms' design, intended use, pollution and humidity loads.
    b) satisfactory air quality in the building regarding odour; and
    c) indoor air that does not contain harmful concentrations of pollutants that pose health hazards or cause irritation.
2. Buildings and buildings' ventilation systems shall be sited and designed to ensure the quality of supply air. If the quality of the outside air is unsatisfactory it shall be purified before being piped into the building to prevent health risks or the risk of fouling ventilation equipment.
3. Ventilation shall be adapted to the pollution loads from people.
4. Air shall not be piped from rooms with lower air quality requirements to rooms with higher air quality requirements.
5. Air inlets and outlets shall be designed and sited to ensure that pollution from outlets does not re-enter inlets and such that the air entering the inlet is as unpolluted as possible.
6. Circulating air shall not be used if this results in the transfer of pollutants between rooms where people are present.
7. Products for construction works shall emit low levels of or no pollution into the indoor air.

§ 13-3 Ventilation in construction works for the public and work buildings

1. An average supply of fresh air at a minimum rate of 26 m^3 per hour per person shall be supplied due to the pollution caused by people performing light activities. If activities other than light activities are to be performed, the supply of fresh air shall be adapted such that the air quality is satisfactory.
2. The minimum supply rate of fresh air due to pollution from materials, products and systems shall be:
    a) 2.5 m3 per hour per m2 of floor space when the housing unit or rooms are in use 44
    b) 0.7 m3 per hour per m2 of floor space when the housing unit or rooms are not in use.
    c) Rooms with polluting activities and processes shall have adequate extraction to maintain satisfactory air quality.

Section 13-1, paragraph 1, states that the ventilation should be adequate for the rooms use and remove odour and pollutants. Paragraph 1c states that ventilation should cover the odour from persons. Section 13-3, paragraph 1a states that for persons in light activity the minimum requirements of air per person is 26 m^3/h. This is the only performance requirements that are stated in TEK17 about air volumes and persons. For other activities the supply of fresh air should be adapted so the air quality is satisfactory. This implies that the consideration for rooms such as spinning rooms does not have any minimum requirements for air volumes except the ones stated for light activity. For rooms with other activities, TEK17 refers to NS-EN ISO 7730:2015 [7] which is supplementing metabolic rates for activities but are not mentioning activities above 3.5 Met.

TEK17 also refers to Veiledning om klima og luftkvalitet på arbeidsplassen, [8] henceforth referred to as guidance 444, by the Norwegian labour inspection authority. In this guidance ventilation is described as necessary for removing or diluting pollutants which in no other way can be avoided. Ventilation involves removing polluted air and removing it with fresh, filtered air. Since many of the pollutants is contributing to the same effects, mucosal irritation being an example, necessary air volume needs to be increased proportionally with the total pollutant. This means that the air volume cannot be decided by area or person alone to achieve satisfactory air quality. Instead, the ventilation needs to be considered by three main components tied to pollution from:

a) Personal loads
b) Building materials and interior installations.
c) Work or process.

In guidance 444, the sum of a), b) and c) is the calculated air volume. This differs from TEK17, where the largest of the sum of a) + b) and c) is the determined air flow. However, veiledning 444 states that if c) are dominant over a) + b), summation is not necessary. Guidance 444 have similar demands as TEK17 when it comes to minimum requirements. It states that the minimum accepted air volume in new buildings or buildings undergoing extensive renovation should have at least 7,0 l/s per person and 0.7 l/sm^2, to more than 2,0 l/sm^2 for material emissions. Additions for processes and activities, c), must be determined with care. In a spinning room there are two ways to interpret this. Spinning is an activity and the determination of air volume is based on a) + b) since there are no processes. Or spinning is considered a type of work and therefor the determination of air volume is based on c) alone. In either case, if the interpretation leans towards a) + b) or c), considerations must be made in regards of the air volume in the case of a) and c).

TEK17 and guidance 444 both demand that the ventilation should be adapted to a room's pollutants. TEK17 refers to guidance 444 and guidance 444 is referring further to Anbefalte faglige normer for inneklima, anbefalte faglige normer in short, by the national institute of public health. Anbefalte faglige normer for inneklima mentions a lot of maximum values for different pollutants. It is CO_2 that is of interest with its defined maximum value of 1000 ppm. This value is also mentioned in the guidance 444. Spinning is a human activity, and naturally the main source of pollution is people. CO2 production from the metabolism will be high, and the governing parameter when it comes to pollution.

§ 13-4 Thermal indoor climate

1. The thermal indoor climate in rooms intended for continuous occupancy shall be regulated in a manner that promotes health and satisfactory comfort when the rooms are used as intended.
2. In rooms for continuous occupancy it must be possible to open at least one external window or door.
3. The second paragraph does not apply to rooms in work buildings and public buildings where openable windows are undesirable considering their use.

Section 13-4, paragraph 1 is fulfilled if when the following temperatures are complied with table #:

Activity group	Light work	Medium work	Heavy work
Temperautre C	19-26	16-26	10-26

The same table is found in guidance 444. Spinning is arguably considered heavy work and might suggest that the temperature may vary from 10-26 °C. However, the usual clothing is considered light and with the heat generated from the activity the temperatures should be assessed closer in this case.

Guidance 444 have a short mentioning about draught, and states that the air velocity should not exceed 0,15 m/s in rooms with light activity. This means that rooms with higher activity, air velocity may be higher and still be acceptable.

2.2 Theory

This chapter will elaborate upon the theory behind the various aspects of this thesis.

2.2.1 Ventilation principles

The purpose of ventilation is to bring fresh air into a space. By supplying fresh air, a corresponding amount of air needs to be extracted with its pollutions. Pollutions in this case may be odours, heat, or gasses of different compounds. There are two main principles when it comes to ventilation of premises. Mixing ventilation and displacement ventilation.

Mixing ventilation:
The characteristics around this method is that the air is added with a high velocity, which means that the air must be added outside the residence zone. In practice, this means adding the air close to the ceiling. The high velocity creates a mixing process where the fresh air and the used air mixes together so the pollutions in the used air will be diluted. This requires a given air volume and distance from the occupied zone so the air velocity may decrease enough to not create any problems with draught. [9]

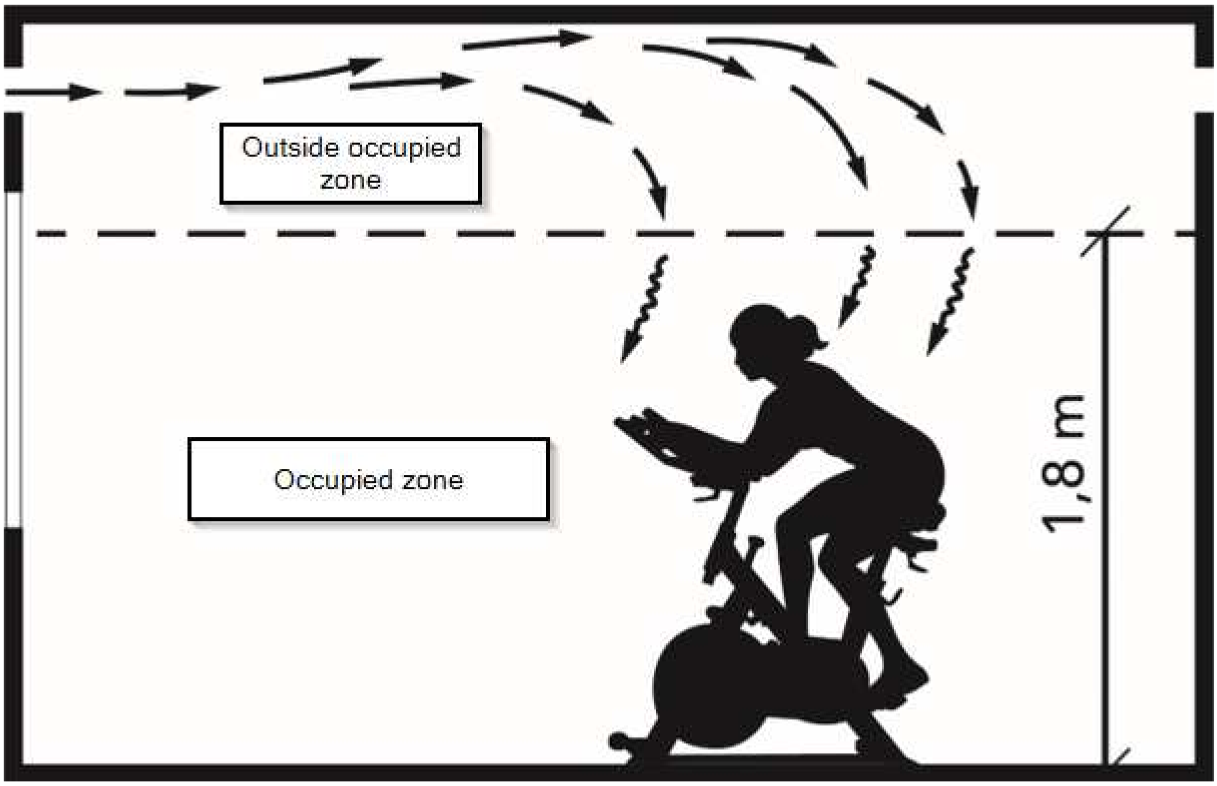


Figure 1 - Principle of mixing ventilation

Displacement ventilation
Displacement Ventilation utilizes low velocity diffusers usually located on or near the floor to introduce cooled air. This air, because of its greater density, spreads across the floor. Heat-generating objects such as people and computers warm the air causing the fresh ventilation air to rise around the object up to the ceiling where it is then removed from the space. [9]

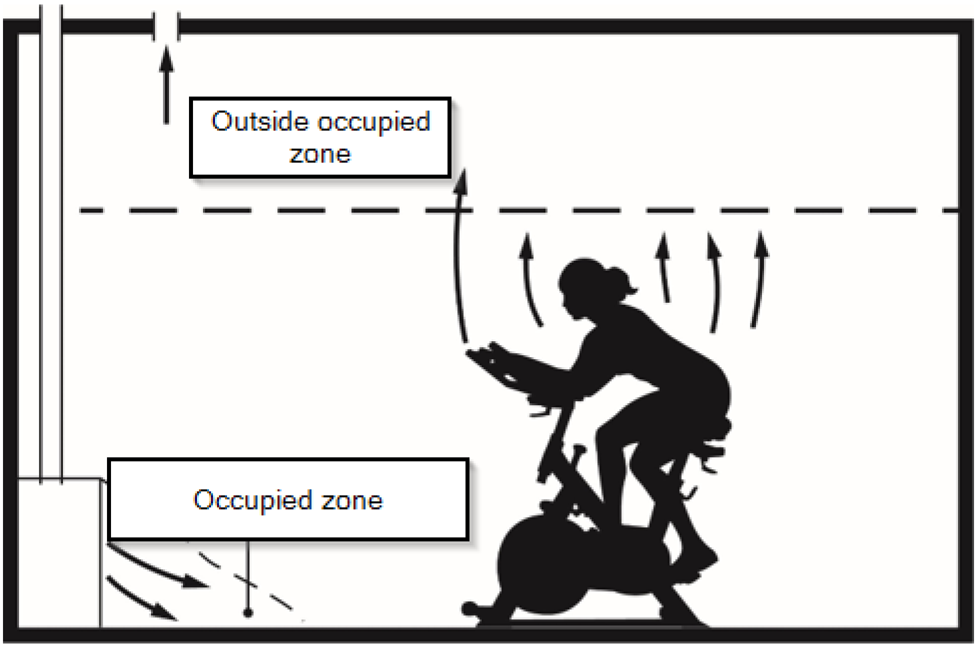


Figure 2 - Principle of displacement ventilation

Air change efficiency and ventilation efficiency
The air change efficiency is the ratio between the shortest possible air change time and the actual air change time. The air change efficiency describes how efficient the ventilated air is transported through the room before being exhausted compared to a direct transport of the air to the exhaust. In an ideal mixing system in a room, the air change efficiency is 50%. In a displacement system the air change efficiency is between 50-100%, where piston flow is considered 100%. [10]

The term ventilation efficiency is used somewhat inconsistently but are usually related to the ventilation systems ability to remove heat and/or pollutions. Ventøk describes ventilation efficiency as the relation between the exhaust air's age and the pollution's age, measured in the exhaust. The ventilation efficiency describes how efficient pollutants are being transported out of the residential zone in relation to a direct transport out of the room without any mixing. [10] The ventilation efficiency is equal to 1 (100%) at ideal mixing ventilation but may go towards infinity by direct transportation out of the room without mixing. [9]

2.2.2 Indoor climate

The indoor climate is made up of a number of measurable physical, chemical and biological factor. The World Health Organisation (WHO) has defined indoor climate as:

- Thermal environment
- Atmospheric environment
- Acoustic environment
- Actinic environment
- Mechanical environment

In Norway the terms psychosocial and aesthetic are added to the previously mentioned definition of indoor climate. These are not measurable terms and will not be assessed further as it is outside the area of interest. Mechanical and actinic environment will also not be assessed as these two terms are also outside the area of interest.

Atmospheric environment
Atmospheric environment includes the air's chemical and physical composition. It involves CO_2 and other gasses, particles, and dust among other pollutants. In this thesis, pollutants are mainly referred to as CO_2. In spinning rooms, the main source of pollution are people contaminating the air with odour, CO_2, heat and

humidity. CO_2 is often used as the governing parameter in relation to these factors. Carbon dioxide (CO_2) is created by combustion and is produced by metabolism in the human body. The gas is colourless and odourless. Anbefalte faglige normer states that the maximum value of CO_2 concentrations is 1000 ppm. 1 ppm = 1,8 $\mu g/m^3$. It is not documented that exceedance of this value is giving health issues, but elevated CO_2 values is however being followed by a sense of heavy air and bad smell. This will, to varying degrees affect a certain percentage of any population. [11] High levels of CO_2 are usually caused by deficient ventilation in relation to the total number of people in the location. As stated by guidance 444, the levels should be below 1000 ppm. [8]

Acoustic environment.
Noise is defined as unwanted sound. In a room like a spinning room there will be many noise sources exceeding the noise generated from the ventilation system, but it's important to keep the noise from the ventilation under control to ensure that the addition of the noise is negligible in comparison to other noise sources in the room.

Thermal environment
Temperature is a physical quantity that defines if something is cold or hot and is related to average kinetic energy of atoms and molecules in a system. The perception of temperature in an indoor environment situation is affected by different conditions. Air temperature and surface temperature are two temperatures who along with any draught makes it possible to calculate the operative temperature which are used in predictions regarding the perception of the thermal environment. Activity and clothing are other conditions affecting the perception. Relative humidity has little impact on the comfort feeling between 20% to 60%. Higher relative humidity will increase the production of sweat and may increase the feeling of discomfort. Lower humidity increases the heat loss through more rapid evaporation and the air is feeling colder. Draught or local cooling may appear with a combination of air velocity and temperature, or radiation to cold surfaces. If the air temperature is to low, air movement is more likely to be felt as draught. Too big vertical temperature difference may give discomfort.

2.2.3 PMV and PPD

The thermal indoor environment is often deviating from the optimal from person to person. It is therefore of interest to know how a random group of people will assess the given indoor environment and if the deviation is acceptable. For this purpose, it is introduced two standardized indexes.

PMV index
The PMV index predicts a random assessment of the thermal environment. The index is based on a seven-step scale from cold to hot.

Table 1 - PMV index

Scale	-3	-2	-1	0	+1	+2	+3
Assessment	Cold	Cool	Slight cool	Neutral	Slight warm	Warm	hot

PPD index
When the PMV index is calculated, the PPD index can be calculated or read by figure 3. The PPD index predicts the percentage of people being dissatisfied with a given thermal environment by a given activity and clothing. People have different thermal preferences, and it is not possible to obtain an indoor climate who satisfies all. The least dissatisfied people expected to reach is five percent. [12]

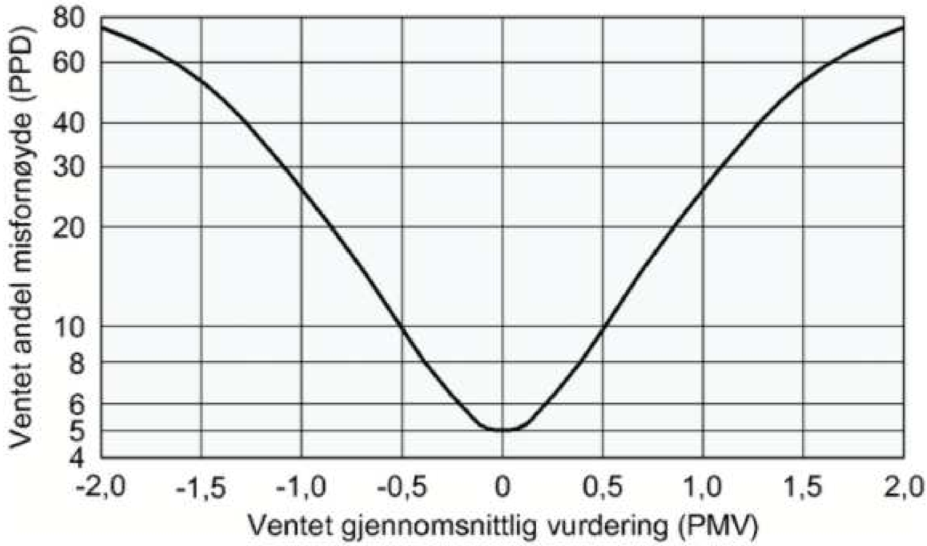


Figure 3 - PPD and PMV [5]

Vertical temperature differences and floor heating are not being represented and must be assessed separately. These two factors may create discomfort as well. [12]

2.2.4 MET

The energy that is developed in the oxidation process in the human body is called metabolism. Some of this energy is used to perform work, but most of the energy goes to the inner heat production. The energy is expressed as W/m^2. The body's surface area for an average person is 1,8 m2. For an average person seated at rest, the energy developed is 58.2 W/m^2. This is referred to as 1 Met. [13]

Air volumes and met

Calculations of air volumes based on met is done by the following formula: [5]

$$q_p = 15000M \cdot \frac{T_s}{T_i(C_i - C_o)} \cdot \frac{1}{\epsilon_v}$$

q_p is the air volume

T_s is the supply air temperature in Kelvin

T_i is the room temperature in Kelvin

C_i is the indoor concentration of CO2 in ppm

C_o is the outdoor concentration of CO2 in ppm

ϵ_v is the ventilation efficiency

The table below shows calculated air volumes at different activity levels, provided C_i=1000 ppm, C_o=400 ppm and T_i=T_s.

Table 2 - Metabolic rate and air volumes based on activity

Activity	Met	Air volume (m^3/h)/person
Seated person	1,1	26
Moderate work (business, storage, lab)	2	50
Heavy work (machine work, workshop)	3	75
Moderate exercise (sports hall, fitness-center)	6	150
Intense exercise (spinning)	10	250

2.2.5 CFD simulation

The three-dimensional unsteady form of the Navier-Stokes equations describes how the velocity, pressure, temperature, and density of a moving fluid are related. The equations are a set of coupled differential equations and could, in theory, be solved for a given flow problem by using methods from calculus. But in practice these equations are too difficult to solve analytically. High speed computers have been used to solve approximations to the equations using a variety of techniques. This area of study is called computational fluid dynamics or CFD. [14]

3 Methods and case introduction

This chapter will introduce the room of interest and the methods used to assess the different designs and scenarios presented in this thesis.

3.1 The spinning room

The room has an area of 108.5 m^2, and the dimension are 8.5 m x 12.8 m. The ceiling height is 3 m. The storey height is 3.46 m, which means that the available space above the ceiling is 460 mm. There are windows placed on the northern facade. The space is filled with exercise bicycles which are placed on different levels like an auditorium.

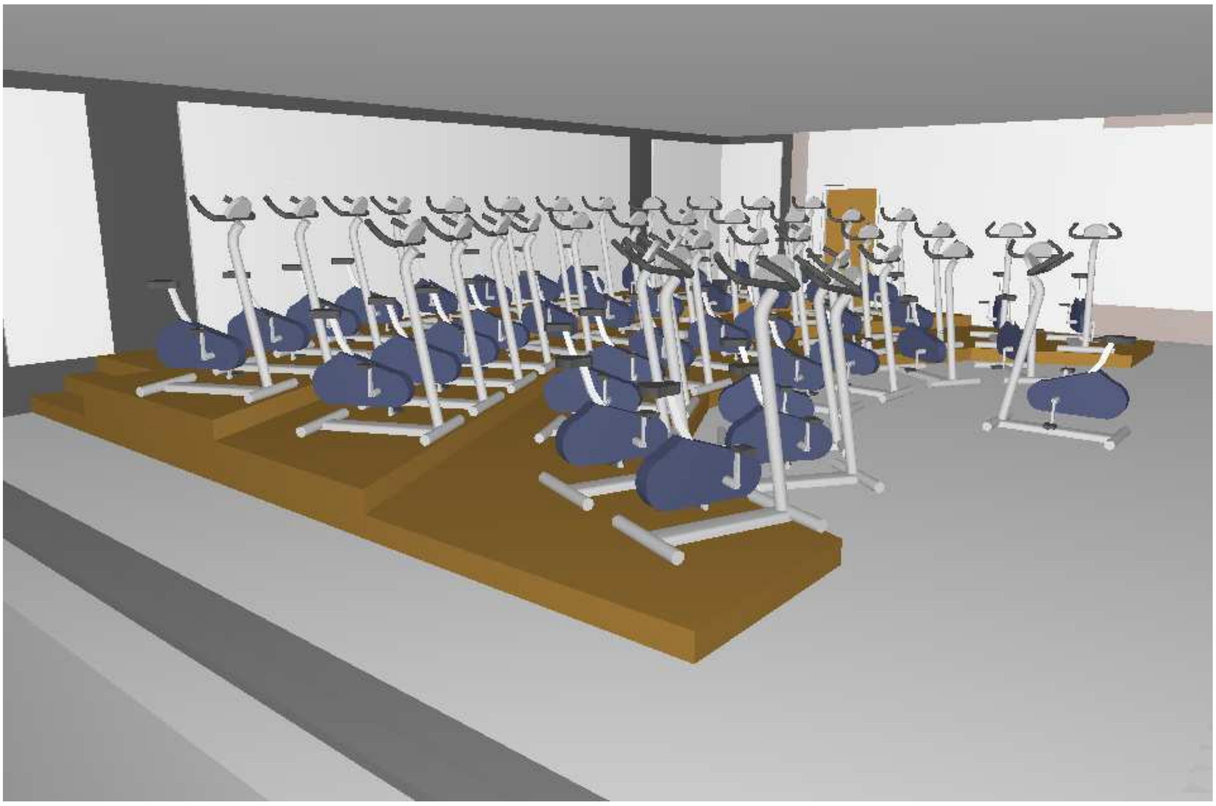
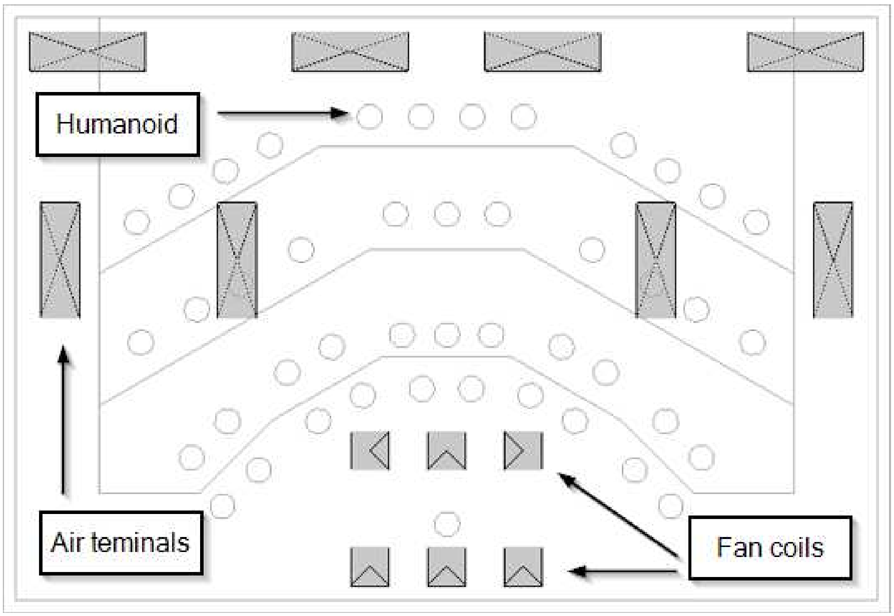


Figure 4 - Visualisation of the spinning room

The room is heated by floor heating and cooling is provided by three fan coils, mounted in the ceiling. The initial calculated power needed for the floor heating in winter conditions are 16.6 kW and the power requirement for cooling in summer conditions are 41.4 kW. The air volume is calculated to be 8 350 m^3/h. The calculations are based on a metabolic rate of 8.5 and a ventilation efficiency of 1.2. The ventilation principle is displacement ventilation and the air is distributed from air terminals in the ceiling and through the plateau on which the bicycles are placed, functioning as a large distribution chamber. The air is distributed from the chamber through wall diffusers placed on the sides and between the levels of the plateau.

3.2 Creation of the simulation model.

The CAD model was created in Revit 2020. Spinning bikes and humans were modelled as cylinders with head and mouth representing a Carbon dioxide emitter. Placement of air terminals and fan coils are according to original plan view for ventilation and the placement of humanoid objects are according to the architectural plan view. Refer to figure 5 for visualisation of the humanoid and the modelled plan view.



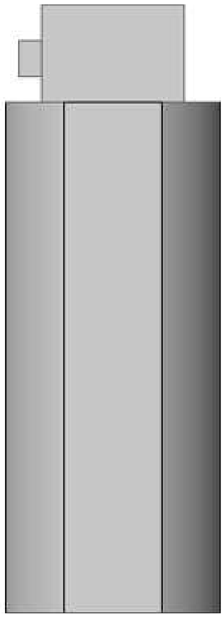


Figure 5 - Plan view and humanoid used in simulation

3.3 Simplification of the model

The model is simplified further for quicker simulations and calculations. As the room is somewhat symmetric the middle section is filtered out and remodelled somewhat to keep all of the components from the ventilations system and used for further assessing. This model is being imported into Fusion 360 for further simplification and preparation for importing to the CFD software. Creating voids and combining elements are the main operations performed in Fusion for faster deployment of boundary conditions and material characteristics. It is this model that will be used, with its alternate versions, in the different scenarios.

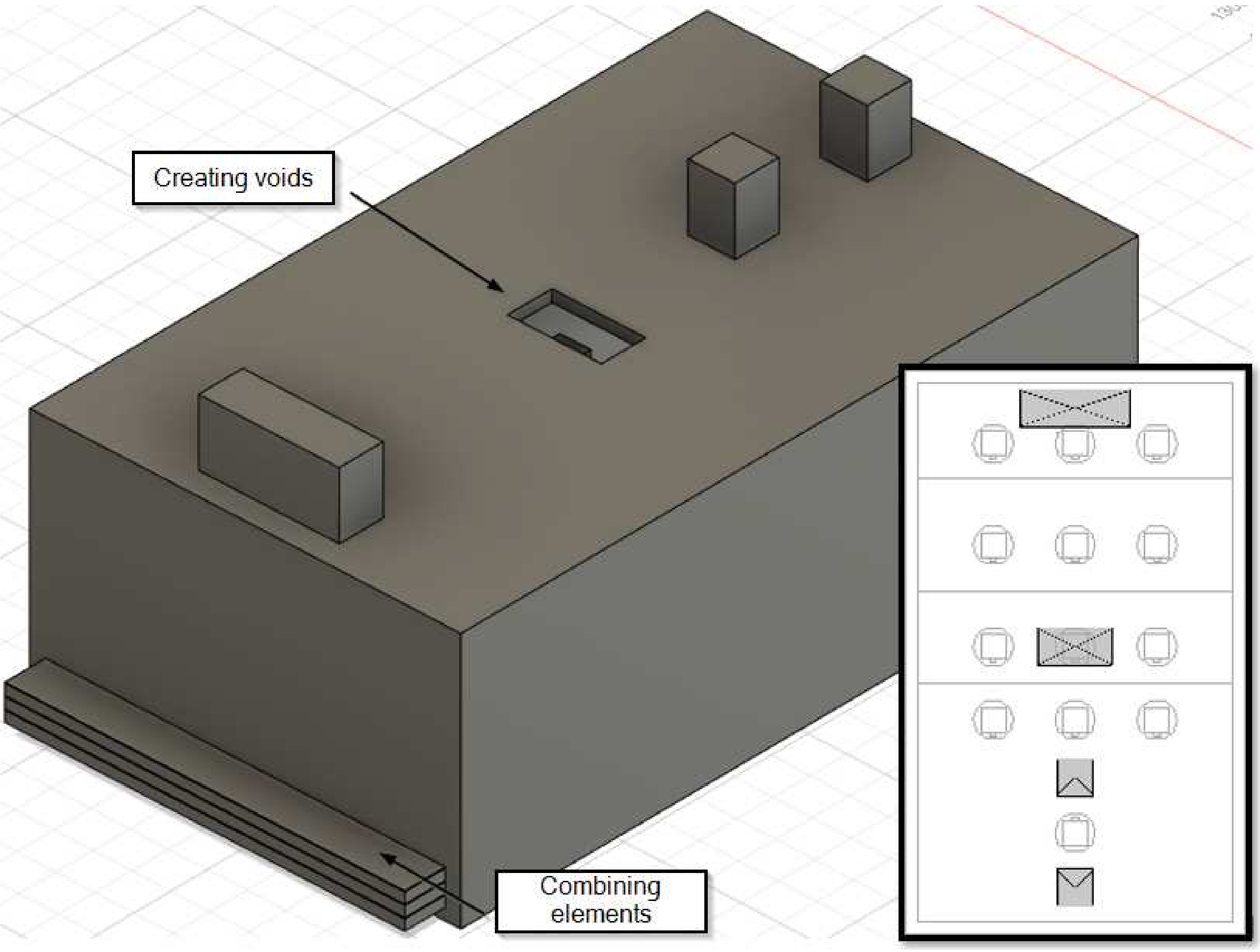


Figure 6 - Simplifications in Fusion and Revit

3.4 CFD simulations and set up

The software used for CFD simulations is Autodesk CFD. The following tables presents the boundary conditions and material characteristics used in all scenarios. Changes in either will be pointed out in relevant chapter. Reasoning behind the different inputs will be assessed in the result and discussion chapters.

Table 1 - Material assignment

Parts	Assigned material
Walls	Concrete
Floor	Concrete
Roof	Concrete
Level	Wood (Soft)
Humans	Human
Air	Air, Air properties with scalar variable, Moist air

Table 2 - Boundary conditions

Boundary	Assigned boundary conditions
Walls[1]	Varying temperatures
Walls[2]	Film coefficient (0.22 w/mK^2)
Floor	Varying temperatures
Roof	Film coefficient (0.22 W/m^{2}K)
Humans	Total heat generation (314 W)
Exhaust terminals	Total volume flow divided by number of exhaust terminals. 1 is given Pressure outlet of 0.
Supply terminals	Varying volume flows
CO2 emitter, mouth (Breath simulating)	Volume flow rate (Varying), Temperature (35 °C), Scalar (0.04), Relative humidity (1)

The turbulence model used in the simulation is the $STT - k - \omega$. The solution is set to calculate flow, heat generation and radiation. The scalar set up is based on the density of air, where the scalar value of 0 is the density of air at 18 °C and the value 0 is the density of air with a concentration of 40000 PPM at $35\ °C$. The latter representing addition of warm air filled with CO_2 from breathing. The input of 0 as scalar value for the supply air means that the addition of the outdoor airs ppm must be added to the results shown in this thesis. The reasoning for not adding a scalar value to the air was to prevent impacts on the simulation. The density is calculated by using a weighted average of the density of CO_2 and air. Each scenario was simulated with 3 cases, with variations in the breathing rate, $2\ m^3/h, 3\ m^3/h, 5\ m^3/h$, where the first represents the ventilation rates calculated by using an averaged met value, the second representing the transition between low and high activity periods during the exercise and the last representing peak activity and the breathing rate of $10\ Met$.

The mesh is distributed by using the automatic mesh sizing function given in the software. The mesh is altered under the advanced options to a lower resolution factor, 0.2, compared to the initial value of 1. The total number of nodes varies from roughly $500\ 000$ to $900\ 000$. The CO_2 emitters are suppressed from the mesh.

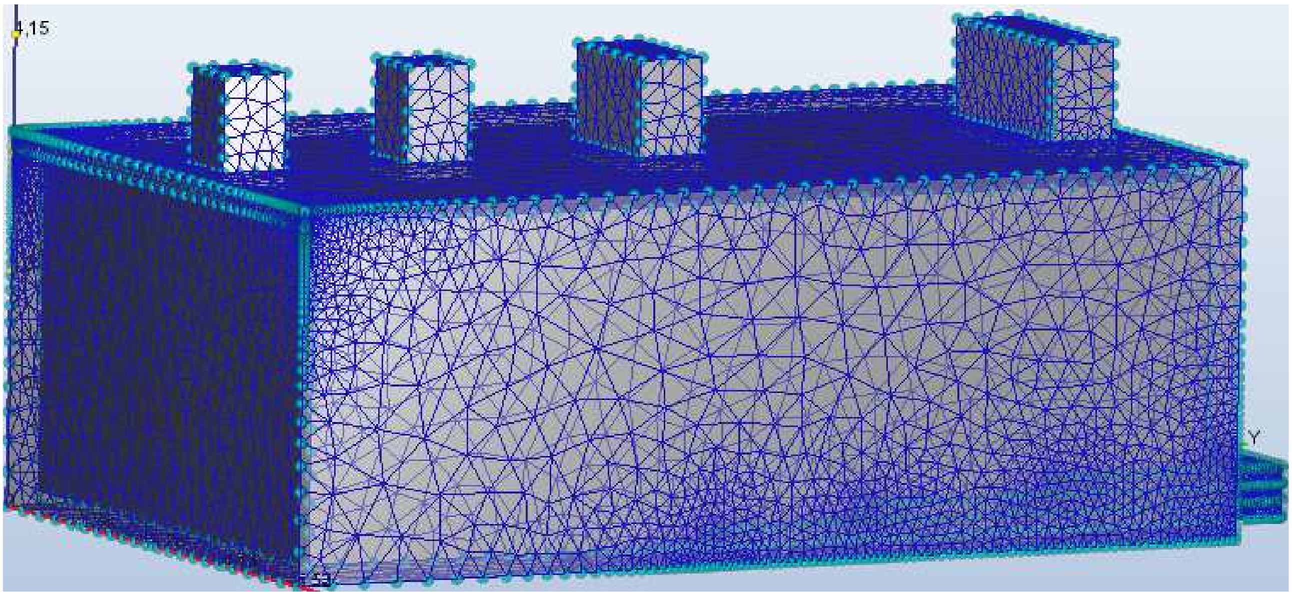


Figure 7 - Visualisation of the mesh grid

3.4.1 Scenario 1 – Fan coil units

The scenario was run as a steady-state simulation. The goal is to assess the dimensional outdoor climate data to assess the changes in the indoor climate and to look at a day warmer than the normal, as the temperature rises with the changing climate.

[1] Inner

[2] Outer

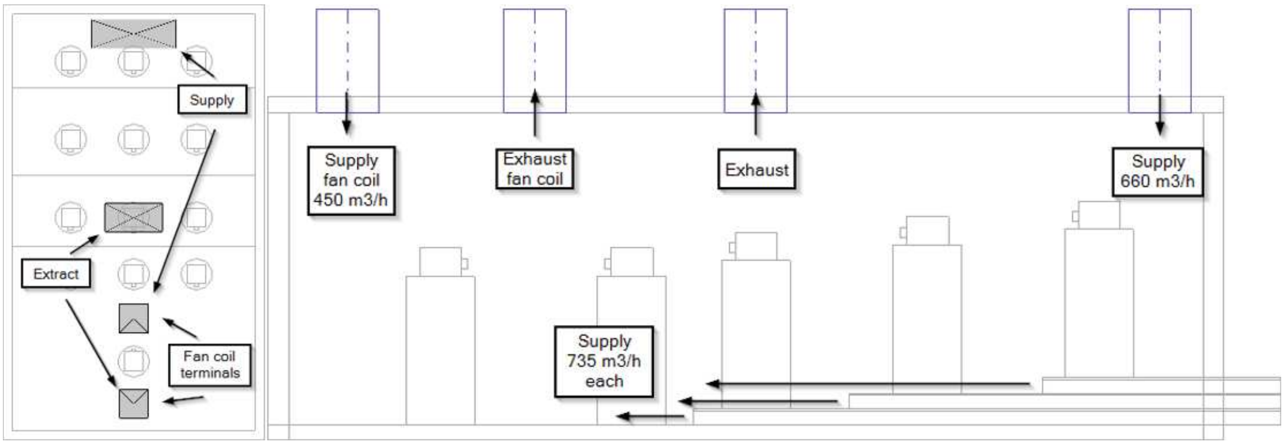


Figure 8 - Plan view and section of scenario 1

Specific boundary conditions for this scenario is:

Table 3 – Specific boundary conditions scenario 1

Supply air terminal, ceiling	Volume flow rate ($660\ m^3/h$), Temperature ($18\ °C$)
Supply levels (each)	Volume flow rate ($735\ m^3/h$), Temperature ($18\ °C$)
Supply fan coil[3]	Volume flow rate ($450\ m^3/h$), Temperature ($18\ °C$), Scalar (0.0003)
Film coefficient reference temperature	$27\ °C$ (Scenario a), $21.5\ °C$ (Scenario b)

3.4.2 Scenario 2 – Radiant cooled ceiling

As an alternative cooling method radiant panel ceiling replaces the FCU. The radiant ceiling is given a temperature of 18 °C and a surface area which yields a similar cooling power as the fan coil. All other boundary conditions are equal to scenario 1. To make room for the panel ceiling the supply terminal in the ceiling is removed and another terminal is added at the wall on top of the plateau. As removal of the fan coil liberates space from the ceiling, the ceiling is raised by $200mm$.

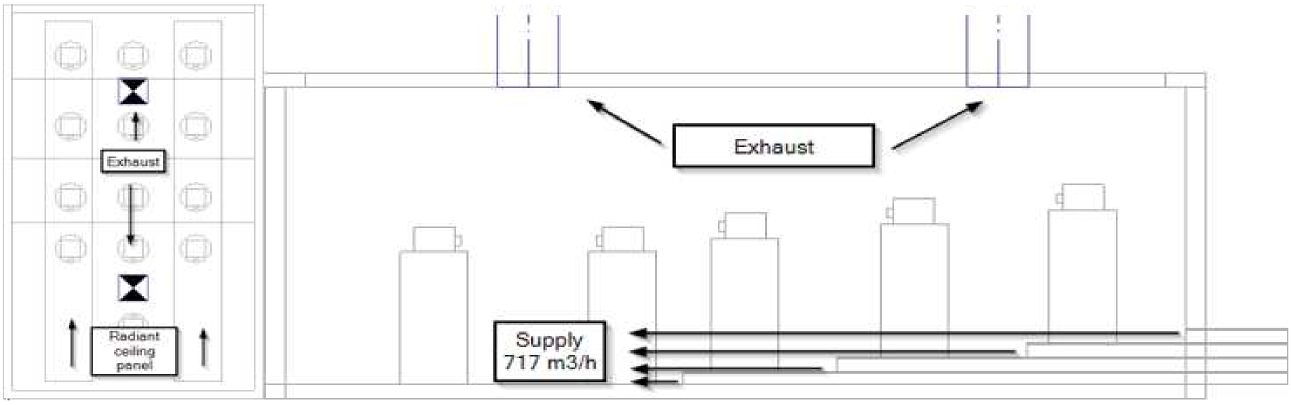


Figure 9 - Plan view and section scenario 2

[3] Supply fan coil is given a scalar value of 0.0003 for simulating return air.

Specific boundary conditions under this scenario are:

Table 3 - Specific boundary conditions for scenario 2

Supply levels (each)	Volume flow rate (717 m^3/h), Temperature ($18\ °C$)
Film coefficient reference temperature	27 °C (Scenario a), $21.5\ °C$ (Scenario b)

3.4.3 Scenario 3 – Radiant cooled ceiling and reduced air volumes

The number of steps in the previous designs have been reduced, made $50\ mm$ taller in an attempt to lower the air velocity out of the supply terminals while at the same time increasing the space above the occupied zone. As a cooled ceiling has a relative low surface temperature, considerations around relative humidity and the risk of condensation is being conducted in this simulation. The size of the cooled ceiling is also increased to account for decreased air volumes in the scenario.

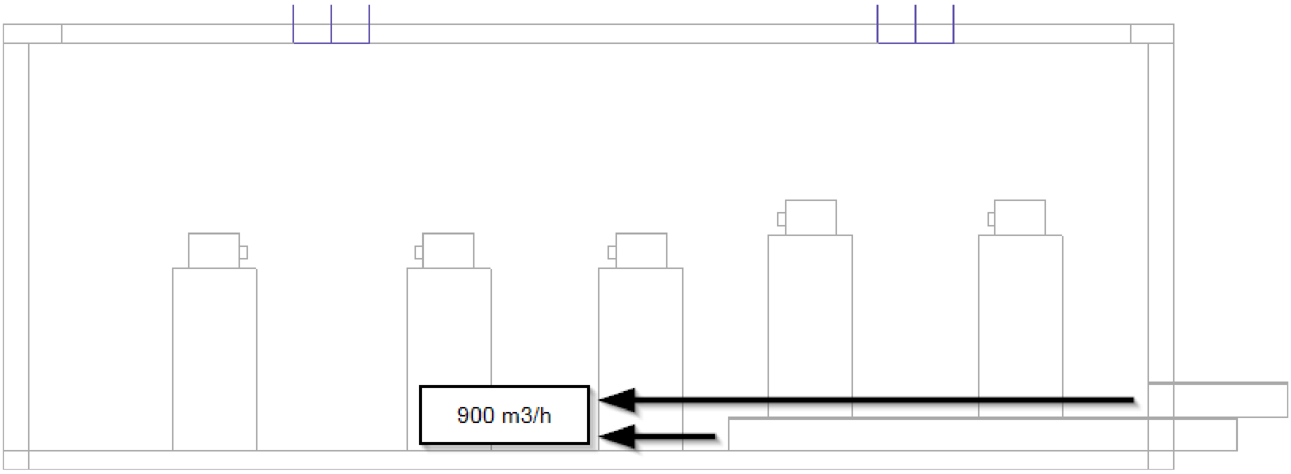


Figure 10 - Radiant cooled ceiling, optimized design section view

In this scenario the air volume is reduced and calculated based on a metabolic rate of 6.6 Met. The air flow is calculated to be $1800\ m^3/h$. The air temperature is lowered $1\ °C$ to $17\ °C$ to increase the cooling power of the air to compensate for the reduced air volume. Scenario 2 showed that the supply air needs to be condensed. The relative humidity of the supply air is therefore set to $50\ \%$. The heat generation of humans is kept the same as in previous scenarios. The heat generation is based on the average metabolic rate which is discussed in the discussion chapter. The cooling power of the supply air is calculated to be $3.06\ kW$ and the cooling power of the cooled ceiling is $1.02\ kW$. The temperature is set to $18\ °C$. A complete overview of relevant calculations is to be found in appendix A.

4 Results

The results of the different simulations will be presented in this chapter with highlights around the findings. For reference of the different planes see figure 8. The scenarios are labelled a or b which relates to the outdoor temperature, where a) is 27 °C and b) is $21.5\ °C$. As the scenarios are divided into three cases with different respiratory rates referencing the different scenario and case will take the form of scenario 1.2a or 2.3b which in these examples corresponds to scenario 1 with an outdoor temperature of $27\ °C$ and with a respiratory rate of $2\ m^3/h$ and scenario 2 with an outdoor temperature of $21.5\ °C$ and with a respiratory rate of $3\ m^3/h$.

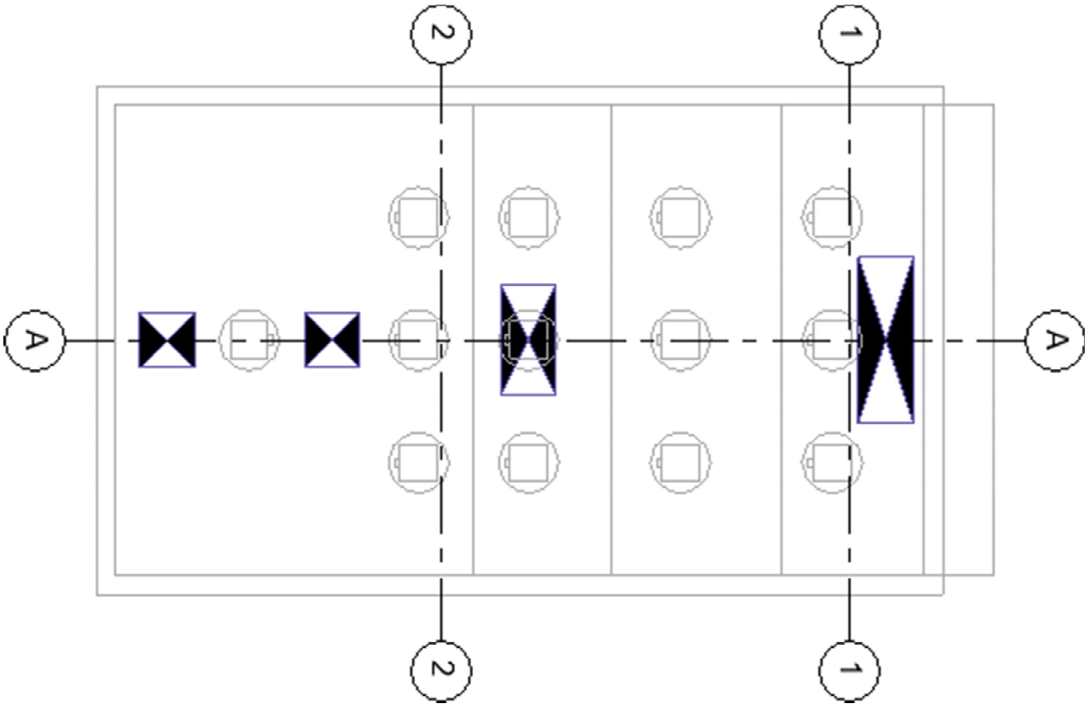


Figure 11 - Grids for reference of planes

4.1 Scenario 1 – Fan coil unit

Comparing the results between the respiratory rate in scenario 1a and 1b the flow is close to identical as seen in the figures in the following subchapter. The cool supply air is spreading along the floor and when meeting warm surfaces, the air starts to rise. The flow is nearly identical in all instances, pointing to that the differences in the breathing rates and the outdoor temperature does not affect the flow in a notable way in the occupancy zone. Looking at figure 10, it is plausible the air flow and velocity will cause a draught perceptible by people near or just below the fan coil. As the goal is to reduce the air volumes, the cooling capacity of the fan coil needs to be increased which will lead to increased air volumes supplied by the fan coil, which again, may lead to an increased discomfort caused by draught. This may lead to a state of discomfort and considerations around the fan coil unit should be made. The flow pattern in the near region of the fan coil also suggests that some of the air from the supply part of the fan coil is being drawn directly into the exhaust part, indicating a short circuit. A greater space between the supply and exhaust terminal may eliminate this. The air velocity out of the ground based supply terminals is also above the recommended velocity of $0.15\ m/s$.

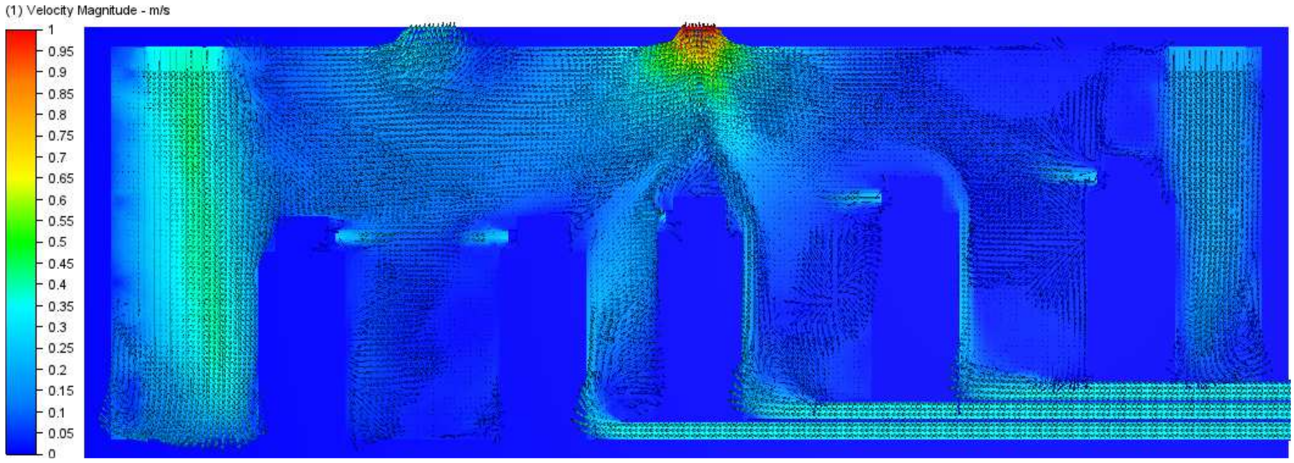


Figure 12 - Velocity, scenario 1.5b, axis A

Figure 11 shows the iso surface where the velocity is 0.2 m/s or above and shows the distribution of supply air along the floor. It also shows the velocity of the supply air from the fan coil supply terminal is above $0.2\ m/s$, further increasing the potential for discomfort for people in the near region.

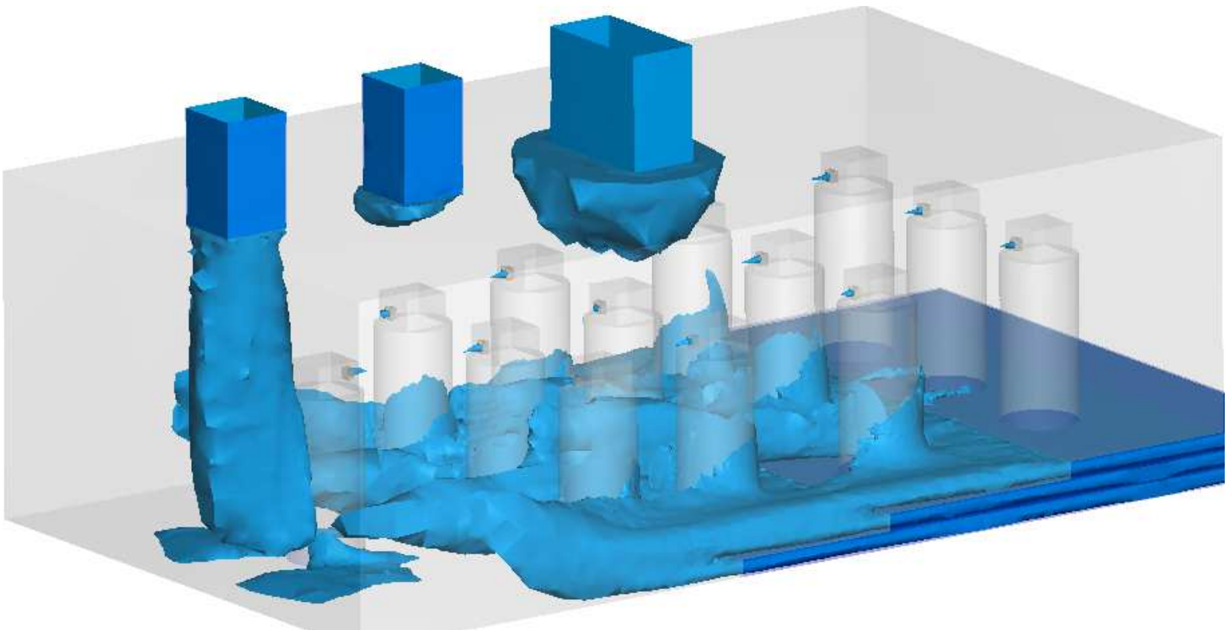


Figure 13 - Velocity, scenario 1.5b iso surface

The differences in average air temperature[4] in the room between the scenarios and cases are shown in the diagram below:

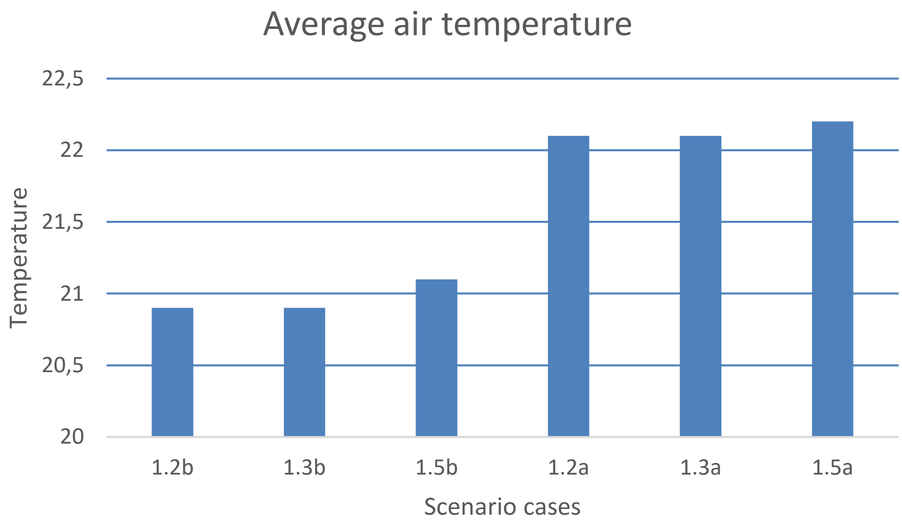


Figure 14 - Average air temperature scemario 1

An increase in breath rate affects the temperature slightly when the respiratory rate is increased to 5 m^3/h. The temperatures are around the $21\ °C$ mark at scenario b) and increasing to above $22\ °C$ in scenario a). Analysing the section views from the simulations (appendix B) reveals that the temperature is higher at the centre of the room and colder at the walls, caused by the ceiling supply terminals and the fan coil. The displacement ventilation system is also creating a stratification of temperature in the room leading to the average being an inaccurate value in this scenario. By plotting the temperature in a given point in the simulation in areas with higher temperatures, more towards the centre of the room, temperature values that are more accurate in determinate the comfort of the occupants is achieved. The temperature gradient is also attained by this. The following figures show the temperature plots:

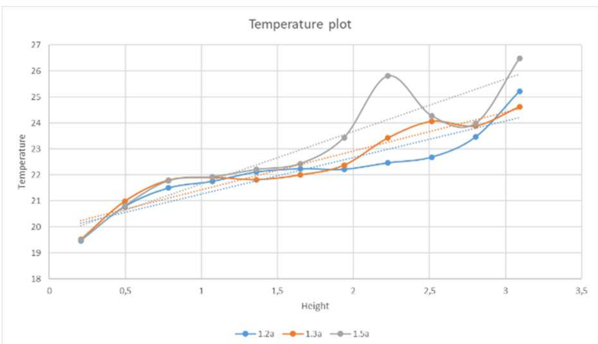


Figure 16 - Temperature plot scenario 1a

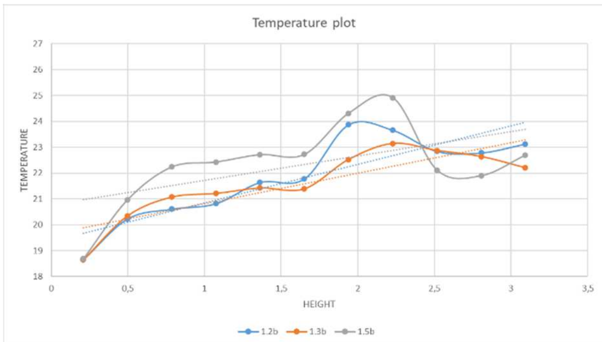


Figure 15 - Temperature plot scenario 1b

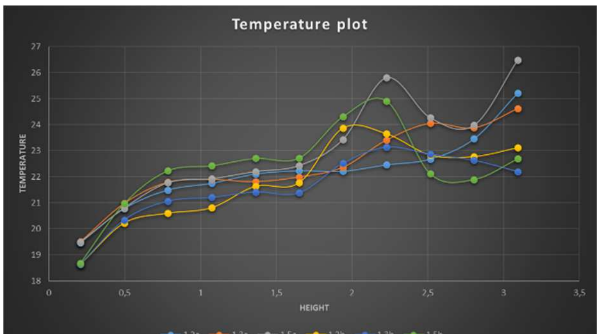


Figure 17 - Temperature plot scenario 1

[4] The average temperature of the air volume is calculated by the software.

Analysing the temperature plots, it is evident that the temperature is somewhat higher around the spinning athletes meaning that even though the air temperatures are averaging at about 21-22 °C, most of the occupants will experience air temperatures above this. Around $1.8\ m$ there is a sudden jump in the temperature which is caused by the temperature points going through the breath from the athletes. The values above $2\ m$ are therefore not accurate, however it is assumed that the temperature is stabilized and valid at heights from around $2.6\ m$. Looking at the temperature difference between $0.3\ m$ and $1.7 m$, the difference in this region is between $2.5 - 3\ °C$.

Figure 18 visualizes the distribution of CO_2.

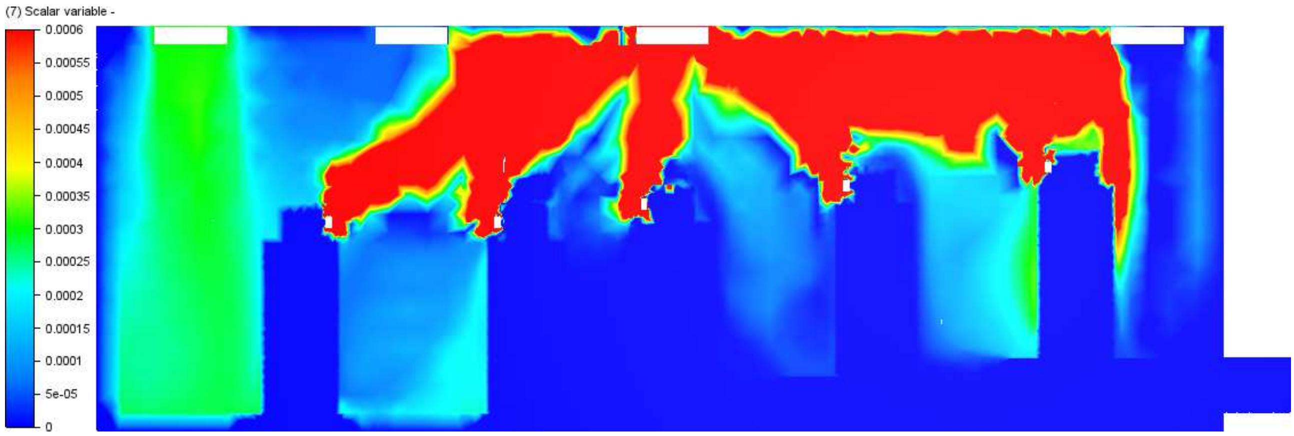


Figure 18 - CO_2 distribution, scenario 1.5b, axis A

The image is showing that CO_2 is accumulating at the ceiling. The fan coil supply air terminal is given a value of 300 ppm simulating recycling of air with this amount of CO_2 concentration. This value is only an estimation and may differ in the different cases and scenarios. The ceiling supply terminal is interacting with the stratification of CO_2 and creating a mixing effect at the terminal, leading to increased CO_2 concentrations in the occupied zones and decreasing the ventilation efficiency. The following images show the density of the accumulation of CO_2 at different heights in the different simulations.

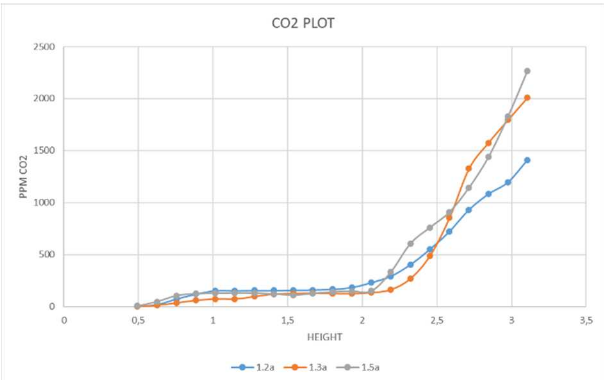


Figure 20 - CO_2 plot scenario 1a

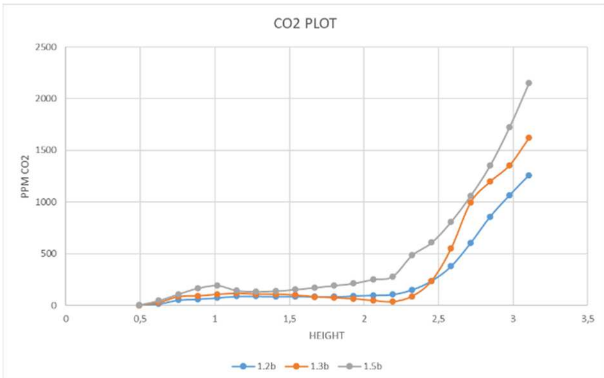


Figure 19 - CO_2 plot scenario 1b

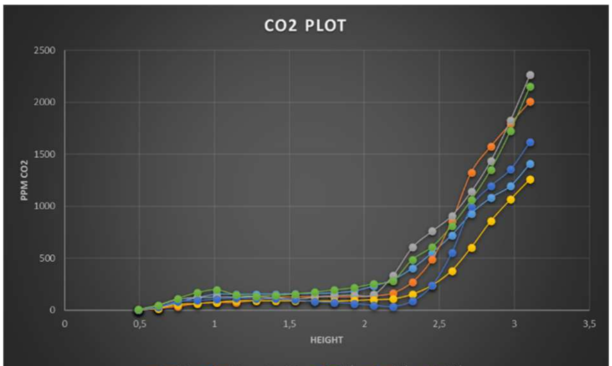


Figure 21 - CO_2 plot scenario 1

As expected, the density of the accumulation of CO_2 is increasing in relation with height. A rapid increase is happening at around 2.3 m which is slightly above the heads of the spinning athletes sitting at the top level in the room indicating the effect the respiratory has on the increase of CO_2. The lower parts of the room are not affected by the respiratory rate indicating that the displacement effect is in place. The case with $5\ m^3/h$ has as expected higher values of ppm at the ceiling compared to the other cases caused by the respiratory rate, with the other cases clustered below this. The exception is scenario 1.3a where the ppm values are at par with the $5\ m^3/h$ cases. Comparing the results in plane views (shown in subchapter) it would appear like the accumulation is denser at scenario 1.3a than in the corresponding scenario 1.3b. Slight variations in the different cases is expected, but this deviation may be caused by an error in the simulation. Another explanation may be differences of the scalar value at that point in the simulation model and thus creating a variation in the result.

The following subchapters contains images of the results that is discussed in this chapter.

4.1.1 Velocity scenario 1

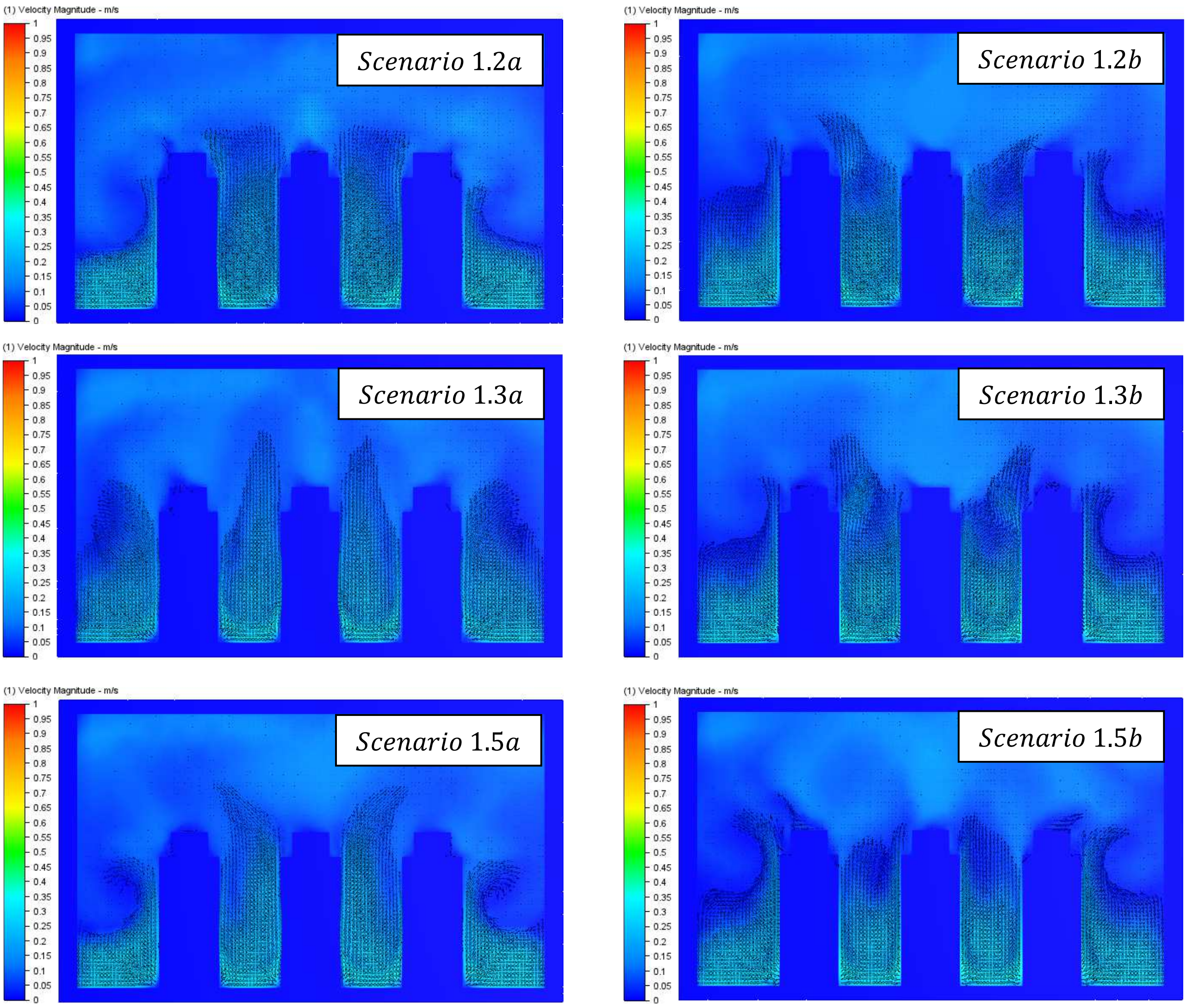


Figure 22 - Velocities scenario 1, axis 2

4.1.2 CO_2 distribution scenario 1

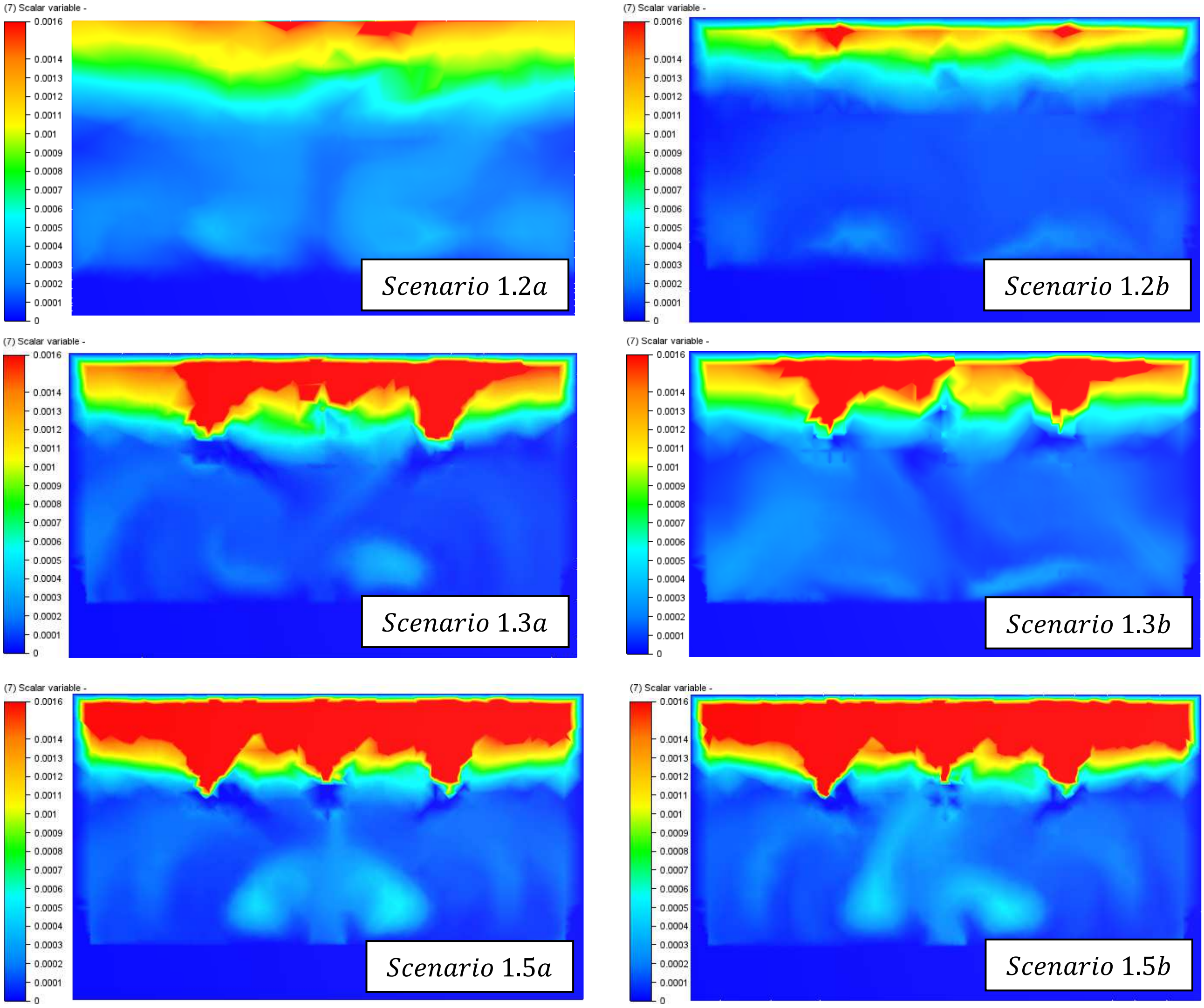


Figure 23 - CO2 distributions scenario axis 1

4.2 Scenario 2 – Radiant cooled ceiling.

As in the previous scenario there are negligible differences in the flow between case 1 and 2 meaning that the respiration rate has negligible impact of the air flow in the room. As the fan coil is substituted by a radiant cooled ceiling and the ceiling supply terminal is replaced with another supply terminal at the steps, the risk of draught from these two terminals are eliminated. Figure 24 illustrates the flow in scenario 2 for comparison to figure 12.

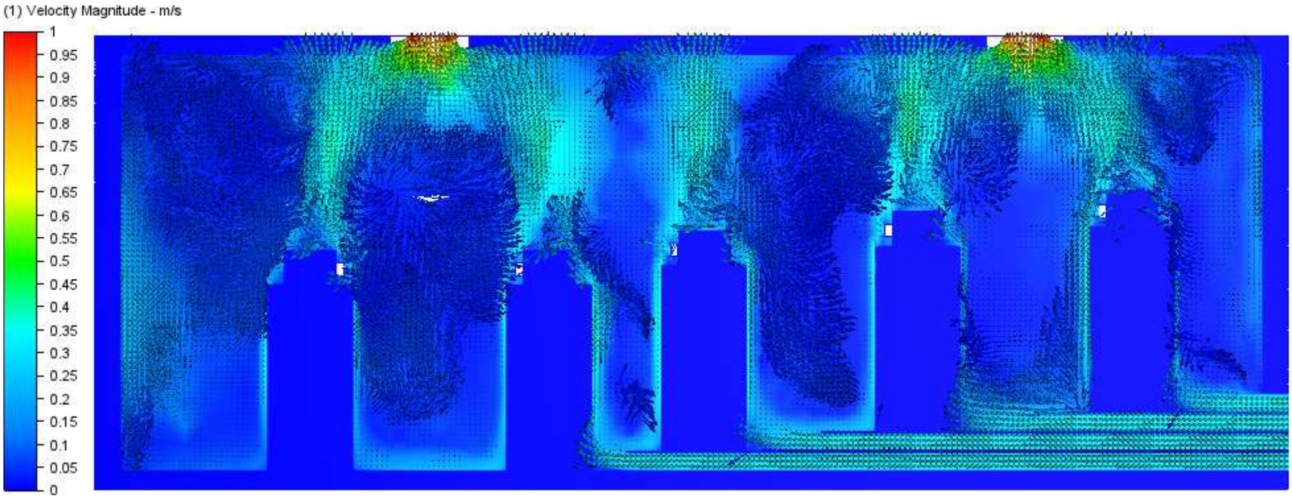


Figure 24 - Velocity scenario 2, axis A

There is still a risk, as the steps are kept at the same size, that the velocity out from the ground supply terminals may cause discomfort caused by draught. This is visualized in figure 24 showing the iso surface of the velocity at 2 m/s or higher.

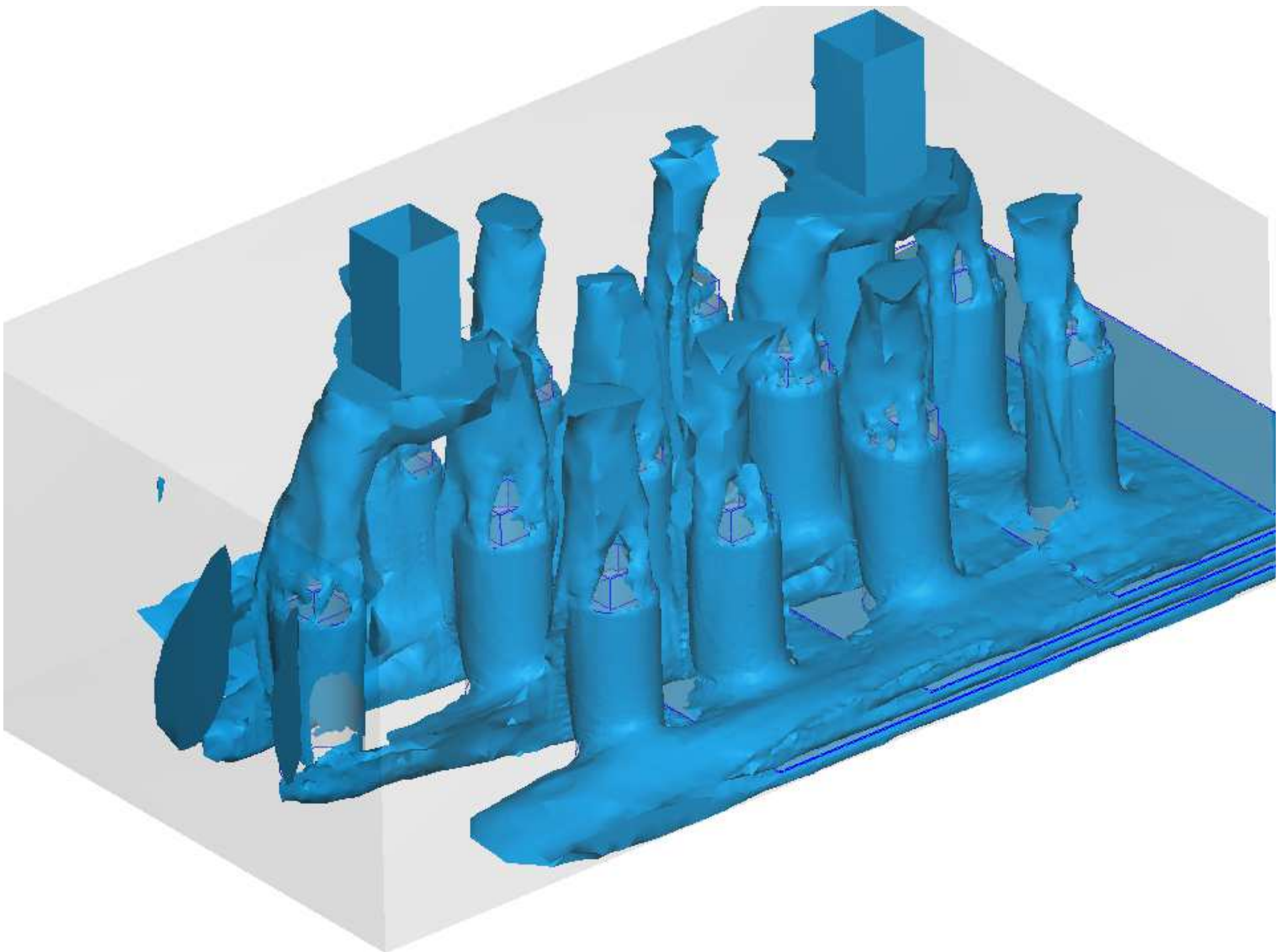


Figure 25 – Iso surface velocity scenario 2

With no fan coil or ceiling supply terminal the temperature is more evenly distributed in the room. The thermal stratification is also more obvious in this scenario for the same reason. Looking at the temperature plots along the z-axis (height) the rise in temperature is similar in all the cases, except a deviation in scenario 1.2a where there is a dip in the temperature rise at around 2.3 m. Looking at each scenario separate, the temperature gradient and temperature are in close resemblance with each other. The temperature differences between the lower and upper part of the human body is in both scenarios slightly above $2\ °C$ showing that the cooled ceiling has a lower thermal stratification than scenario 1.

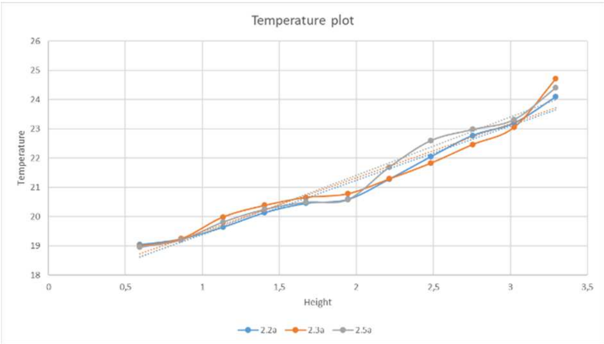


Figure 28 - Temperature plot scenario 2a

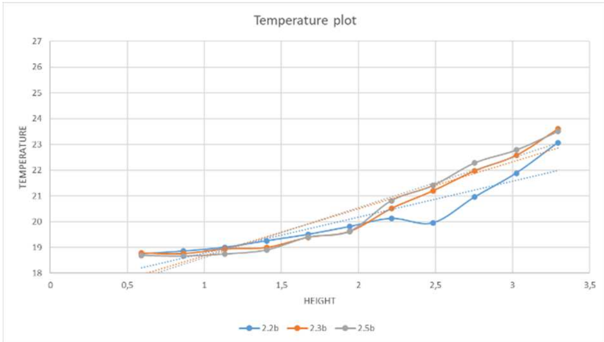


Figure 26 - Temperature plot scenario 2b

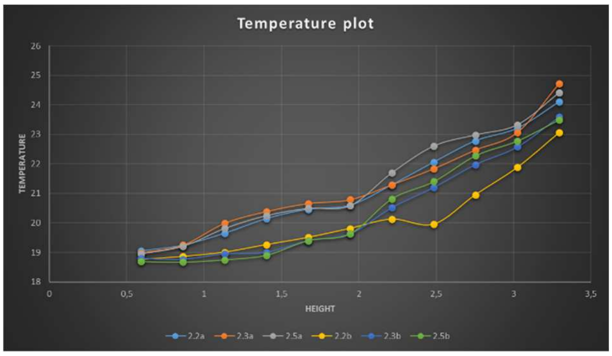


Figure 27 - Temperature plot scenario 2

The mean temperature is very similar in all cases, where scenario a) exhibits a higher average than scenario b), as expected cause by the different outdoor temperature. Comparing the average temperature with the temperatures given in the temperature plot, it is revealed that using average temperature as an indication of the temperature in the occupied zone is a more accurate approximation than in scenario 1 as the temperatures are in closer resemblance.

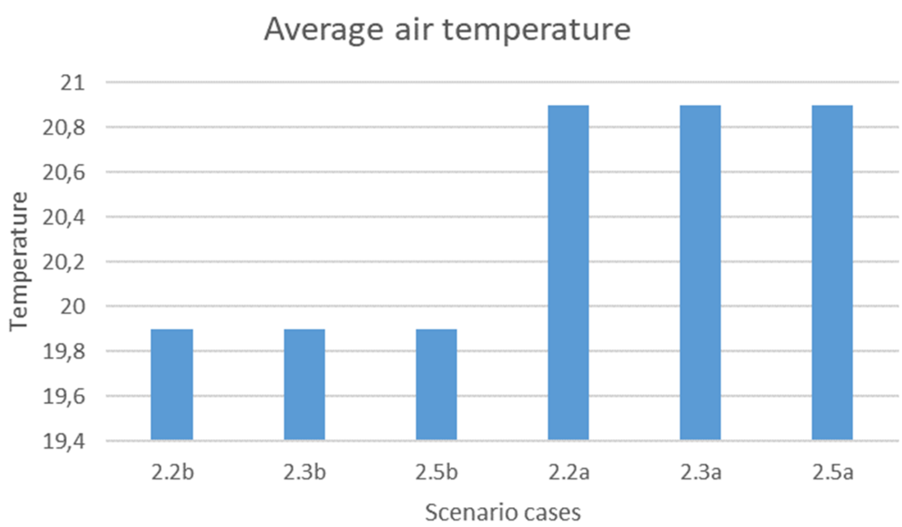


Figure 29 - Average temperatures scenario 2

The CO_2 distribution is in scenario 2a and scenario 2b developing as predicted and in the same pattern as scenario 1, where the CO_2 is accumulated at the ceiling, and air of better quality is found in the occupied zone. Analysing the graphs in the images below, (showcasing the CO_2 concentrations along the z-axis in a given point), is also confirming that the air quality in the occupied zone is satisfactory. There are lower concentrations in scenario 2 than 1 caused by the removal of the fan coil and its recycling of air. The density of the accumulation is increasing with the respiratory rate as seen in the images found in the following subchapter.

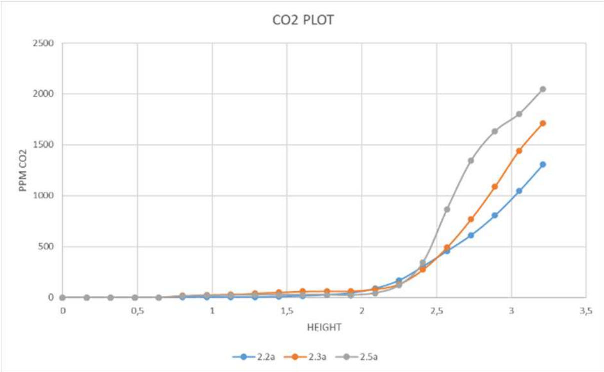


Figure 31 - CO_2 plot scenario 2a

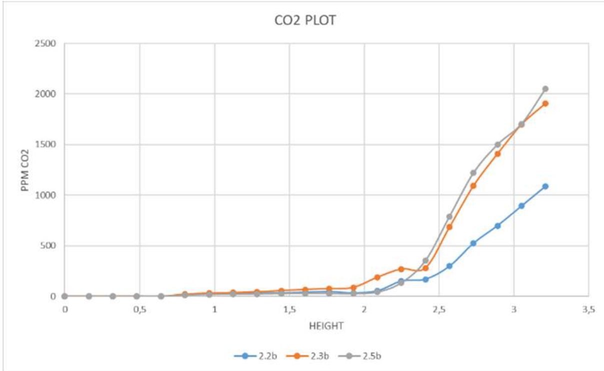


Figure 30 - CO_2 plot scenario 2b

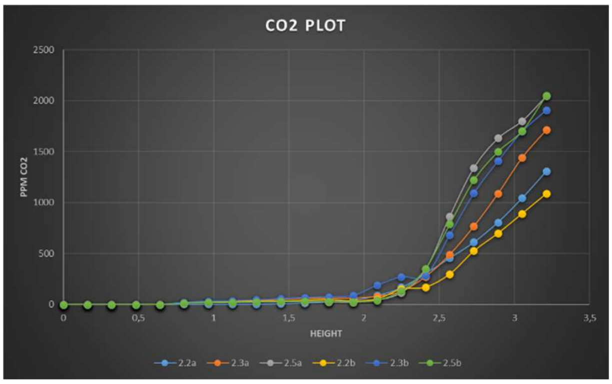


Figure 32 - CO_2 plot scenario 2

As there are cold surfaces in this scenario, it is important to look at the relative humidity to see if there is any danger of condensation. As seen in the images the humidity of the air is at 100 % when approaching the cold surfaces of the cooled ceiling and the air will start to condense when the air temperature is decreased. Taking into account that the accumulation of moisture (caused by sweating occupants) is not included in the simulations, there must be assessments made around the relative humidity of the supply air.

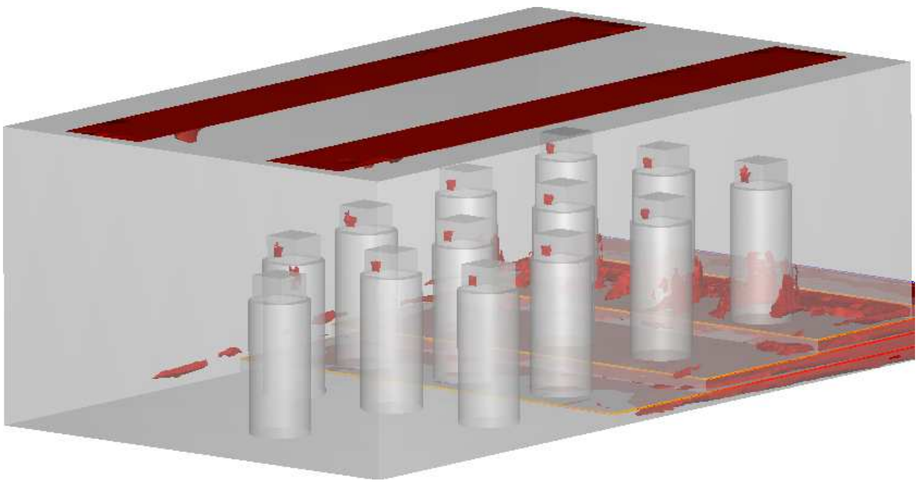


Figure 33 - Iso surface relative humidity

The following subchapters contains the images of the result that is discussed.

4.2.1 Velocity scenario 2

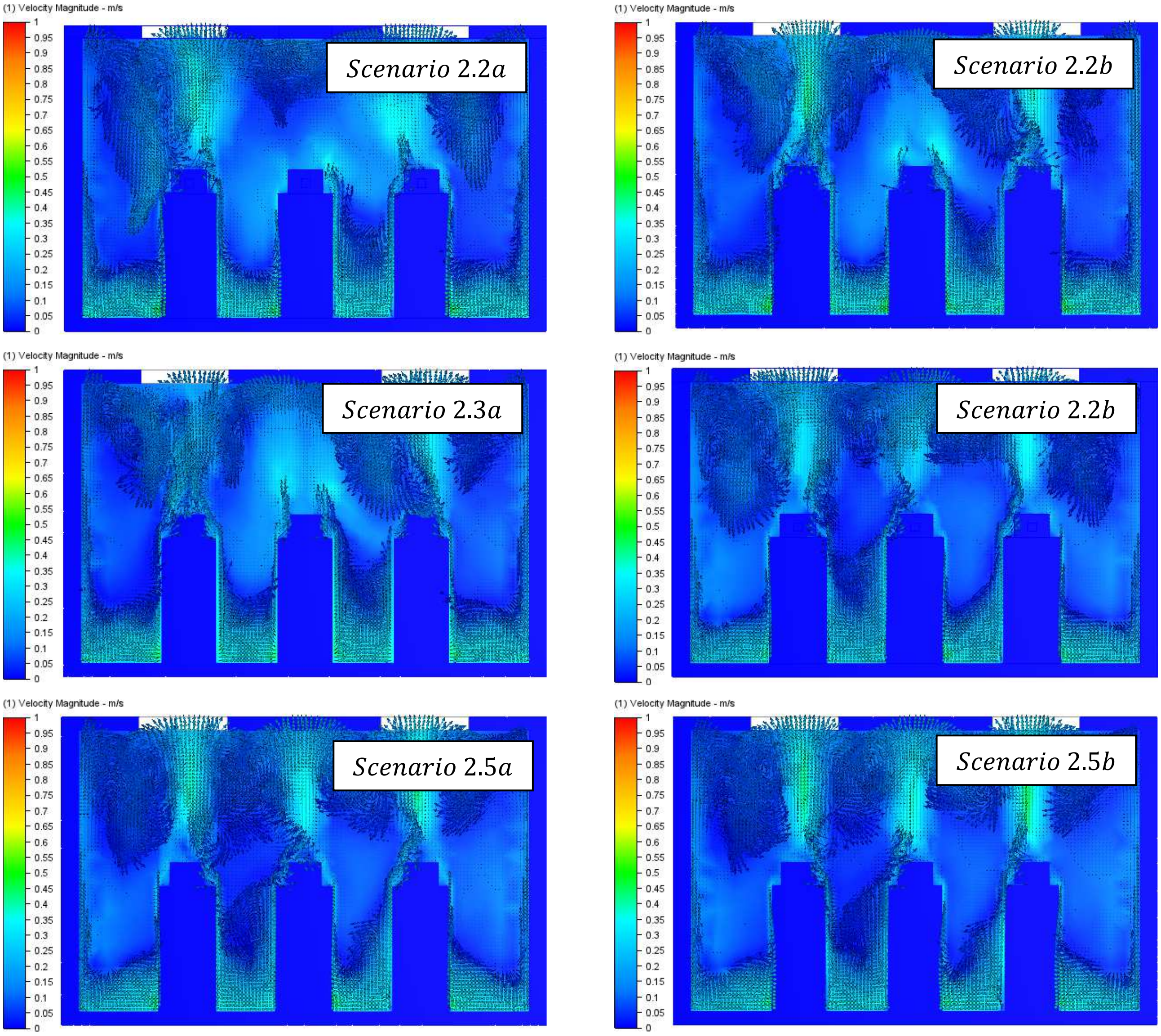


Figure 34 - Velocities scenario 2 axis 2

4.2.2 CO_2 distribution scenario 2

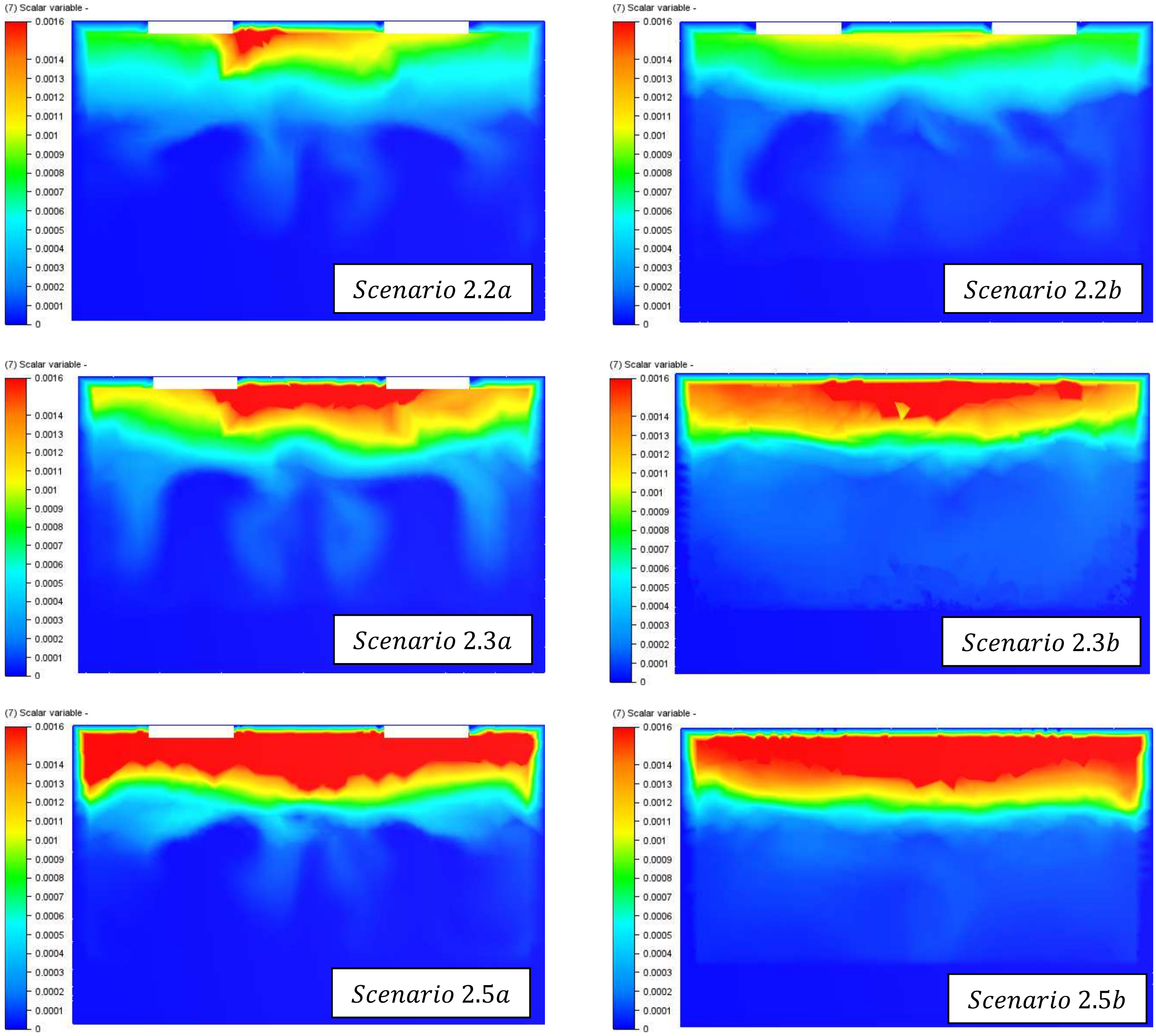


Figure 35 - CO2 distribution scenario 2 axis 1

4.3 Scenario 3 - Radiant cooled ceiling and reduced air volumes

Again, the flow is close to identical in every case again pointing out that the impact of the humans breathing has no notable impact of the air flow in the room. At the ceiling the air starts to fall in the vicinity of the cooled ceiling. By increasing the height of the steps, the spinning athletes are standing on and thus increasing the outlet size of the supply terminal the velocity is decreased and around the athletes the velocity is close to 0.2 m/s. The iso surface image is showing areas with a velocity of $0.2\ m/s$ or above.

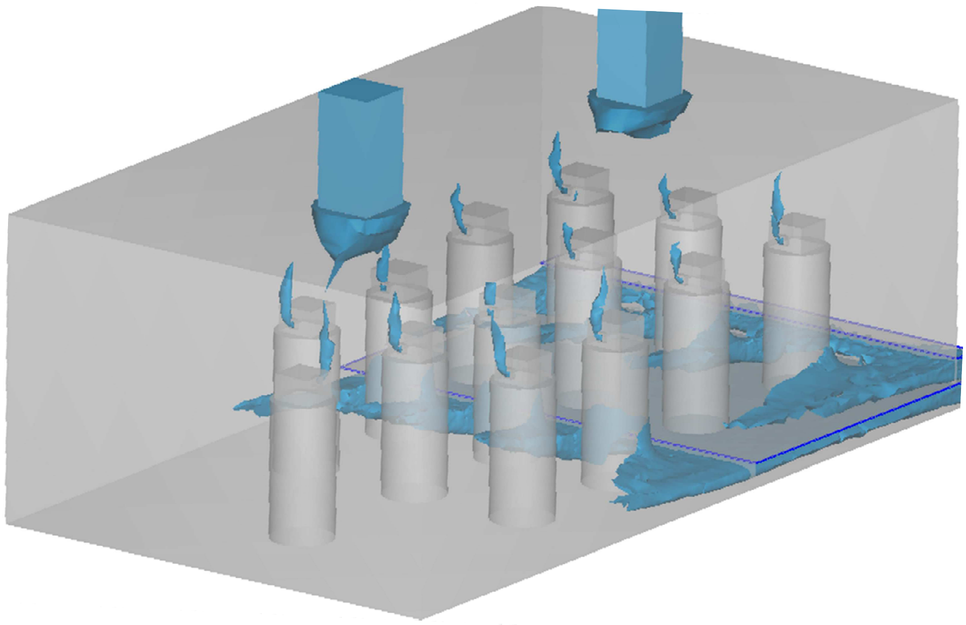


Figure 36 - Iso surface scenario 3

The average temperatures are shown in the figure below and as seen in the previous scenarios there are small differences in the temperature in each case. The temperature is higher in this scenario than in scenario 2, which relates to the reduced air volumes. Looking at the temperature plots in the below graphs it is evident that the readings up to about 1 m are being interfered with heat from the surrounding spinning athletes. This is an indication of the model being "too crowded" for lack of a better term, and that the simulations should be simplified further to achieve more precise readings. However, by analysing the trend lines the temperature between ankles and head is approximated to be slightly above $2\ °C$ which correlates to scenario 2. An attempt to extract the operative temperature was conducted in this scenario, but the software failed to give any reasonable values for operative temperature for the humans. It may be because the heat generated by the spinners exceeds the levels normally generated by humans and are a major contributor to the mean radiant temperature in the software's calculations (due to a high temperature of the skin). This implies that there is a weakness in the simulation. To approximate the operative temperature a coarse calculation of the mean radiant temperature is conducted, and the operative temperature based on this estimation is calculated to be $20\ °C$ (based on scenario 3a). This is in range of the temperature interval given by guidance 444 of a temperature interval of $16 - 26\ °C$. The estimations of operative temperature can be found in appendix C.

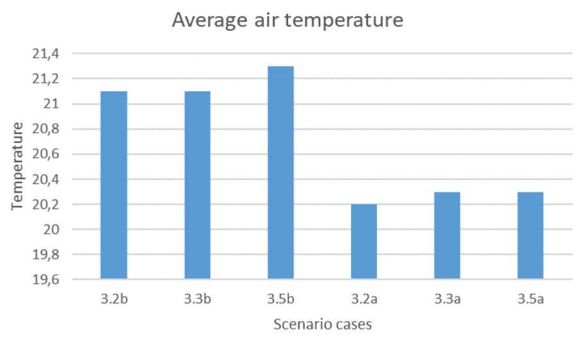


Figure 37 - Average temperature scenario 3

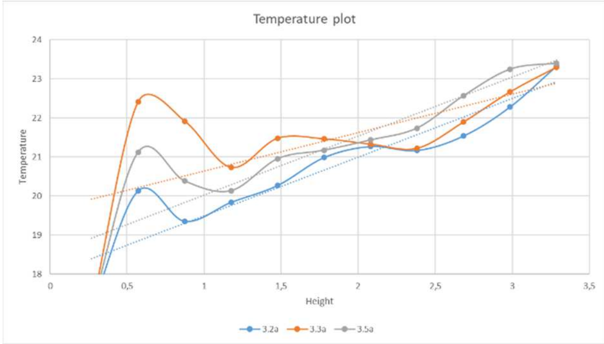


Figure 39 - Temperature plot scenario 2a

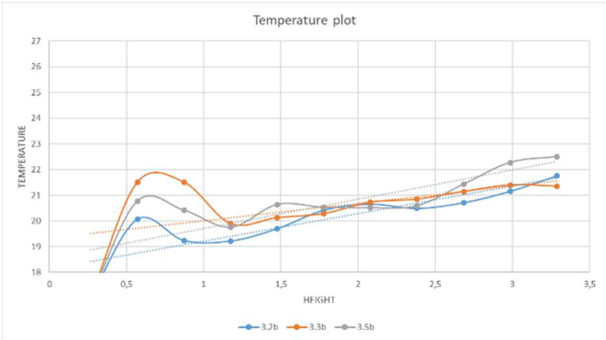


Figure 40 - Temperature plot scenario 3b

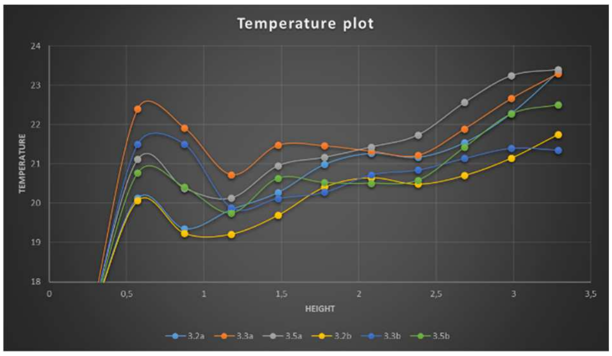


Figure 38 - Temperature plot scenario 3

As in the previous scenarios the CO_2 is accumulated at the ceiling. The observant reader may notice that the density of the CO_2 is in each corresponding case closely resemble one another. However, by comparing the graphs it becomes apparent that the air is denser at a lower height in scenario 3 than in the other scenarios indicating that the decreased air volumes are leading to increased ppm CO_2. Values above $1000\ ppm$ close to the occupied zone is only found in case 5a).

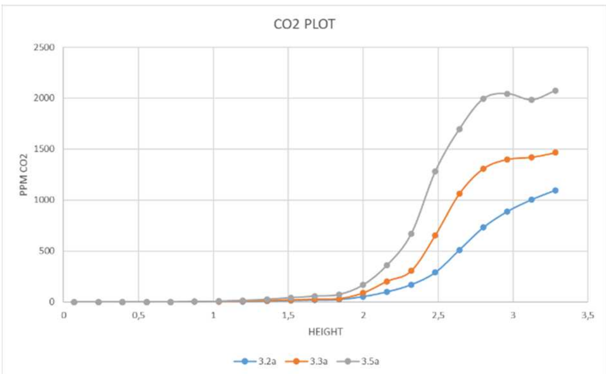


Figure 42 - CO_2 distribution scenario 3a

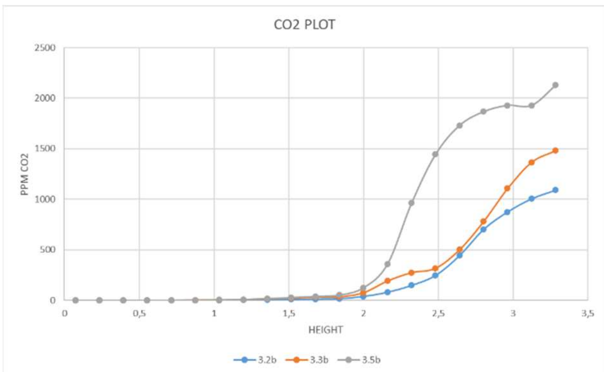


Figure 41 - CO_2 distribution scenario 3b

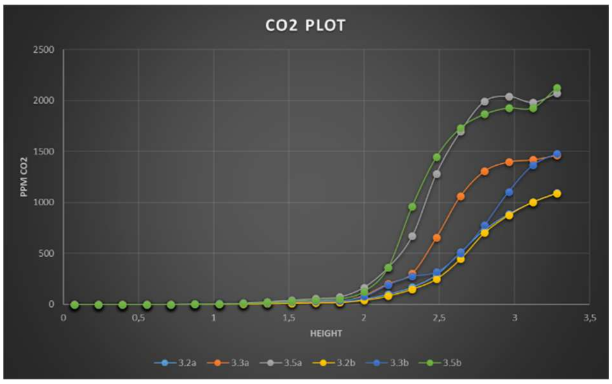


Figure 44 - Temperature plot scenario 3

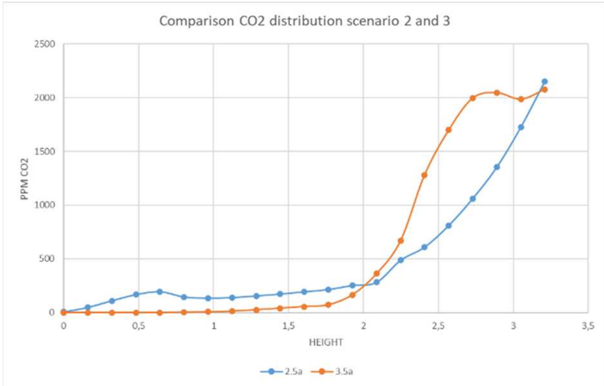


Figure 43 - Comparison of CO2 distribution scenario 2 and 3

Analysing the image below the relative humidity seems to be in the region of 45-50 %. The simulation is not taking into account the moisture added by sweating and by adding this the actual relative humidity is $57 - 62\ \%$. With this knowledge, it is clear that there are no danger of condensation at the cooled ceiling as the dew point temperature at $21\ °C$ and $62\ \%$ relative humidity is $13.45\ °C$. More details about the calculation can be found in chapter 4.4.

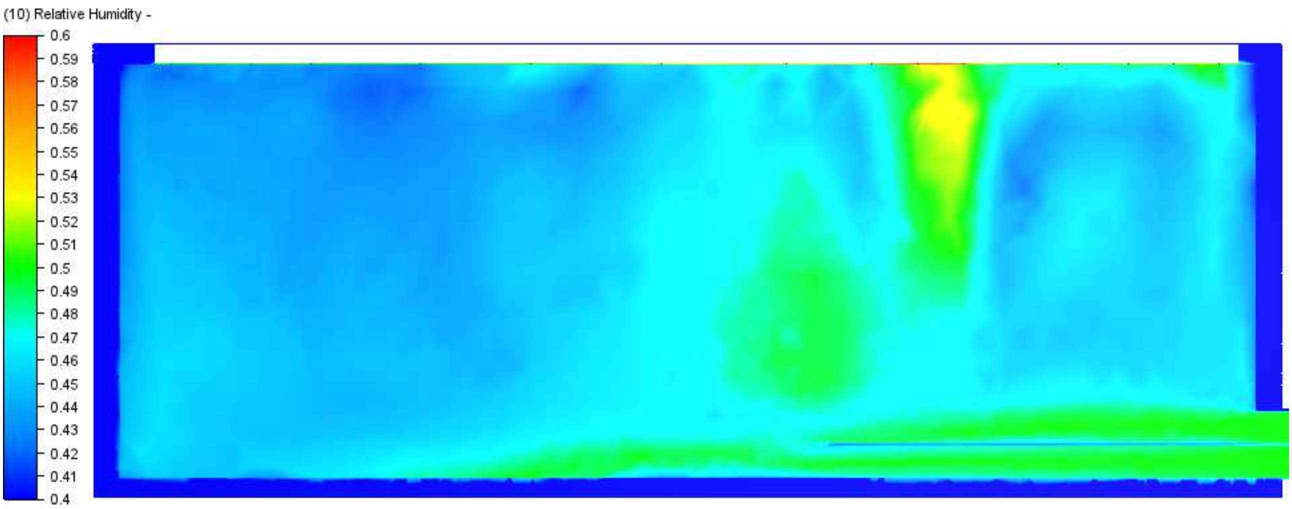


Figure 45 - Relative humidity scenario 3.5a, axis A

The following subchapters contains the images of the result discussed.

4.3.1 Velocity scenario 3

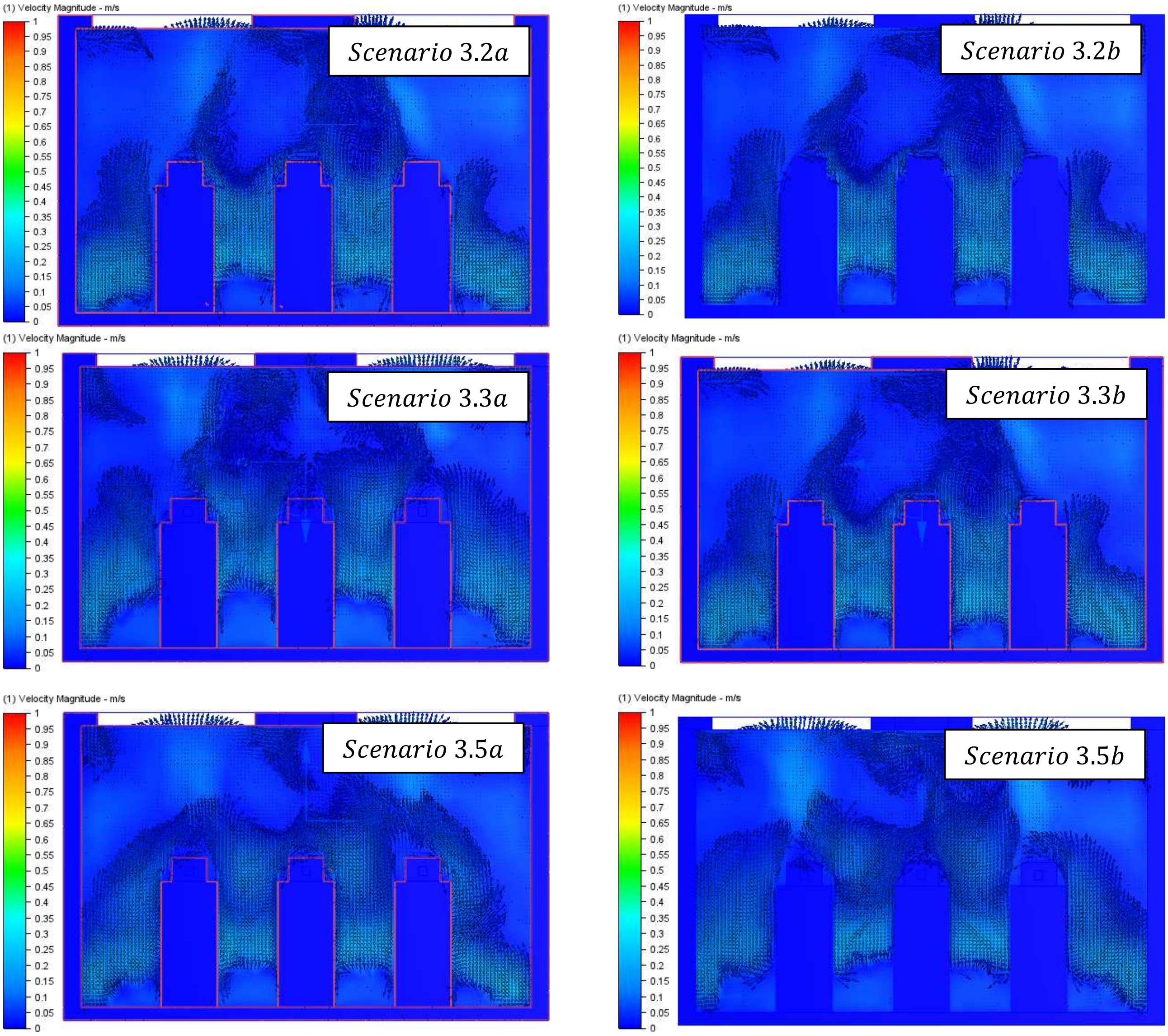


Figure 46 - Velocities scenario 3

4.3.2 CO_2 distribution scenario 3

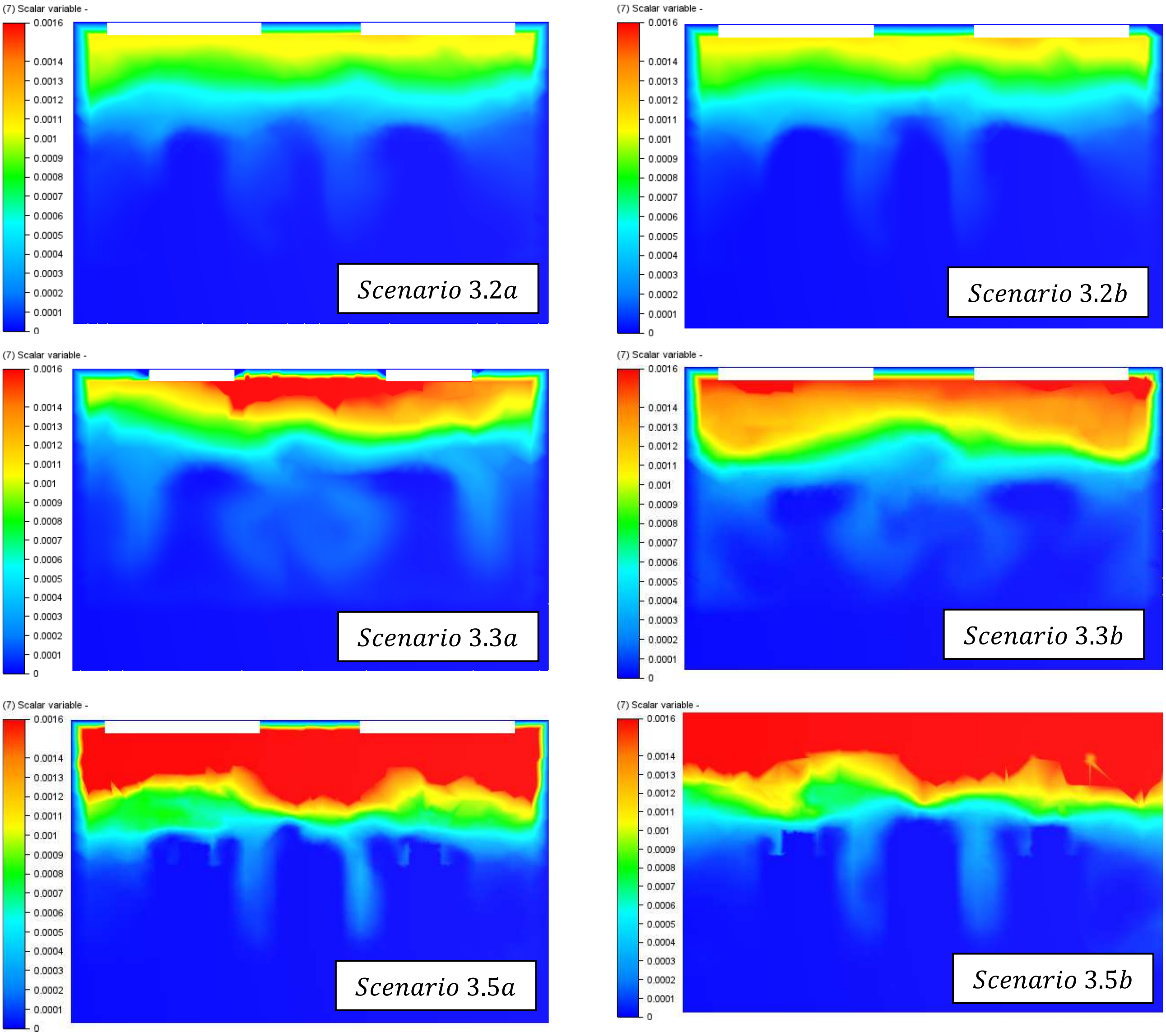


Figure 47 - CO2 distributions scenario 3

4.4 Comparison of simulation results against mathematical models.

To control the validity of the results from the simulations, comparisons against steady-state equations are conducted. Only the simulations for scenario 3 will be compared against steady-state equations. For full calculations see appendix D.

4.4.1 Air temperature

Energy balance:

$$Q_{tot}=Q_{air}+Q_{breath}+Q_{ceiling}+Q_{conduction\ 1}+Q_{conduction\ 2} \tag{2}$$

$$Q_{tot}=\dot{V}_1 C_p \rho_1 (T_{air}-T_{supply})+\dot{V}_2 C_p \rho_2 (T_{air}-T_{breath})+Q_{ceiling} + U_1 m_1^2 (T_{air}-T_{inner})+U_2 m_2^2 (T_{air}-T_{outer}) \tag{3}$$

Solving for T_{air} gives an air temperature of $22.1\ °C$, which is above the temperature given in the simulation. However, note that the cooled ceiling temperature was lowered from $18.7\ °C$ to $18\ °C$, giving the cooled ceiling a higher cooling capacity than used in this calculation. Calculations based on the lower surface temperature of the cooled ceiling gives an air temperature of $21.7\ °C$. The differences in the simulation and the steady-state calculations are $20.2\ °C - 21.7 °C = 1.5\ °C$. The differences in the air temperature is assumed to be caused by the displacement ventilation and simulation set up where the inside walls are given a temperature of $20\ °C$. The energy balance is calculated by using conduction as heat transfer.

4.4.2 CO_2 distribution

Mass balance pollution: [15]

$$V\frac{dC}{dt}=S+C_a nV-CnV-KC \tag{4}$$

Steady state solution:

$$C_\infty = \frac{S/V + C_a n}{n} \tag{5}$$

Solving for C_∞:
$C_\infty \approx 1867\ mg/m^3 \approx 1040\ ppm$

As the steady state solution is based on a perfectly mixed ventilation system with a ventilation efficiency of 1 and there is no way to extract the average value of the scalar value in the simulation, one must rely on a visual analysis between the two. As the CO_2 concentration of the steady state is $1040\ ppm$ we would expect lower values based on the increased ventilation efficiency of the displacement ventilation. The exception would be at the ceiling where the CO_2 is accumulating. As seen in the CO_2 plots shown in each result chapter, most parts of the spinning room have less than the steady state concentration with the exception of the ceiling. At the ceiling the accumulation leads to a denser layer of CO_2. This provides further support for the validity of the simulation.

4.4.3 Relative humidity

As the simulation does not take into account the moisture from sweat, some extra considerations should be attended.

Latent heat flow due to moisture in air:

$$Q_L = h_{e,vap}\, \varrho q_{air} \Delta x$$

Solving for Δx gives

$$\Delta x = 2.83 \cdot 10^{-3}\ kg_{H_2O}/kg_{air}$$

This gives a relative humidity of 57 % which naturally is higher than the simulation which gave a value around of 45 %. By subtracting the moisture added to the air by sweat, a basis of comparison is established. The relative humidity on this basis is calculated to be 46 %. These two values correlate closely with deviations represented in image 45.

5 Discussion

This chapter will discuss the set up and results of this thesis. For calculations around values given in this chapter, see appendix E for details.

One of the most uncertain input values for the simulation is the heat generated by humans as the metabolic rate is closely connected to the individuals. Byggforsk is recommending 10 Met for spinning as an activity. [5] This corresponds to a heat generation of approximately $582\ W/m^2$ per person. The compendium of physical activities however operates with a value of $8.5\ Met$. It is important to point out that $1\ Met$ in the compendium of physical activities equals $1\ Kcal \cdot kg^{-1} \cdot h^{-1}$ which corresponds to $45.2\ W/m^2$ for a standard person weighing $70\ kg$ and with a surface area of $1.8\ m^2$. This means that for an activity like spinning with $8.5\ Met,$ the heat generation per person according to the physical compendium is $384\ W/m^2$. The compendium uses $3.5\ ml/\ (kg \cdot minute)$ as a standard value for the resting metabolic rate (RMR) and for estimating Met values. [16] The method of using standard $Mets$ have received some criticism for being to inaccurate as age, height, mass and sex affects the metabolism, and therefore the standard Met value may underestimate the true energy cost of physical activities. Kozey et al [17] demonstrated the differences of using standard $Mets$ and corrected $Mets$, where corrected $Mets$ are a fraction of standard $Mets$ divided by the Met value given by the Harris-Benedicte RMR [18] showing a misclassification of $12.2\ \%$ using standard $Mets$ compared to $8.4\ \%$ using corrected $Mets$. This method however is not suitable for design options for HVAC systems in a spinning rooms where everyone in the population are possible candidates for a spinning class. Havenith et al [19] refers to ISO 8996 and states that the heat generation at intense to maximum activity is $290\ W/m^2$. M.J Fletcher et al [20] calculated the average Met value during multiphased exercises and evaluates that the average in a spinning class based on BS EN ISO 7730:2005; is $4.7\ Met$ and $6.6\ Met$ based on compendium of physical activities (both in W/m^2). The average between these two are $5.65\ Met$ and for simplicity $6\ Met$ is being used in this thesis for approximation of the heat generation of people which corresponds to $349\ W/m^2$. For a standard person this equals a total energy output of $628\ W$. This heat generation is dispersed between latent and sensible heat and it is the sensible heat that is the interesting part of the heat generation in the simulations. The ratio between latent and sensible heat is dynamic and is affected by sweat [21], clothing [22] and surrounding temperature. [23] This topic is complex and highly variable so the search for a right value for sensible heat is not prioritised. Engineering educators [24] states that the ratio for sensible heat ranges from 70% for light activity to 40% for heavy work. In this thesis the ratio of sensible heat is set to 50% bringing the input value of heat generation per person in the simulation to $314\ W$. Latent heat in the form of vaporized sweat is based on the assumption that all the latent heat ($314\ W$) is vaporized. Converting this power into mass gives a sweat production of one human approximately $0.45\ l/s$. This may seem low, but as Gagning and Crandall [25] states, the maximum sweat production usually lies between 1.5 l/h-2.5 l/h but all of this is not evaporized and contributing to the cooling of humans. Most of this is collected by the clothes, which contributes on the cooling on its own way. [26] For simplicity all of the cooling power of the latent heat is being treated as vaporized sweat from the skin of the humans.

The CO_2 production of humans is another parameter with uncertainties. By using Ashrae [27], either by diagram or formula, the amount of CO_2 generation can be found based on activity. Using $6\ Met$ as metabolic rate gives a production of $1.54\ l/min$. A. Persily and L. de Jonge presented in 2017 an alternative approach to estimate CO_2 generation of building occupants. [28] By using their approach calculated CO_2 productions equals $1.27\ l/min$. Davies et al [29] shows the relation between CO_2 production and ventilation rates of persons and gives a ventilation rate of (approximately) $37\ l/min$ (Ashrae) and $33\ l/min$ (A. Persily and L. de Jonge). These ventilation rates may seem low as the ventilation rate may go as high as $100\ l/min$ [30] during high and intense work out, however the metabolic rate value that is used in the calculations are based on an average Met value as discussed earlier. However, it may be a period of time during the exercise session where the CO_2 production is larger than the estimates given here, based on the cooldown period the body needs to lower the metabolic rate [31] which is why the scenarios are simulated with different ventilation rates for the humans to see how big of an impact the different respiration ratios have. In both cases the relation between CO_2 and air (breath) are approximately $4\ \%$ which is also confirmed to be a usual value by Mateescu [32]. The breathing for humans is therefore set with a value of $40\ 000\ ppm$. The breathing is also given a $100\%\ RH$.

The velocity of the air out of the supply terminals and around the ankles is around $0.15 - 0.3\ m/s$. By calculating the draught rate given by NS-EN ISO 7730:2005 [7] the percentage of people predicted to be bothered by draught is close to 14 %. This prediction is not a good estimation of the predicted percentage however, as NS-EN ISO 7730:2005 states that the equation is applicable at activities around $1.2\ Met$ and the neck region. Draugth is not perceived in the same matter when the activity levels are high and when people are feeling warmer than neutral, which is plausible that they do in a spinning session. The prediction is therefore assumed to be lower than $14\ \%$. As the humans are elevated slightly from the ground because they are sitting on a bike, is also contributing to the assumation that the predicted perecentage is lower. The assessments around draught and elevated air velocities is therefore concluded to not be a major concern in the simulated scenarios. The exception is the supply air from the fan coil scenarios where the placement of the supply terminal is leading to cold air with relative high velocity blowing at the person in the close vicinity and increasing the feeling of discomfort. This shows that the placement of the supply terminal of fan coils need considerations to avoid increased local discomfort due to draught.

The simulations showed that the radiant cooled ceiling has a decreased thermal stratification compared to the fan coil scenario. This is in accordance with the results presented by Schiavon et al [33] wich states that the stratification will decrease when a larger portion of the cooling load is removed by the cooled ceiling and the ratio between the cooling load and air flow is increased, but increase with the temperature of the cooled ceiling. Schiavon et al [34] states that when the cooled ceiling temperature is minimum $18\ °C$, a minimum temperature gradient of $1.5\ °C$ is ensured. By using the equation given by Schiavon et al [34] the predicted temperature gradient would be $2.1\ °C$ which correlates well with the result from the simulation.

When it comes to the recycling of air which is shown in the fan coil scenario and the possibilities of increased levels of CO_2 due to this some assessments should be made. Anbefalte faglige normer states that the concentration should not exceed $1000\ ppm$, however, elevated values above this alone is not enough to determine the air quality which is also mentioned by Mudarri [35]. Anbefalte faglige normer says that levels up to $9000\ mg/m^3$ ($5000\ ppm$) isolated does not have any negative impact of health, sensory percieption or performance of work [11]. Taheri et al [36] showed that the fan coil unit is capable of reducing the amount of pollutants in the air, and based on this it is considered that the recycled air from the fan coil is not decreasing the air quality despite the probability of raised levels of CO_2.

The estimation of the operative temperature in scenario 3 indicated an operative temperature of around 20 °C. This is in the acceptable region given by guidance 444, but according to the guidance for sports halls, the temperature should be $18\ °C$. This temperature however is the air temperature. Guidance for sport halls is not mentioning operative temperature. In either case the temperature is above the recommendations given by the guidance for sport halls. Jaax [37] found in his master thesis that the preferred air temperature at high activity should be between $15-16\ °C$. Zhai et al [38] however, stated through their results that the design temperature should be in the region of $22-24\ °C$ which are significally higher than mentioned earlier. This is based on an activity level of $4.5\ Met$ which are lower than the expected metabolism in a spinning room, but still it points at higher temperatures than $18\ °C$ as a design criteria. A key point based on this discussion is that there is a wide range of preferred temperatures of each individual.

5.1 Limitations

The limitations of the thesis will be presented in this chapter.

- Sensible values of the operative temperature is not extracted from the simulation. The temperature may be affected by the human models which have a heat generation of 314 W. This leads to unnatural high "skin" temperature of the humans.
- There are differences in the air temperature calculated by the energy balance and the simulation. The differences may be due to slight differences in the simulation set up versus the mathematical equation.
- Some of the readings, in particular while plotting the graphs, have values interrupted by a too "crowded" model. The CFD simulation should have been simplified further. 2-3 persons should be adequate to prevent too much interference and less computational time.
- The simulation is not validated by experimental data. The corona pandemic has made it impossible to do on-field readings for comparisons to the simulation.

5.2 Further research

Based on the results and limitations the following suggestion for further research.

- The scenarios can be simplified further and thus ensuring that the results from this simulation is valid.
- The simulations can be validated through on-field measures or as an experiment.

6 Conclusions

CFD simulations where conducted to lower the air volumes and optimize the design of a spinning room. The simulations are based on an existing spinning room simplified for the simulation.

The reduced air volumes are sufficient to ensure good air quality in the room. The reduced air volumes need to be compensated with a greater cooling capacity of the cooling system. Scenario 1 presented a possible draught risk at the use of fan coils, which would only be greater with the need for greater cooling capacity by fan coils. With the use of radiant cooled ceiling, the supply air needs to be condensed to avoid condensation at the cooled ceiling as scenario 2 reveals. Simulation of scenario 3 revealed that although there may be periods when the CO_2 concentrations are above the recommended values given by faglige normer, when the spinning session is at its peak, the overall air quality is good. Based on the results of scenario 3, recommended air volumes per person in a spinning room is suggested to be lowered from the recommendations given by guidance 421.503 and the guidance from the ministry of culture, from 250 m^3/h to $\mathbf{150}\ m^3/h$. The simulations should be validated by physical measurements before this recommendation is valid.

7 References

[1] Olje og Energidepartementer, «Energifakta Norge,» Last Friday, 2021. [Internett]. Available: https://energifaktanorge.no/et-baerekraftig-og-sikkert-energisystem/baerekraftige-bygg/. [Funnet 06 June 2021].

[2] Standard Norge, «Standard Norge,» Standard Online, 27 Mai 2021. [Internett]. Available: https://www.standard.no/nyheter/nyhetsarkiv/kvalitet-og-risiko/2020-nyheter/na-heves-standarden-for-norske-treningsentre-/. [Funnet 04 June 2021].

[3] International Energy Agency, «Buildings - A source of enormous untapped efficiency potential,» International Energy Agency, 2021. [Internett]. Available: https://www.iea.org/topics/buildings. [Funnet 04 June 2021].

[4] Kulturdepartementet, Idrettshaller. Planlegging og bygging, Oslo: Kulturdepartementet, 2016.

[5] Sintef Byggforsk, 421-503: Luftmengder i ventilasjonsanlegg. Krav og anbefalinger, Oslo: Sintef Byggforsk, 2017.

[6] Direktoratet for byggkvalitet, «Byggteknisk forskrift (TEK17) med veiledning,» Direktoratet for byggkvalitet, 23 February 2021. [Internett]. Available: https://dibk.no/regelverk/byggteknisk-forskrift-tek17/. [Funnet 23 February 2021].

[7] Norsk standard, Ergonomi i termisk miljø. Analytisk bestemmelse og tolkning av termisk velbefinnende ved kalkulering av PMV- og PPD-index og lokal termisk komfort, Oslo: Standard norge, 2006.

[8] Arbeidstilsynet, «Luftkvalitet på arbeidsplassen,» Arbeidstilsynet, Trondheim, 2016.

[9] Sintef Byggforsk, 552.351 - Fordeling av ventilasjonsluft i rom, Oslo: Sintef Byggforsk, 1996.

[10] Nemitek, «Kompetansebiblioteket,» Nemitek, 23 February 2021. [Internett]. Available: http://kompetansebiblioteket.no/Ventok.aspx. [Funnet 23 February 2021].

[11] Folkehelseinstituttet, «Anbefalte faglige normer for inneklima,» Nasjonalt Folkehelseinstitutt, Oslo, 2015.

[12] Sintef Byggforsk, 421.501 - Termisk inneklima. Betingelser, tilrettelegging og målinger, Oslo: Sintef Byggforsk, 2017.

[13] S. Ingebrigtsen, Ventilasjonsteknikk del 1, Oslo: Skarland Press, 2017.

[14] NASA, «National aeronatuics and space administration,» 13 Mai 2021. [Internett]. Available: https://www.grc.nasa.gov/www/k-12/airplane/nseqs.html. [Funnet 06 June 2021].

[15] W. P. E. Gilbert M. Masters, Introduction to environmental engineering and science, Essex: Pearson Education Limited, 2014.

[16] H. W. H. S. M. N. B. J. D. T.-L. C. G. J. V. J. W.-G. M. L. A. Ainsworth BE, «The Compendium of Physical Activities Tracking Guide,» Healthy Lifestyles Research Center, College of Nursing & Health Innovation, Arizona State University, 2011. [Internett]. Available: https://sites.google.com/site/compendiumofphysicalactivities/home. [Funnet 11 April 2021].

[17] K. L. J. S. P. F. Sarah Kozey, «Errors in MET estimates of physical activities using 3.5 ml x kg(-1) x min(-1) as the baseline oxygen consumption,» *Journal of physical activity,* pp. 508-516, 7 July 2010.

[18] F. G. B. J. Arthur Harris, «A biometric study of human basal metabolism,» *Proc Natl Acad Sci USA,* pp. 370-373, 8 October 1918.

[19] I. H. K. P. George Havenith, «Personal factors in thermal comfort assessment: clothing properties and metabolic heat production,» *Energy and buildings,* vol. 2002, nr. 34, pp. 581-591, 2002.

[20] D. G. A. H. C. M.J Fletcher, «A modified approach to metabolic rate determination for thermal comfort prediction during high metabolic rate activities,» *Building and environment,* 12 September 2020.

[21] C. G. C. Daniel Gagnon, «Chapter 13 - Sweating as a heat loss thermoeffector,» *Handbook of clinical neurology,* vol. 2018, nr. 156, pp. 211-232, 2018.

[22] M. G. R. X. W. P. B. V. C. E. d. H. I. H. K. K. H. M. W. N. George Havenith, «Apparent latent heat of evaporation from clothing: attentuation and "heat pipe" effects,» *Journal of Applied Physiology,* pp. 142-149, 18 October 2018.

[23] E. F. D. James D. Hardy, «Regulation of Heat Loss from the Human Body,» *Proc Natl Acad Sci USA,* pp. 624-631, 23 December 1937.

[24] Engineer educators, «Engineer educators,» Engineer educators, 4 16 2021. [Internett]. Available: https://engineer-educators.com/topic/3-heat-transfer-from-the-human-body/. [Funnet 4 16 2021].

[25] C. G. C. Daniel Gagning, Handbook of clinical neurology, Dallas: Elsevier B.V, 2018.

[26] M. G. R. X. W. P. B. V. C. E. d. H. I. H. K. K. H. M. W. N. George Havenith, «Evaporative cooling: effective latent heat of evaporation in relation to distance from skin,» *Journal of applied physiology,* pp. 778-785, 14 January 2013.

[27] ASHRAE, «Ventilation for acceptable indoor air quality,» ASHRAE, Atalanta, 2016.

[28] L. d. J. A. Persily, «Carbon dioxide generations rates for building occupants,» *Indoor Air,* pp. 868-879, 14 March 2017.

[29] B. J. W. K. W. James A. Davis, «The relation of ventilation to metabolic rate during moderate exercise in man,» *European Journal of Applied Physiology,* pp. 97-108, 17 March 1980.

[30] Breathe (Sheff), «NCBI,» US National Library of Medicine National Institutes of Health, 16 March 2016. [Internett]. Available: https://www.ncbi.nlm.nih.gov/pmc/articles/PMC4818249/#. [Funnet 23 April 2021].

[31] M. L. B. C. Y. Z. Y. G. B. L. Wenjie Ji, «A new method to study human metabloc rate changes and thermal comfort in physical exercise by CO2 measurement in an airtight chamber,» *Energy and Buildings,* vol. 2018, nr. 177, pp. 402-412, 2018.

[32] C. Mateescu, «Researchgate,» ResearchGate, 26 January 2018. [Internett]. Available: https://www.researchgate.net/post/What-is-the-CO2-generation-ppm-h-by-a-human-at-different-levels-of-excercizes. [Funnet 24 April 2021].

[33] F. S. B. B. T. J. R. Stefano Schiavon, «Chilled ceiling and displacement ventilation system: Laboratory study with high cooling load,» *Science and Technology for the built environment,* 18 April 2015.

[34] B. T. F. B. J. R. Stefano Schiavon, «Room air stratification in combined chilled ceiling and displacement ventilation systems,» UC Berkeley: Center for the built environment, Berkeley, 2012.

[35] D. H. Mudarri, «Potential correction factors for interpreting CO2 measurements in buildings.,» *ASHRAE Transactions,* vol. 1997, nr. 103(2), pp. 333-341, 1997.

[36] S. A. Z. H. H. M. S. Morteza Taheri, «Numerical investigation of the effects of fan-coil airflow direction on distribution and deposition of indoor pollutant particles,» *Journal of building engineering,* 27 May 2020.

[37] J. R. Jaax, «Investigation of thermal comfort (thermally neutral) conditions for three activity levels,» Kansas state university, Kansas, 1967.

[38] S. Z. Y. G. W. S. L. Y. H. Z. E. A. Yongchao Zhai, «Preferred temperatures with and without air movement during moderate exercise,» *Energy and Buildings,* p. 109565, 27 October 2020.

[39] R. A. A. S.-O. M. A. S. Tomasz Cholewa, «On the heat transfer coefficient between heated/cooled radiant ceiling and room,» *Applied Thermal Engineering,* pp. 76-84, 5 February 2017.

Appendices

Appendix A – Cooled ceiling surface temperature

Cooling capacity supply air:

$Q_{air}=C_p \varrho \dot{V} \Delta T$

Where:
$C_p = 1.006\ kJ/m^3$
$\rho = 1.216\ kg/m^3$
$\dot{V} = 1800\ m^3/h$
$\Delta T = 5\ °C$

Solving for Q_{air}
$Q_{air} \approx 3.06\ kW$

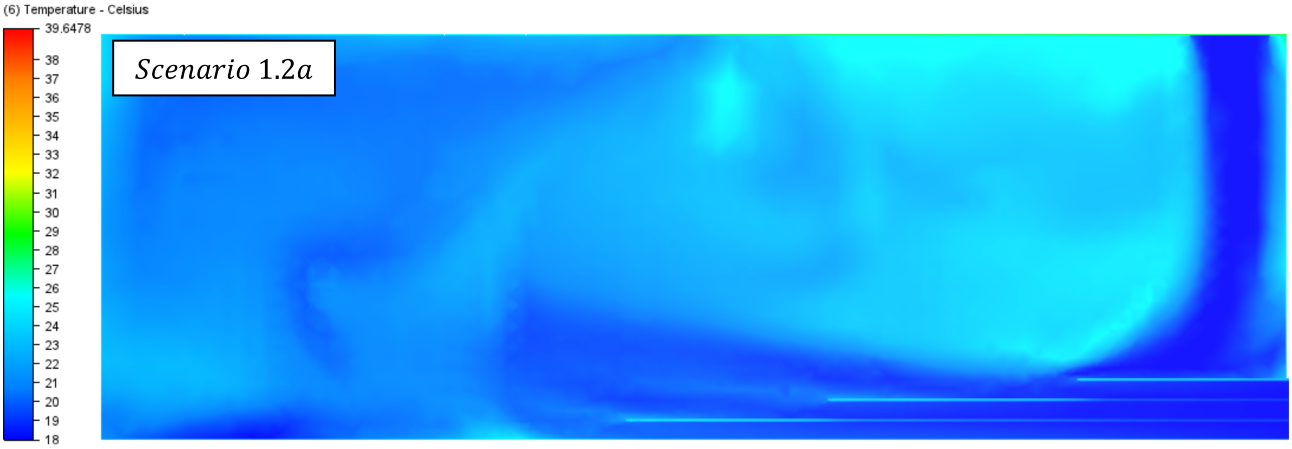
Cooling demand cooled ceiling:
$Q_{tot} - Q_{air} = 4.08\ kW - 3.06\ kW = 1.02\ kW$
$q_{flux} = 1.02 \dfrac{kW}{30.24\ m^2} \approx 33.7\ W$

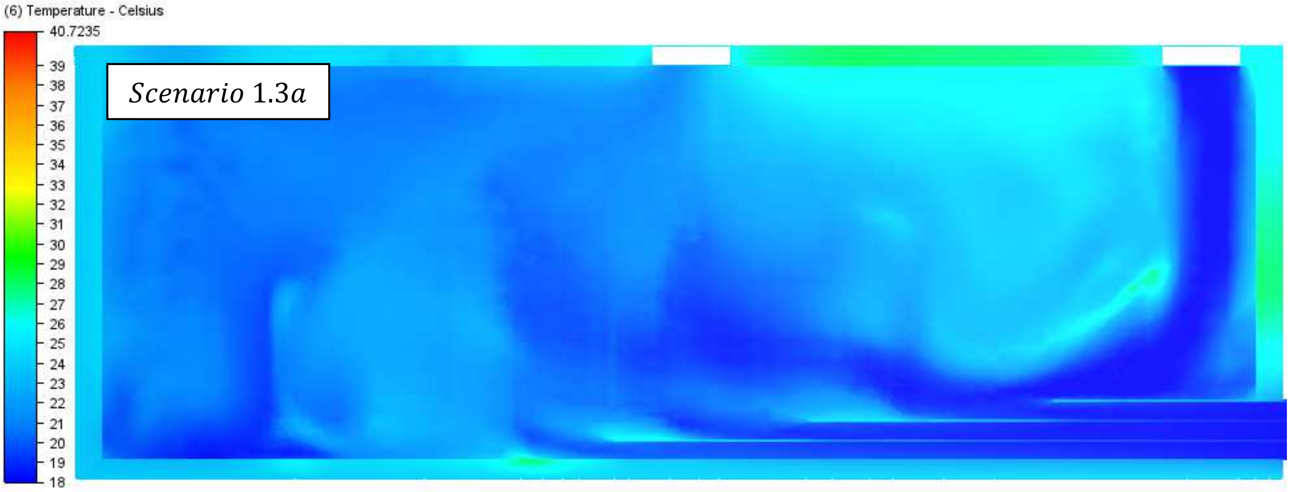
Determining the surface temperature of the cooled ceiling:
The equation is acquired from Cholewa et al, On the heat transfer coefficient between heated/cooled radiant ceiling and room [39]

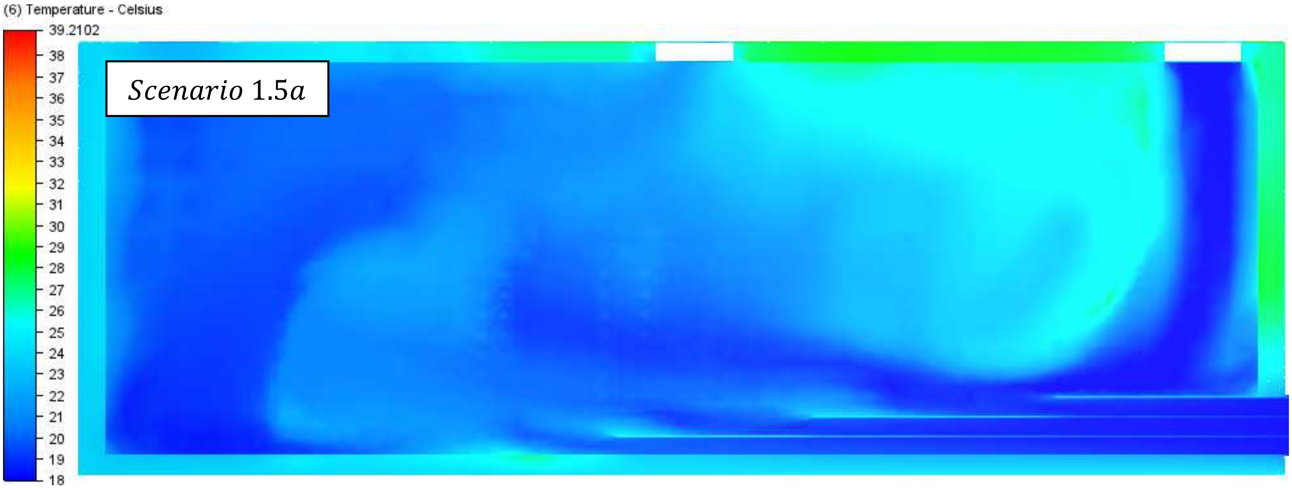
$$T_s = T_o - \frac{q_{flux}}{h_{tot}}$$
$$h_{tot} = 9.36(T_o - Ts)^{0.1}$$

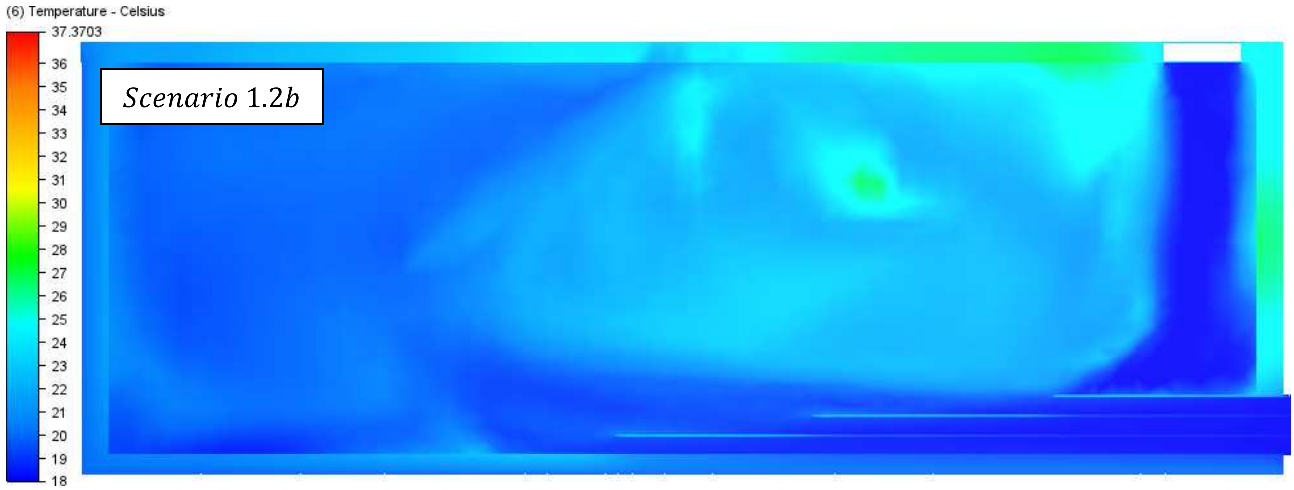
Where:
$T_o = 22\ °C$
$q_{flux} = 33.7\ W$
Substituting h_{tot} in equation 1 with equation 2 and solving for T_s
$T_s \approx 18.8$
As the dewpoint temperature at $20.2\ °C$ and $57\ \%\ RH$ is approximately $12\ °C$ the temperature of the ceiling is set to $T_s = 18\ °C$.

Appendix B – Section view of temperatures scenario 1








(6) Temperature - Celsius
37.3703
36
35
34
33
32
31
30
29
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27
26
25
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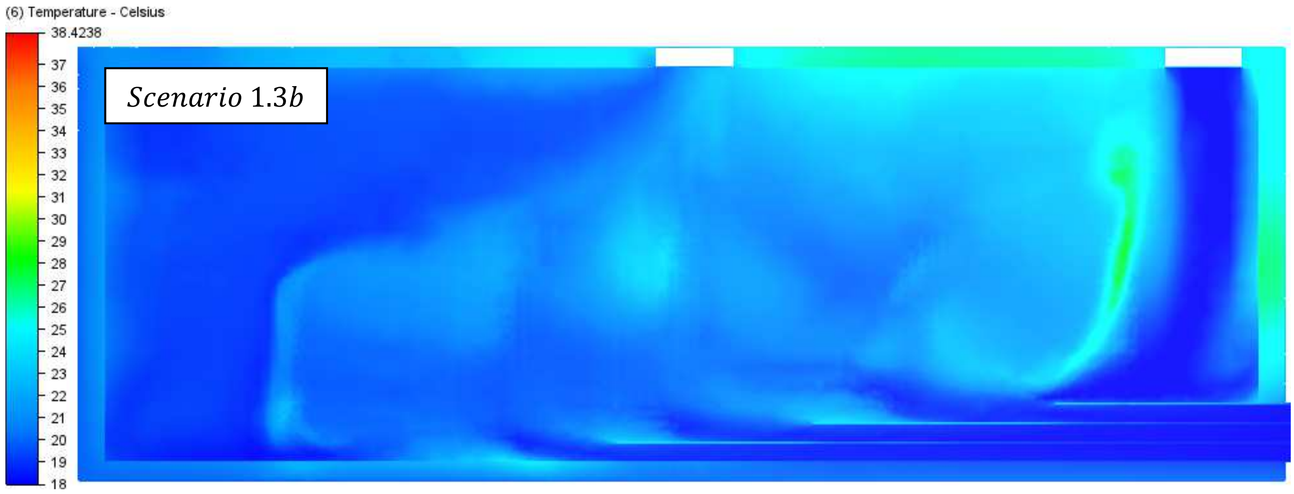
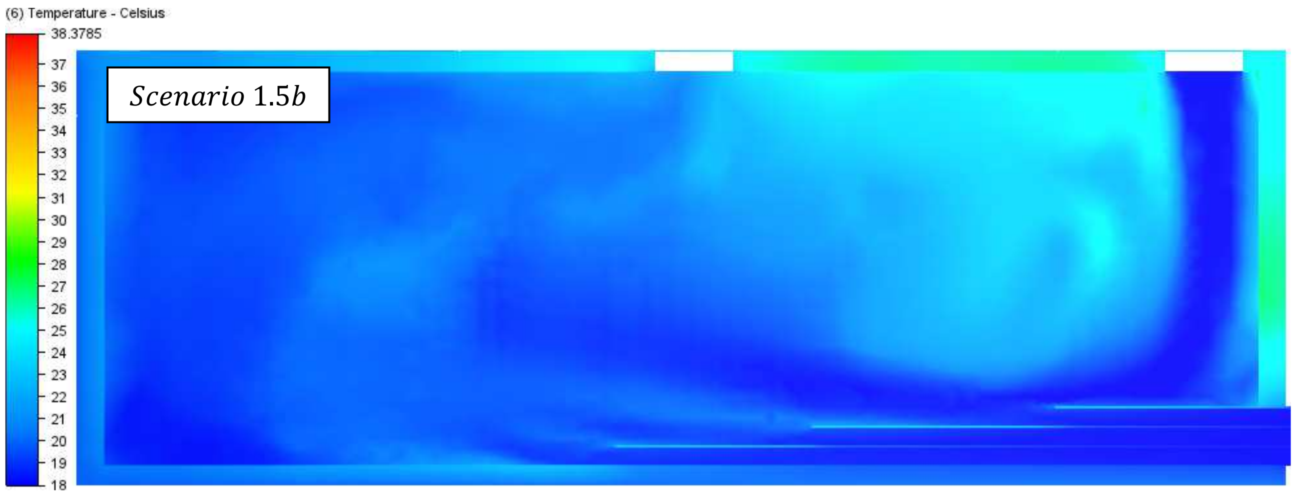
23
22
21
20
19
18
Scenario 1.2b


(6) Temperature - Celsius
38.4238
37
36
35
34
33
32
31
30
29
28
27
26
25
24
23
22
21
20
19
18
Scenario 1.3b


(6) Temperature - Celsius
38.3785
37
36
35
34
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Scenario 1.5b

Appendix C – Estimation of operative temperature

Every temperature is from the simulation which gave the following results:
Temperature walls $=T_w=20$ °C
Temperature floor $= Tf = 20\,°C$
Temperature ceiling $= 21.1\,°C$
Temperature cooled ceiling $= 18\,°C$

Assumptions and simplifications:

- The radiant temperature from neighbouring spinning athletes are neglected.
- Every surface area has the same fraction of view factor with the exception of the ceiling.
- The ceiling is divided into two parts, each with an equal view factor.

Coarse simplification of the view factor
View factor walls $= F_w = \frac{1}{6}$
view factor floor $= F_f = \frac{1}{6}$
View factor ceiling $= F_c = \frac{1}{12}$
view factor cooled ceiling $= F_{cc} = \frac{1}{12}$

$$T_{mean} = 4F_w \cdot 20\,°C + F_f \cdot 20\,°C + F_c \cdot 21.1 + F_{cc} \cdot 18\,°C = 19.9\,°C$$

Operative temperature:

$$\frac{T_{air} + T_{mean}}{2} \approx 20\,°C$$

Appendix D – Comparison of results

Conversions between units is left out of the presentation of the calculations.

Energy balance:
Assumptions and simplifications:

- The inner walls and floor are assumed to have a U-value of 1.1 W/m^{2}K.
- The temperature of the nearby rooms is assumed to be $20\ °C$.
- The total energy flow is based on the sensible heat of the humans.

$$Q_{tot} = Q_{air} + Q_{breath} + Q_{ceiling} + Q_{conduction\ 1} + Q_{conduction\ 2}$$
$$Q_{tot} = \dot{V}_1 C_p \rho_1 \left(T_{air} - T_{supply}\right) + \dot{V}_2 C_p \rho_2 (T_{air} - T_{breath}) + Q_{ceiling} + U_1 m_1^2 (T_{air} - T_{inner}) + U_2 m_2^2 (T_{air} - T_{outer})$$

Where:
$Q_{tot} = 4.082\ kW$
$\dot{V}_1 = 1800\ m^3/h$
$C_p = 1.006\ kJ/kg\ K$
$\rho_1 = 1.216\ kg/m^3$ at $17\ °C$
$T_{supply} = 17\ °C$
$\dot{V}_2 = 65\ m^3/h$
$\rho_2 = 1.146\ kg/m^3$ at $35\ °C$
$T_{breath} = 35\ °C$
$Q_{ceiling} = 1.02\ kW$ (With a surface temperature of $18.8\ °C$)
$U_1 = 1.1\ W/m^2K$
$m_1^2 = 99.7\ m^2$
$T_{inner} = 20\ °C$
$U_2 = 0.22\ W/m^2K$
$m_2^2 = 69.4\ m^2$
$T_{outer} = 21.5\ °C$

Solving for T_{air}
$T_{air} \approx 22.1\ °C$

Cooling power of ceiling at surface temperature of $18\ °C$:
$T_s=T_o-q_{flux}/9.36(T_o-T_s)^{0.1}$
$T_s = 18\ °C$
$T_o = 22\ °C$
Solving for q_{flux}
$q_{flux} = 43\ W/m^2$ which gives $Q_{ceiling} \approx 1300\ W$ at a surface area of $30.24\ m^2$ of the celing.

Using this as $Q_{ceiling}$ at the energy balance gives an air temperature of $T_{air} = 21.7\ °C$

Mass balance pollution:
The equation is acquired from Gilbert and Wendell, introduction to environmental engineering and science [15]

$$V\frac{dC}{dt} = S + C_a nV - CnV - KCV$$

By setting $K = 0$ (decay) and $dC/dt = 0$ the steady state solution takes the form:

$$C_\infty = \frac{S/V + C_a n}{n}$$

Where:
$C_\infty = CO_2$ concentrations at steady state
S = source emission rate = $n_{humans}\dot{V}_2\rho_{CO_2}\ \left[\frac{mg}{h}\right]$
$\dot{V}_2$=1.54 l/minute
$\rho_{CO_2} = 1.724\ kg/m^3$ at $35\ °C$
$C_a = 720\ mg/m^3$
$n = \frac{1800\ m^3/h}{132.7\ m^3} \approx 13.6\ h^{-1}$
$V = 132.7\ m^3$

Solving for C_∞ gives:
$C_\infty \approx 1867\ mg/m^3 \approx 1040\ ppm$

Latent heat flow due to moisture in air*:*

$$Q_L = h_{ew,vap}\ \rho_{air} q_{air}\Delta x$$

Where:
$Q_{L_{tot}} = 4082\ kW$
$h_{ew,vap} = 2439.3\ kJ/kg$
$\rho_{air} = 1.181\ kg/m^3$
$q_{air} = 1800\ m^3/h$

Solving for Δx gives
$\Delta x = 2.83 \cdot 10^{-3}\ kg_{H_20}/kg_{air}$

Estimation of moisture added to air by sweat

Calculating latent heat by breathing:
$Q_{L_{breath}} = \dot{V}_{breath} h_{ew,vap}\rho_{air}(x_2 - x_1)$

Where:
$\dot{V}_{breath} = 65\ m^3/h$
$h_{ew,vap} = 2417.9\ kJ/kg$
$\rho_{air} = 1.146\ kg/m^3$ at $35\ °C$
$x_1 = 0.007351\ kg_{H_2O}/kg_{air}$ at $20.2\ °C$ and $50\ \%\ RH$
$x_2 = 0.036548\ kg_{H_2O}/kg_{air}$ at $35\ °C$ and $99.9\ \%\ RH$

Solving for $Q_{L_{breath}}$
$Q_{L_{breath}} = 1.46\ kW$

Calculating latent heat by sweating
$Q_{L_{sweat}} = Q_{L_{tot}} - Q_{L_{breath}} = 4.082\ kW - 1.46kW \approx 2.62\ kW$

Calculating Δx based on latent heat in sweat
$Q_{L_{sweat}} = \dot{V}_{supply} h_{ew,vap}\rho_{air}\Delta x$

Where:
$Q_{L_{sweat}} = 2.62\ kW$
$\dot{V}_{supply} = 1800\ m^3/h$
$h_{ew,vap} = 2417.9\ kJ/kg$
$\rho_{air} = 1.216\ kg/m^3$ at $17\ °C$

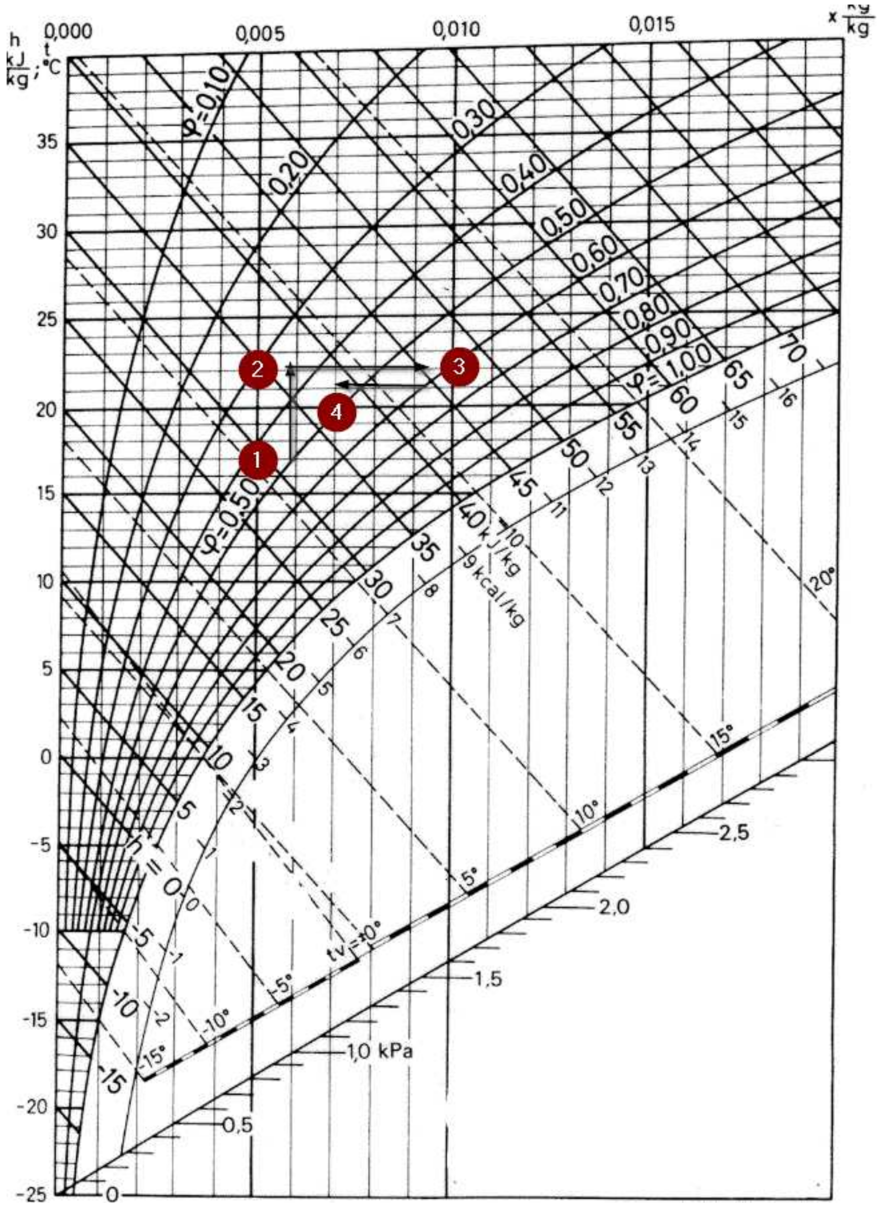
Solving for Δx
$\Delta x \approx 0.002\ kg_{H_2O}/kg_{air}$

Comparing Relative humidity with simulation

$$x_{17\,°C\ and\ 50\,\%\,Rh} \approx 0.006\frac{kg_{H_2O}}{kg_{air}}$$
$$RH_{20.2\,°C\ and\ x=0.006} \approx 41\ \%$$
$$RH_{20.2\,°C\ and\ x=0.0084\_} \approx 57\ \%$$

Subtracting the latent heat from sweat:

$$RH_{20.2\,°C\ and\ x=0.0064\_} \approx 46\ \%$$



Appendix E - Calculations done in discussion

Heat generation Byggforsk:

$$10\ \text{Met}\cdot 58.2\ \text{W/m}^2 = 582\ \text{W/m}^2$$

Heat generation Compendium of physical activities: [16]

$$1\frac{kcal}{kg\cdot h} = 4184\frac{J}{kg\cdot\ h} = \ 4184\frac{J}{kg\cdot\ h}\cdot\frac{1h}{3600\ s} \approx 1.16\frac{W}{kg}$$

A standard person is $70\ kg$ and has a surface area of $1.8\ m^2$:

$$1.16\frac{W}{kg}\cdot 70\ kg\cdot\frac{1}{1.8\ m^2} \approx 45.2\ W/m^2$$

Heat generation at $\boldsymbol{8.5\ Met}$:

$$8.5\ Met\cdot 45.2\frac{W}{m^2} \approx 384\ W/m^2$$

Total energy generation per person:
Where 1 standard person has a surface area of $1.8\ m^2$.

$$384\frac{W}{m^2}\cdot 1.8\ m^2 \approx 692\ W$$

CO_2 production by using equations from A. Persily and L. de Jonge: [28]

$$CO_2\ L/s\ = V_{CO2}\ = RQ\ \cdot BMR\ \cdot M\cdot 0.000569$$

Where
$RQ = 0.83$
$M = metabolic\ value$
$BMR = 0.048\cdot m + 3.653$
Where m is body mass in $\boldsymbol{kg}$. 80 kg is chosen
$BMR = 0.048\cdot 80 + 3.653 = 7.493$

$$CO_2\ L/s\ = V_{CO2}\ = 0.83\ \cdot 7.493\ \cdot 6\cdot 0.000569\ \approx 0.0212\ l/s \approx 1.27\ l/min$$

Ratio Air and CO_2 in breath

$$\frac{1.54}{37}\ L/minute \approx 0.041$$
$$\frac{1.27}{33}\ L/minute \approx 0.038$$

Estimation of amount of moist produced by sweating
Calculating the latent heat generated in an hour

$$3600\ \text{s}\cdot 314\ \text{W} \approx 1130\ \text{kJ}$$

Amount of mol vaporized

$$\frac{1130\ kJ}{43\ kJ/mol} \approx 26.3\ mol$$

Converting mol to liters

$$26.3\ mol\cdot 18\frac{g}{mol} \approx 473\ g = 0.473\ l$$

Calculation of draught rate:

$$DR = (34 - t_{air})(v_{air} - 0.05)^{0.62}(0.37 v_{air} Tu + 3.14)$$

Where:
$t_{air} = 20.0$
$v_{air} = 0.3\ m/s$
$Tu = 0.4$

Solving for $\boldsymbol{DR}$ *gives*
$DR = 13.6$

Calculation of temperature gradient.
Equation is acquired from Schiavon et al [34]

$$s=0.157t_p-0.059\frac{CC}{V_{air}}-0.235$$

Where:
$s =$ temperature difference between $0.1\ and\ 1.1\ m$
$t_p = 18\ °C$
$CC =$ total room design cooling road
$V_{air} =$ displacement ventilation flow