POLITECNICO DI TORINO

Collegio di Ingegneria Energetica e Nucleare

Master of Science Course
in Energy and Nuclear Engineering

Master of Science Thesis

The effect of local exhaust and supply air terminal on the mitigation of indoor airborne transmission risk and thermal comfort

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October 2024

Abstract

Nowadays, people spend the majority of their time indoors. It is important to ensure a healthy indoor environment for the occupants as well as their well-being. Moreover, it is important to guarantee clean air to the occupants without compromising their thermal comfort. Generally, the improved IAQ is in contrast with energy-saving measures. For this reason, there is a crescent interest in focusing only on the occupied zone with the application of local exhausting.

The thesis focuses on the effects of the application of local exhaust and the locations of air terminal devices on indoor airborne transmission. The study was conducted in a test chamber at Aalto University, set up as a meeting room. The room hosted 6 occupants, including an infected person simulated by a breathing manikin. The pathogen or virus exhaled by the manikin was simulated by the tracer gas SF_6. Every experiment lasted 3 hours, considering the average occupation time of a meeting room. The configuration studied was in the cooling season with two different heat gains, 36 W/m² and 67 W/m². All the cooling power requested was provided by the cooled supply air, with an indoor set-up temperature of 25.5°C. For each heat gain, the air terminal device-perforated duct was located at three different locations. In two configurations the air was supplied from the floor level, on the window side, or on the corridor side. Also, the performances were studied placing the perforated duct at the height of 1.7 m on the corridor side.

To analyse the IAQ, the contaminant concentration, the contaminant removal efficiency, and the infection risk were calculated. Moreover, the effects on thermal comfort were analysed for each of the 6 occupants. The vertical thermal gradient, the air velocity, the heat removal efficiency, and the draught risk were calculated.

The obtained results showed that the ventilation system studied allows to reach very low infection risk values, in particular supplying from the floor level. The infection risk reached with the lifted configuration is still within the limits, with a risk of 1.5% after 3 hours. The window configuration is the best one for removing pollutants, reaching a CRE of 5.56. On the other hand, the corridor case shows more uniformity of contaminant concentration among the different occupants of the room. However, the higher heat gains often lead to high airflow rates that cause local discomfort. In particular, close to the supply air inlets it is often measured a very high air velocity and draught risk.

Table of contents

1 Introduction

1.1 Background

The importance of studying the indoor environment comes from the need to provide clean air to occupants and guarantee thermal comfort. Indoor environment affects everybody's daily life as, in nowadays society, people spend most of their time indoors due to work, school, leisure, sport, or in houses [1], [2]. Studies show that the amount of time spent in closed spaces accounts for more than 90% of the day [2]. The goals of ventilation systems are to maintain the required comfort, by heating or cooling the environment and controlling the humidity. Clean air is also needed to remove bio-affluents, allergens, and general pollutants [3]. If done efficiently, this prevents the spread of diseases by controlling the viruses and bacteria present in the air. Moreover, considering that closed spaces are usually densely occupied, the exposure to several types of pollutants is high [3]. The ventilation role is then important for people's health and for people's well-being.

The focus on ventilation's role in air quality increased after the recent COVID-19 pandemic, which underlined the need to pay more attention to how diseases spread in closed environments [4]-[7]. A particular focus was on workplaces as well as hospitals and schools. With the pandemic ongoing, intuitions were made in understanding how the virus spreads. It was discovered that the infection spread due to even sharing the indoor space with an infected person, and not only by direct contact [4]. Further studies were then conducted on airborne pathogens: those are the pathogens generated in the respiratory system of the infected person and exhaled in the air as a way of propagation [3]. The public health intervention included vaccination and insulation. However, previous studies that applied mathematical models demonstrated that those measures are not enough to stop the outbreak of a disease in our modern society [3]. This sets the need for new requirements for ventilation and air distribution systems [2], [4], [5], [8]. The efficiency of the air distribution needs to be improved to mitigate the risk of indoor airborne infection [7], [9]. Furthermore, this way of contagion has been demonstrated to be the predominant one for other important respiratory diseases, such as measles, varicella, tuberculosis [3], measles, and chickenpox [10]. This extends the problem further than the pandemic we experienced worldwide.

The study of indoor airborne transmission is multidisciplinary, with the need for knowledge covering different fields. Among those, are engineering and the studying of the built environment and ventilation techniques [3]. It is particularly important to properly design a ventilation system, as an inadequate air distribution system could even enhance airborne transmission [3], [11]. It is noted that poorly ventilated buildings help spread diseases [4]. The impact of good ventilation is powerful on the economic side too, as an insufficient IAQ could lead to absenteeism or reduced productivity of occupants[3].

Moreover, the attention to ventilation systems is increasing lately also due to the importance of de-carbonizing the building sector, in the bigger scheme of mitigating climate change. This sector is responsible for 40% of the overall energy consumption and emits 36% of the total greenhouse gas emissions in the world [12]. Among this energy consumed, a significant fraction is used to maintain the indoor air quality and comfort conditions [2], [11]. Generally, the challenge Is to improve air quality and maintain indoor comfort while decreasing energy consumption [1]. Studying the air distribution methods to guarantee clean air in indoor environments is important from an energy-efficiency point of view. The urgency is to adopt stronger requirements and increase building efficiency, which is a worldwide matter [12]. As a higher ventilation rate could help reduce airborne transmission, it will result in higher energy consumption too [13]. Moreover, a higher ventilation rate is not necessarily the best option [4], as it cannot guarantee a decrease in infection risk [5].

To reduce the energy consumption of the ventilation, one approach is to focus only on the occupied zone of the indoor environment. The occupied zones are usually only a small volume compared to the total volume of the space considered [14]. Ventilation systems could be implemented to guarantee that the standards of comfort and IAQ are respected only in the occupied zone [15]. Controlling only that small part would allow to reduce energy consumption and costs, still assuring the required air quality to the occupants. Moreover, the inhaled air by each occupant is only 1% of the required supplied outdoor air [2]. The average air demand for breathing is around 0.1 l/s per person [16], while the supply air flow rate for offices and meeting rooms is 8 l/s per person [17]. Supplying more clean air than necessary has a higher energy demand and therefore higher costs too. The local approach includes personalised ventilation systems and personalised exhaust, while this thesis will focus on personalised exhaust only.

The approach of controlling only the occupied zone is already used in large places such as theatres. It is also well known in industries in which there could be hazardous pollutants for workers' health [15]. Those pollutants and chemicals need to be captured before they can spread in the air and be inhaled by the occupants [2]. The challenge is nowadays to apply local micro-environment to conventional rooms and offices without sacrificing thermal comfort. Results of previous studies already showed that the indoor climate of micro-environment systems with less energy use was superior to traditional mixed ventilation [14], [18]. The local approach is promising, and it is worthy to be further investigated [5].

Another aspect that must be considered is thermal comfort, which must not be compromised by indoor air quality. Limits and norms on the requirements for thermal comfort are already existing and should be considered in the study of air quality. The limits regard indoor temperature and air velocity values, as well as the vertical thermal gradient. It has been proved that a comfortable indoor environment prevents absenteeism in offices and improves the performance of pupils in schools [1], [4], [19]. Limits are set also to the draught risk, which is hard to avoid with heat gains higher than 60 W/m^2 [2]. For this reason, in this elaborate typical heat gain values will be studied to investigate its impact.

As stated before, the trend in ventilation is going in the direction of creating a micro-environment. From the thermal comfort side, this would allow to satisfy more people creating personalised environments regulated by the single occupant. In general, for energy consumption reduction, indoor air quality improvement, and thermal comfort, there is a need to shift from a uniform approach, standardly used with mixed ventilation, to a personalised environment [2], [20]. The main principles are to remove/reduce pollutants, supply clean air where, when and as much as needed, control the air distribution, and actively involve the occupants in controlling the environment while designing an HVAC system [20].

1.2 Research Questions

The main research questions that motivate this thesis are the following:

- What is the effect of local exhaust combined with a perforated duct on airborne transmission?
- What is the effect of the position of the perforated duct on indoor air quality and thermal comfort?
- What is the effect of typical heat gain levels on the indoor environment?

1.3 Thesis Structure

The thesis, after this introduction, is divided into a literature review and an experimental part. The literature review focuses on previous studies regarding mitigating airborne transmission in indoor environments, and the different applications of local exhaust and perforated ducts as a supply air system. The practical part will present, instead, the analysis conducted in the HVAC thermal chamber at Aalto University, with the aim of analysing the application of local exhaust and the location of a perforated duct air terminal in a simulated meeting room. The methods used and the results are reported in this thesis, concluding with a discussion about it. The elaboration ends with a conclusion, including the limits of this analysis and suggesting further studies and improvements.

2 Literature Review

2.1 Airborne Transmission Mitigation

The mitigation of airborne indoor transmission is a multidisciplinary problem, that can be approached from different fields. Technology and engineering, used for the design of the ventilation, air distribution and air cleaning systems, should work along with the behavioural measures that can be adopted.

2.1.1 Air Cleaning

There are several techniques studied and applied to clean the indoor air, by capturing or killing the present pathogens. Among the most common [3]:

- Dilution: lowering the concentration of pollutants by mixing the indoor air with clean air. It is one of the best-known methods, where it is assumed perfect mixing in practical applications.
- Filtration: it is used to physically remove the particles in the air [9]. It is a fixed solution while directly applying the filters in the Air Handling Unit (AHU). Using filters could be an opportunity to save energy, with the application of room air recirculation. This technique allows to reuse of the already heated or cooled indoor air as supply air, saving energy and reducing the costs. It is proven that under certain conditions, using recirculated air with high-efficiency filters reduced the contaminant concentration similarly to an all-outside air system [13]. On the other hand, adding filters to the ventilation systems still presents challenges. A high-performance filter leads to high-pressure drops, resulting in the use of auxiliary fans or air pumps to maintain the supply air flux. This increases the energy consumption [13], and consequently the costs [9]. Filtration represents a solution only if the filters used have the capacity to capture the particles ranging in the sizes of the pathogen of concern [7]. Moreover, filtration is effective only if maintained or changed properly and regularly [3], [7]. The filter can be of different categories, made of fibrous media under the category of high-efficiency particulate air filters (HEPA), or ultra-low particulate air filters (ULPA). Also, gas-phase air cleaning methods are applied, including adsorbent air filters such as activated carbon. Filters can also well apply to portable devices, that recirculate indoor air.
- Air disinfection: instead of removing harmful pathogens or particles, they are inactivated [9]. This approach can be implemented either in portable air-cleaning devices or in stand-alone solutions installed on the wall. The indoor air recirculates in the inactivating device, to be supplied again in the room when it is cleaned. There are several technologies implemented, and one possibility is the Ultraviolet Germicidal Irradiation (UVGI). The light emitted at a wavelength of 253.7 nm makes the pathogens harmless by damaging their DNA/RNA. Effectiveness depends on the intensity of the light and on the exposure time. The side effects on humans are that it could cause skin erythema and light sensitivity. It has been proven its efficiency under laboratory conditions, with an effect on the reduction of infection risk equal to doubling the ventilation rate [10]. The advantage compared to filters is that no pressure loss is

produced [9]. To be inactivated, viruses and bacteria have to receive a chronic dose of UV. One-pass exposure could not be enough, but recirculating the air can be effective [13]. Another technology is the Photocatalytic Oxidation: a catalyst accelerates the reaction producing radicals that can oxidise pathogens. A possible problem is the secondary reaction of radicals to form secondary chemical species worsening the IAQ level. Other technologies are electrostatic precipitators (ESP), ionizers, photocatalytic oxidation, plasma, and ozone generators. The portable configuration of these technologies may be beneficial in smaller rooms, but they have to be properly designed to actually impact the air quality [21].

2.1.2 Ventilation Strategies

Ventilation is recognized to be one of the principal ways to mitigate airborne indoor transmission. Therefore, its design should be implemented considering its impact on the infection risk. The effects of varying the different parameters involved in the ventilation systems have been widely studied.

Assuring an adequate ventilation rate is necessary to reduce the risk of airborne transmission in confined spaces [22], [23]. Increasing the supply air flow rate is one possible way to ensure that the occupants of the room receive enough clean air. Higher ventilation rates can dilute the contaminated air indoors more rapidly and decrease the risk of cross-infection [24]. The ventilation standards are designed for the perceived air quality and need to be integrated with the airborne transmission considerations after the COVID-19 pandemic [8]. In this context, Buonomano et al. studied the impact of the ventilation rate on the mitigation of indoor airborne transmission [25]. The highest virus transmission rates were discovered in new buildings during the winter season. The reason could be lower ventilation due to high insulation not supported by adequate mechanical ventilation. The study estimated the ventilation rate required to keep the infection risk indoors under safety limits. The case study was represented by a single generic room, simulated in all the possible final uses of an indoor space described by the ASHRAE Standard 62.1. According to the function of the room, the standard indicates the respective occupancy density. The findings show that a higher ventilation rate improves air quality but with the need to find compromises with energy consumption and costs.

A higher ventilation rate increases energy consumption and, moreover, for certain cases, the ideal flow of supply air presents technological challenges [24]. The ventilation system needed could not easily fit into the available space for already existing buildings. The problem regards the comfort as well as the economic feasibility. The existing ventilation systems are designed to work in an economical and energy-efficient way, allowing little increase in the ventilation rate [25]. Larger systems would require higher initial investment costs [25]. Apart from the energy consumption increases and the technical feasibility, supplying more air changes indoor air distribution.

The air distribution must be designed properly, as studies proved that just increasing the air change per hour (ACH) could also help spread the contaminants around [26]. In addition, the increasing supply airflow rate can reduce substantially the risk of long-range exposure, but it is less effective in the case of short-range exposure. This last one depends more on the complex interactions of flows, which have to be carefully designed [20].

The air recirculation should be avoided. It is a way to save energy, as the already-used air does not require to be heated as much as the outdoor air [10]. Therefore, it represents a risk for the IAQ as it could be a way to transport pathogens generated inside the air volume. Pathogens could also reach other closed spaces connected to the same ventilation systems. Filters and disinfections could allow to use of recirculation [9], but require a proper design and maintenance, with the need of periodical cleaning or substitution [10]. In general, the applied filters do not properly remove particles of every size. The recirculation of air is then generally risky, requiring specific filters and high maintenance.

Pressure control: it is a technique used to prevent the spread of contaminants from different indoor confined spaces, controlling their pressure difference. The negative pressure of a room is reached by providing a higher portion of the extract air than the supply air [27]. This approach allows for control of the flow direction, guiding the transport of contaminants [24]. It has proven to be efficient in keeping the place in which the contaminants must be contained slightly under pressure. This assures that the infiltration air direction is from the outside to the contaminated room. Pressure control is particularly used in hospitals, preventing pathogens from migrating from the insulation wards to other patients or workers [5]. Protective environments use positive pressure to resist the entrance of surrounding contaminated air [24], protecting fragile patients inside. Isolation rooms, instead, use negative pressure to host infected people preventing the spreading of diseases to the surrounding environments [24].

2.1.3 Behavioural Measures

As the COVID-19 pandemic showed, there are several behavioural measures that help reduce indoor airborne transmission. Even if it was clear that those measures cannot work alone [3], [25], they could be considered as a help to an improved ventilation system. Moreover, some measures represent useful guidelines for future situations, like the recent pandemic we faced [25].

- Minimise the number of people within the same indoor environment, avoiding overcrowded rooms. This minimises the concentration of pollutant sources as well as the possible exposed people, establishing a safe occupancy number [10]. Many studies report a positive correlation between overcrowding and disease outcomes [7].
- Make occupancy break, leaving the room empty every hour for 10-20 minutes. The concentration of pollutants builds up from the moment the infected person starts breathing in the room, reaching then a steady state. If the sick person leaves the room the pollutant concentration starts to decay. In the case of classrooms, it was shown that asking the students to leave the room during the breaks between lessons decreased the risk of airborne cross-infection by 35% compared to the reference case [25].
- Short room occupation time to prevent the concentration of the pollutants in the room from building up and reaching too high levels. Moreover, while there are no occupants, the room air can be cleaned using additional ventilation equipment that would normally be perceived as too noisy [25].

2.2 Air Distribution Effects

The efficiency of ventilation systems is highly dependent on air distribution, as it affects the spatial and temporal concentrations of pollutants in the room space [3], [28]. In fact, the air movement surrounding the infected person transports around the exhaled particles, in particular the small ones that remain airborne for a longer time [20]. While the goal of supplying clean air to an indoor environment is to guarantee an appropriate level of IAQ, if the air distribution is not properly designed it could also lead to increased infection risks [5]. With fully mixed ventilation, the average concentration of contaminant in the room is considered, which is easier to predict with the generation ratio given. Instead, when the focus on IAQ regards a specific occupied zone, such as with personalised ventilation systems (PV), the contaminant concentrations are different depending on the location. Adequate comfort and IAQ focused in the occupied zone are reached with an appropriate air distribution design.

The air distribution is complicated to predict and design. The different parts of the indoor volume depend on multiple factors requiring detailed studies. The air distribution in the room changes according to the position of the air suppliers, as it is the main factor in air distribution [5], [13], [29], [30]. It is important to place the air diffusers in the relevant location around the room, to guarantee IAQ in the occupied zone in the most efficient way. Moreover, the air flows in the room are affected also by the geometry of the room, the placement of the exhausts, the number of inlets and outlets, the presence of open windows, and the convection flows due to occupants, equipment, or surface conditions [30]. The ventilation systems should be then designed for the specific environment by analysing the airflow pattern. This could be done, for example, by using smoke tests or CFD simulations to visualise air distribution. In addition, a good IAQ has to be achieved without compromising the comfort of the occupant, assuring that the supply air does not create annoying air movement that may cause draught risk.

Many studies in the area of air distribution focus on healthcare facilities, as they contain large proportions of infectious and vulnerable people. In this kind of environment, the ventilation design greatly affects the possibility of contagion [13]. In hospitals, airborne transmission is the second most prevalent cause of contracting a disease for patients [27]. Liu et al. studied the effect of different inlet positions on the contaminant removal efficiency in an isolation ward [29]. Three cases were studied, supplying air to patients' beds from the ceiling, diagonally to the exhaust and on the opposite side of the exhaust next to the bed. The results show that a close supply from above the patient created an air vortex. This flow may keep some contaminants, increasing the infection risks among healthcare workers. The designed airflow path starts from the supply, reaches the contamination source, and ends with the exhaust. The designed airflow direction should provide clean air to the occupant, collect the contaminants, and then be directed to the exhaust.

The effects of different exhaust positions have been investigated by Nielsen et al. [31] in a two-bed hospital ward. The supply air was injected through a downward ventilation system, providing cooler and heavier air from above the patients. The study shows that a high location of the return openings results in a lower cross-infection risk. Moreover, a two-bed hospital ward with downward ventilation was studied by Qian et al. as well [32]. In this case, also the location of the supply openings was changed. A differentiation was made between the different-sized particles. It is reported that a higher exhaust location seems more efficient in removing fine particles. On the other hand, bigger particles are better collected from exhaust placed at the bottom, as large particles are heavier and tend to reach the floor. The optimal position and configuration could then depend on the targeted particles to be removed. Conducting experiments in an isolation room, Cheong et al. concluded that the extract grill should be placed near the infection source [27]. Moreover, the placement of the inlets and outlets should allow the clean supply of air to go from the cleanest zone to the most contaminated one.

The influence of inlets and outlet placement is investigated by Villafruela et al. in isolation rooms for infected patients [33]. The study calculates the capacity to renew air, remove contaminants, and the risk of airborne infection. A configuration with a supply air terminal unit located in the ceiling in front of the bed and air exhaust just above the patient allows a good contaminant change efficiency and a low risk of infection. Moreover, it was registered that the infection risk near the air supply inlets was lower than in the measurement points of the room far away, as shown by other studies too [28]. Choudhary et al. [30] investigated the influence of room geometry, air terminal unit location, and the presence of an open window on the air distribution. In the case study, without changing the other parameters, the location of the supply air terminal unit changed the air distribution. When installed on the end of the room, it allowed the contaminants from one end to travel longer indoor distances before being removed, compared to the case in which is installed at the centre of the ceiling.

Experiments in a full-scale chamber were conducted to investigate the impact of the air distribution method on particle dispersion by Jurelionis et al. [34]. The study was supported by a computational fluid dynamics simulation too. The ceiling diffusers (with one or with four supply directions) and low air supply velocity installed on the wall were compared. The exhaust, for both cases, consisted of in-ceiling diffusers on the opposite side of the chamber. The contaminant was released close to the supply and, in another test, on the opposite side from the supply diffusers. The results of the simulation and of the experiments show that, with a one-way supply, if the source is close to the inlets the droplets are collected in the stream easily. On the other hand, only if the momentum flux of the jet is higher enough, they can reach the ceiling exhaust. The four-way supply transported the particles in a more efficient manner. Instead, the wall-installed supply shows a less uniform distribution of the particles around the room, as those remained elevated and travelled for the whole length of the volume. In conclusion, in this study, it is stated that the risk of infection varies around the room in the different supply configurations. While supplying from the ceiling, reaching a mixed air environment, the risk is higher next to the exhausted. Supplying from the wall, the risk is higher for the receiver seated at lower zones.

It is proven that the location and strength of heat gains, compared to the location of the supply air, has an influence as well on the air distribution pattern around the room [35]. The performance of the air distribution as a function of the influence of cold and warm surfaces changes with the placement and type of air diffusers. It was demonstrated that a cold wall or window, for example, could create downdraft reducing the ventilation efficiency. The polluted air is brought back down to the occupied zone instead of reaching the ceiling exhaust [36]. In the experimental studies conducted by Liang et al. [37], it is shown that heat gains affect air distribution. Even with mixed ventilation, the indoor temperature was not uniform with the presence of different thermal plumes. The region close to the supply air jets in the cooling season presents a lower temperature. The result showed that the temperature is uniform, also with mechanical ventilation, only if the heat gains are uniform as well. It is important to consider the positions and the power of the main, and fixed, heat gains while thinking about the optimal air distribution and the supply and exhaust positions.

Moreover, the inlet position is important for the thermal comfort as well. Higher values of air velocity are registered in proximity to the air supply, with then a higher draught risk for the occupants [30]. Zhao et al. [35] studied experimentally the effect of different air distributions on the thermal comfort of the room occupants. The results change in the different studied cases, according to the simulated weather conditions of summer or winter. In the study, four different air supply methods placed in different zones of the room were compared. The inlets studied were a grille attached to the wall on the opposite side of the window, two displacement ventilation units in the corners, a ceiling diffuser, and a perforated duct in the middle of the room. The distribution of the indoor air was visualized with smoke tests. The results show that with the displacement unit, the air spread efficiently over the occupied zone, without being affected by the thermal load. Instead, the air supplied from the ceiling with the diffuser tends to be more sensitive to thermal plumes moving around the colder wall, forming a circulating pattern. The perforated duct from the ceiling also created an unstable flow. It must be considered that this study was not discussing only the changes in the supply location but also comparing the various technologies. Supplying from the ceiling gave a more uniform pattern while the heat gain was not present.

While supplying from the floor the air at a lower temperature can be lifted by the thermal plumes. The heated air manages to carry the pollutants up to the ceiling, in which generally the exhaust is placed [36]. The temperature was lower than the setpoint at the ankles' height with the inlets at the floor level. The air velocity also is higher closer to the supply, which results in less comfort depending on where it is placed. For this reason, comfort can be compromised while the clean air is supplied from the floor level. As the air is colder, the possibility of experiencing draught risk at the ankle level increases. In addition, the vertical temperature gradient could be a problem as it is a critical factor for comfort [36]. This effect is more avoidable in the case of higher spaces, as the gradient experienced by the people in the occupied zone is then minimal. The challenge is to supply from the floor also in regular size rooms such as offices or classrooms, without compromising thermal comfort. Based on the results, the air distribution has a strong impact on the local air temperature measured, the velocity, and consequently the draught risk [35].

2.3 Personalised Ventilation (PV)

Requirements for good IAQ are important in the occupied zone, in which the air inhaled by occupants must not be risky from an infection point of view. Typically, the occupied zone is a small fraction of the total volume in a room [14]. Creating micro-environments could then allow to meet the ventilation requisites and optimize the ventilation use of energy [14]. With personalised ventilation systems, as the clean air is supplied directly to the person, the individual perceived IAQ is better [3]. It was also proven that the possibility to control their own occupied zone increased people's comfort [14], as the temperature, the velocity, and the direction of the flow are personalised by the single occupant [3]. However, the system works only if the occupant is seated and is occupying only the designed micro-environment, not including possible movements that may spread the pathogens [3]. The technique of PV is then suitable for offices, classrooms, as well as meeting rooms.

Moreover, studies showed that changing the approach from a whole mixing system to ventilation focused on the occupied zone not only is energy convenient, but would also allow for improved IAQ [24], [38]. In an office, personalised ventilation and displacement ventilation were compared with the standard mixing ventilation. The numerical results show that the infection risk is the lowest with PV. Respectively, the infection risk calculated after 4 hours for the three ventilation systems is 0.19%, 0.39%, and 2.94% [38]. Focus on the occupants could be achieved using local ventilation systems, on the supply side or on the exhaust side (or both). This thesis will mainly focus on the application of the local exhaust.

The application of local exhaust could improve indoor air quality and reduce the energy consumption of ventilation systems. The application of this method is well-known in industry facilities [39]. In recent years, studies have focused on implementing local exhaust in common public buildings as well, such as offices or meeting rooms. Santarpia et al. measured air samples positivity to COVID-19 [10]. The 63% of samples in hospitals, including patient rooms and hallways, were positive, suggesting a dilution effect when sampling close to the patient. This dilution effect shows the potential of local exhaust as the droplets containing pathogens are closed to the exhaling infected person. The application of local exhaust would be safer in energy and efficiently remove pathogens before letting them spread around the full volume.

Lui et al. showed that mixing techniques do not appear to be as effective as local ventilation in reducing the concentration of aerosol particles in the breathing zone in an isolation ward [29]. Hospital ward ventilation was investigated by Bivolarova et al. as well [26], with the aim of improving air quality while reducing the ventilation rate. The innovative system consisted of suction in the mattress of one patient, directly connected to the exhausting to evacuate bio-affluents before they spread. The clean air was supplied from the ceiling, mixing with the total air volume. The performance was assessed by measuring the pollutant concentration in the occupied zone, consisting of the patient and doctor breathing zones. The performance with the bed integrated system results in better supplying under 1.5 ACH, than without the bed at even higher ventilation rates as 3 ACH and 6 ACH. This leads to improved IAQ, and to energy savings at the same time [26]. Ahmed et al. studied local exhaust ventilation in a two-person office, allowing to reduction of the contaminant concentration up to 61% and improving thermal comfort in the occupied zone [40]. Furthermore, the study focused on the reduction of energy consumption as well, which could reach up to 30%.

The advantages of the application of local exhaust were investigated in aircraft too. In general, the present-day ventilation systems in aircraft use mixing principles without protecting adequately the passengers from airborne transmitted infections [41]. It is necessary to implement systems that supply each passenger with their own ration. Dygert et al. study shows a reduction of passenger exposure of 30-55% to the tracer gas, compared to conventional aircraft. This was achieved by incorporating round orifices in the seat behind the passenger above the shoulders [42]. A similar result was reached considering removing only half of the airflow from these orifices and half through the general exhaust. An even higher reduction (from 65% to 90%) was achievable by exhausting all the airflow through the local system.

Moreover, apart from the effect of personalised exhausting (PE) alone, local exhaust can also increase the performance of the PV systems. Junjing et al. studied the different ventilation performance between a PV system alone and the same system integrated with PE [43]. The PV directly supplies the occupants of a simulated office environment. The results show that the integrated system allows to provide more clean air into the breathing zone. Moreover, the removal efficiency was improved in a wider zone of the room volume. This allows to have good IAQ also around the occupant when it moves, which is one of the challenges of the personal ventilation systems.

3 Methods

The experiments analyse the efficiency of pollutant removal for different configurations of the room's ventilation system to reduce the risk of airborne transmission. The cases studied analyse a working environment, particularly a meeting room, occupied by a total of six people. Three people are seated on each side of the table: the one in the middle of the left side is an infected person. The quality of the air in the breathing zones of the five exposed persons and the risk of being infected are analysed. Along with the air quality analysis, also the thermal comfort of the six-room occupants is investigated. The different configurations are compared to study which one is the more suitable to mitigate airborne transmission without compromising the required level of comfort. The thermal chamber is shown in Figure 1.

Figure 1. Test chamber.

3.1 Test chamber

The test chamber is located in a full-scale room, measuring 5.5 m (length) x 3.8 m (width) x 3.2 m (height) with a total floor area of 21 m^2. The external environment is assumed to be stable as the room is located inside a laboratory hall at Aalto University, avoiding the effects of variations in external conditions. The room is adjusted to be slightly over-pressured, at 1 Pa. All these conditions allow us to reach a steady-state condition during the tests. The infected person is simulated by a thermal breathing manikin, whereas the other five exposed occupants of the room are represented with one heated dummy and four heated cylinders. Moreover, other heat gains are contained in the room constantly in every case of study such as the heated floor, two laptops, lights, and a simulated window. The room was set up to work under two typical total heat gains (mid-season and peak load conditions), and the specific contributions to reach those values are shown in Table 1.

Table 1. Heat gains in the test room.

HEAT FLUX	W/m²	67	36
Total heat flux	W	1400	750
Floor area	m²	21	21
Total heat load	W	1400	750
Manikin	W	80	80
Dummy	W	75	75
4 cylinder dummies	W	300	300
2 laptops	W	80	80
Light	W	90	90
7 Simulated window plates	W	310	80
Electric heating foil on the floor (simulated direct solar radiation)	W	420	0
Equipment of manikin	W	45	45

The electric heating foil (5.0 m x 0.8 m) that covers the floor is used to simulate the direct solar radiation on the perimeter zone, represented in Figure 2 by the marked green area. When it is turned on, the floor in that area is at the temperature of 33.6°C. The window is simulated through heated panels on the wall, on the opposite side of the corridor. In the panels, heated water circulates at a different temperature level according to the heat gain studied. The panels are represented in red in the room scheme, shown in Figure 2.

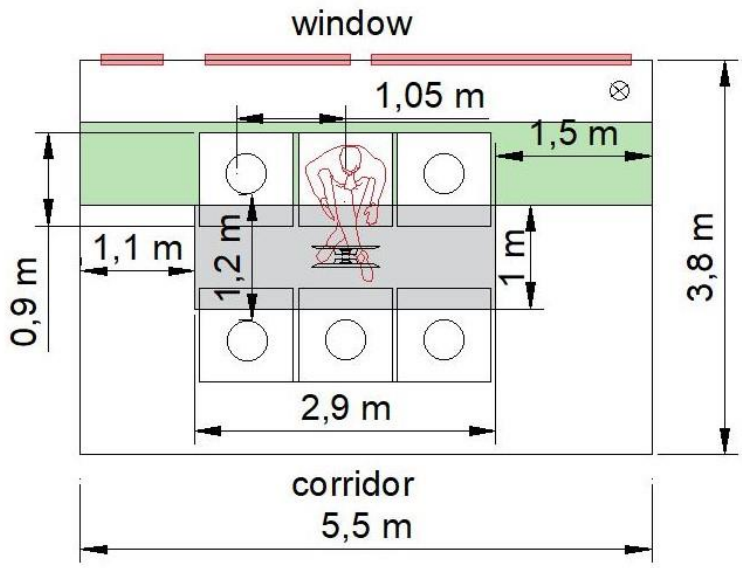

Figure 2. Test chamber configuration and measures, top view.

The table measures 2.9 m x 1 m, and it is placed 1.1 m distant from the entrance door. The occupants' breathing zones, and so the point in which the air quality is measured, are placed at a 1.05 m distance from the ones on the same side of the table. The distance between the ones seated on the corridor side and the ones seated on the window side is instead 1.2 m. In Figure 2, the infected manikin is marked in red colour, collocated in every case studied in the same position, on the window side of the room. Above every occupant's breathing zone, there is a 0.9 m squared-shaped local exhaust, as well as a general exhaust terminal in the top right corner of the room on the ceiling.

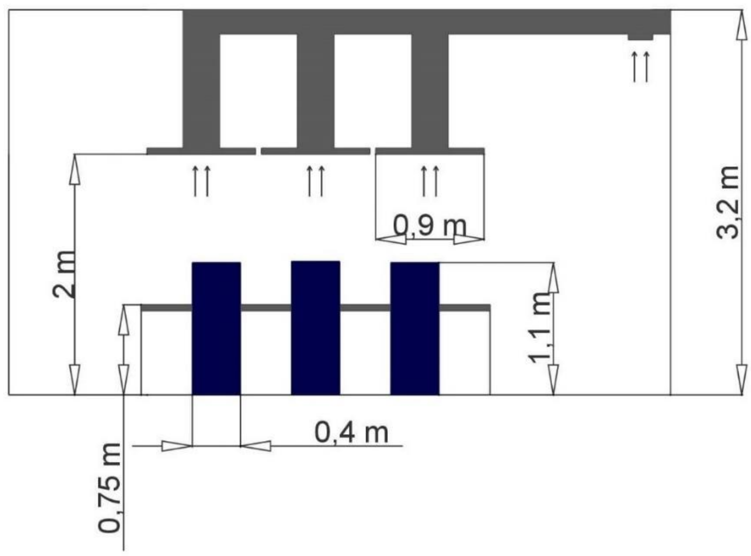

Figure 3. Test chamber configuration and measures, side view.

As shown in Figure 3, the local exhaust is placed at a 2 m distance from the floor level, so 0.3 m over the occupied zone boundary (1.7 m). The breathing zone of the occupants is 1.1 m, and the height of the table is 0.75 m. The diameter of heated dummies is 0.4 m.

3.2 Studied cases

The analysis of indoor air quality and thermal comfort of the room is conducted for six different cases. The supply air system was of a perforated duct, with a length of 5.5 m (as the room) and a diameter of 200 mm. As it is discussed the importance of air distribution on comfort and infection risk [28], it location in the room is changed in three different locations:

attached to the wall on the window side of the floor.
attached to the wall on the corridor side on the floor.
installed at the height of 1,7 m on the wall at the corridor side.

The perforated duct installation is shown in each three positions in Figures 4 and 5.

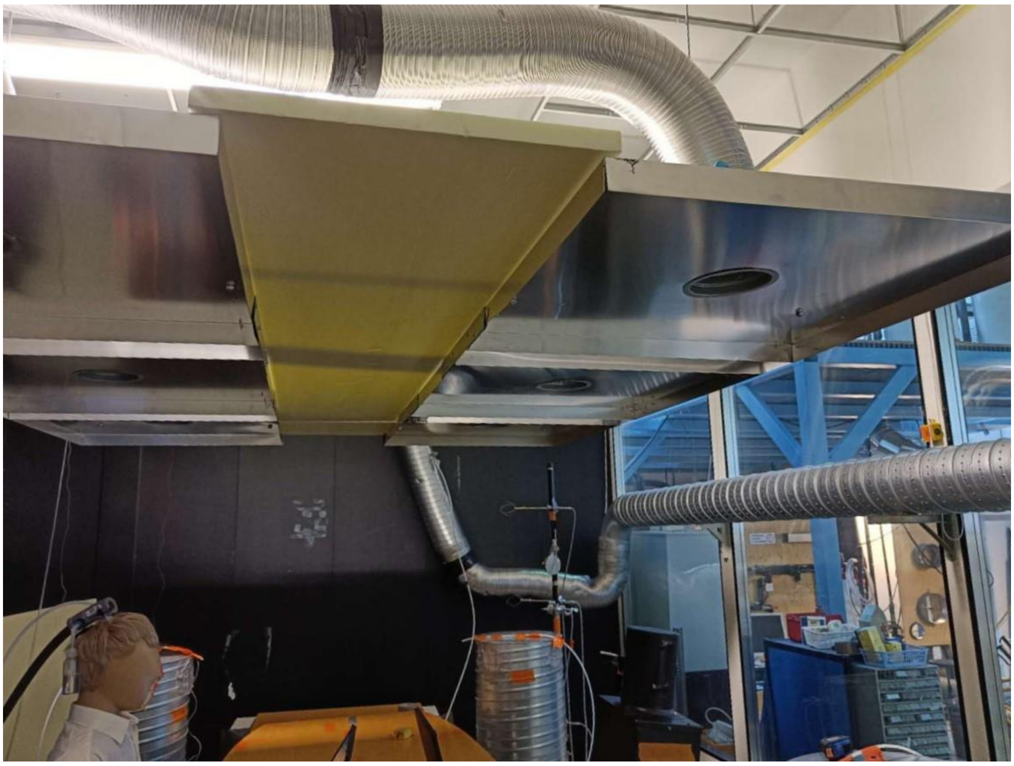

Figure 4. Perforated duct installed on the corridor side wall (simulated window in the laboratory room).

Figure 5. A perforated duct installed at the floor level on the simulated window side (left) and on the corridor side (right).

All these configurations are studied with two different heat gains, obtaining a total of six cases of study:

- Low heat gain (LHG) equal to 36 W/m^2.
- High heat gain (HHG) equal to 67 W/m^2.

These values simulate different indoor conditions and are obtained by changing the temperature of the simulation window. The surface temperature of the window is set to 30°C for the LHG case and to 40°C for the HHG one, respectively. Moreover, in the case of HHG the electric heating foil is used to reach higher heat gain. The other experiment parameters are shown in Table 2.

Table 2. Experiments parameters.

HEAT FLUX	W/m²	67	36
Supply air flow rate	l/s	117	63
Air change rate	h-1	5.6	3.0
Supply air temperature	°C	16	16
Design room air temperature	°C	25.5	25.5

For a meeting room, the minimum airflow rate should be 8 l/s per person [17]. In this case, the supply air is used for cooling too, which leads to higher values of the required airflow rate than the standard ones. In addition, the airflow rates required change according to the heat gain, as it changes the cooling demand. The clean supplied air is set to have the same temperature of 16°C. In the case of LHG, the supply air flow rate is 117 l/s, whereas for the LHG it reaches 63 l/s. The air exhaust from the local exhaust point is imposed, with values of 5 l/s per person in the LHG and 10 l/s per person in the HHG. The remaining air is removed from the general exhaust, with an airflow that depends on the heat gain. The set point for the internal room air temperature is 25.5°C, according to the set point in Finland for indoor temperature in buildings in summer [44]. The deviation from the setpoint is allowed between the boundaries of 21°C and 26°C. The indoor humidity instead was not monitored in these experiments.

3.3 *Instruments*

SF_6 is used in the experiments as the tracer gas, simulating a contagious particle that needs to be evacuated from the indoor environment. SF_6 was chosen as it is ideal, for its chemistry and dimension, to represent aerosols and viruses [28]. The effectiveness of the ventilation system and the infection risk are investigated based on the measures of the tracer concentration gas in the occupants' breathing zones and in general exhaust. To do that, a breathing manikin was connected to an SF_6 source, shown in Figure 6.

Figure 6. Tracer gas source.

The tracer gas dosing rate is 2 ml/s and the breathing rate of the simulated sick person is 6 l/min, according to the average values [16]. As a consequence, the contaminant concentration of the air it is exhaling is 20000 ppm. The thermal settings of the manikin are programmed considering a situation of comfort for the occupants, as the space is simulating a meeting room so that its activity is 1.2 met. The breathing rhythm is then set to complete this cycle: 2.5 s of inhaling, 1 s of pause, 2.5 s of exhaling and 1 s of pause, for a total of 7 s per breathing cycle. Its intensity is set to 55% with an 85-90% humidity in the manikin. Being humidified, the air mixed with the contaminant exhaled by the manikin reaches the temperature of 35°C.

The manikin has 27 separately heated body segments, and it is dimensioned as a 1.75 m tall man. Those different segments are controlled by a computer program, and as in all the cases investigated the occupants are assumed to be in comfort mode, the skin temperature is always maintained as closest as possible to the skin temperature of a person in comfort in the simulated indoor environment. The temperature of the manikin allows, then, to realistically simulate the occupancy of a person with its heat loss. In addition, the manikin was dressed with a thermal insulation of 0.5 Clo according to the average summer clothes in an indoor working space. The manikin is characterized by nostrils shaped as 44.2 mm^2 rounds each and an ellipsoidal mouth with an area of 113.4 mm^2.

The other instruments used in the experiments are summarized in Table 3.

Table 3. Instruments used.

Variable	Model	Accuracy
Temperature Air velocity Turbulence intensity Draught rate Radiant temperature	Omnidirectional probe 54T33 Draught Probe	Airspeed (v): range 0 - 1.0 m/s Uncertainty: ± 2% or ±0.02 m/s on reference velocity Temperature (t): range 0 – 45°C ± 0.2°C on reference temperature 2 Hz
Operative temperature	ComfortSense temperature 54T38	Uncertainty: ± 0.3°C on reference temperature 2 Hz
Pressure difference	IRIS-200	± 5%
Tracer gas concentration	Gasera ONE Multi-gas Sampler and Monitor	Detection limit: 0.37 ppm Repeatability: 0.08%
Temperature Relative humidity	Tinytag plus 2 TGP-4500	Air temperature ±0.5°C, RH ±3% at 25°C

Surface temperature	ThermaCAMTM P60 infrared camera	±0.02·Tmeas

The composition of the gas inhaled by the dummies is analysed, measuring the tracer gas concentration, by a multi-gas analyser platform, Gasera ONE Multi-gas Sampler and Monitor, shown in Figure 7. The station allows us to analyse the gas composition in CO_2, H_2O, SF_6 and C_7H_8, with a focus only on the SF_6 for the scope of this thesis.

Figure 7. Multi-gas analyser equipment.

The measurement is at the breathing zone of the healthy occupants, so at the height of 1.1 m which is supposed to be the head level. The concentrations in the supply air and in the exhaust air are measured too. The first channel, CH1, analysed the supply air, whereas CH2, CH3, CH4, CH5 and CH6 represented the healthy occupants numbered 1, 2, 3, 4 and 5. The last channel, CH7, corresponds to the exhausted air. The different exposed occupants are numbered from 1 to 5 according to the room scheme, in which also the infected person is represented, as shown in Figure 8.

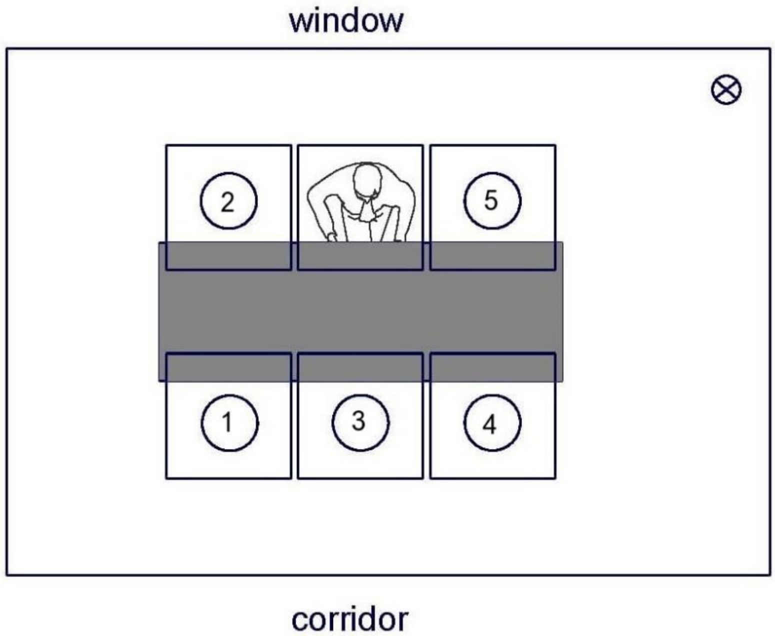

Figure 8. Numbered exposed occupants for indoor quality analysis in the thermal chamber.

In addition, two stations containing hot sphere anemometers, shown in Figure 9, are used to measure local internal temperature, operative temperature, air velocity, turbulence intensity and draught risk. The measurements occur at the heights of 0.1 m, 0.6 m, 1.1 m and 1.7 m. The accuracy of the measurements is ±0.02 m/s referring to the air velocity and ±0.2°C referring to the temperature. The operative temperature, instead, has an accuracy of ±0.3°C.

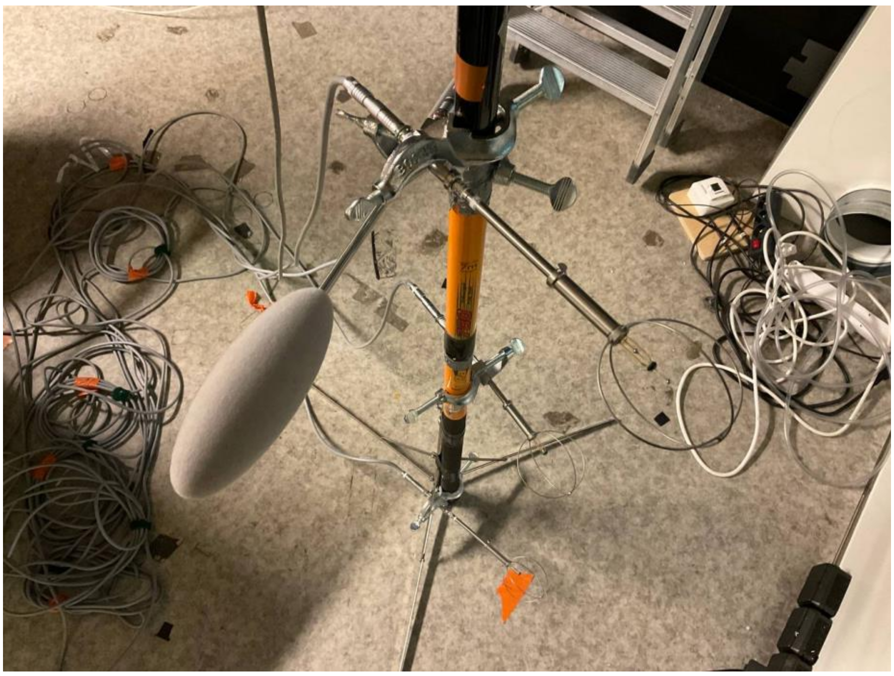

Figure 9. Anemometer measurement pole.

3.4 *Experiment set-up*

The manikin needs to be heated up in advance, and by that, it realistically simulates a person in the room with constant heat loss. All the experiments are conducted under a steady state, so each component needs to be in equilibrium. To do so, between each different experiment, at least one night was waited to ensure a complete air change. The measurements can start with no residual concentration of pollutants due to the previous tests. The multi-gas analyser platform also needs to be heated up before, 30 min in advance. The SF_6 source is open only after the thermal environment steady state is reached, and the experiment can start.

The length of the experiment is set at 3 hours considering an average time of occupancy in a meeting room. The output file obtained shows each data with an 8-minute time step: in fact, every location is subsequently measured for 1.2 min for a total of 22 measurements in the 3 hours.

At the end of the experiment, the source of gas is closed, and the thermal comfort is investigated. The parameters related to thermal comfort are measured in all the occupant's six positions, including the infected person. The measurements are done by placing, after the end of the experiment and the collection of the indoor air quality data, the anemometer stations close to every occupant's seat, behind their back, for 10 minutes. The six occupants of the room are numbered as shown in Figure 10.

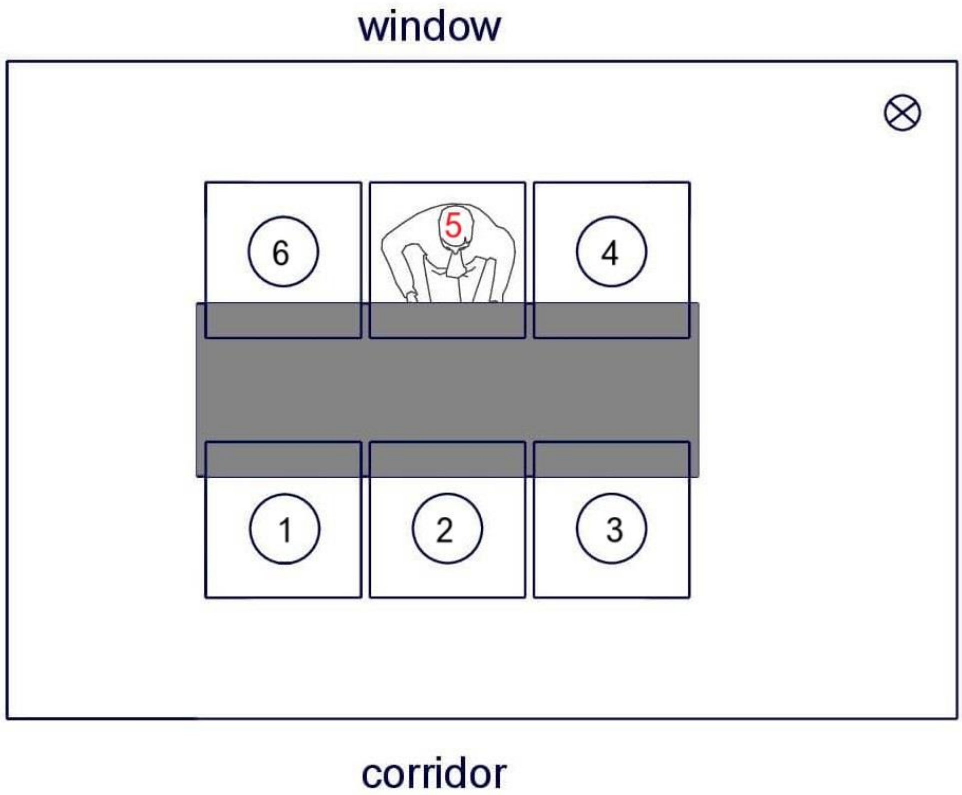

Figure 10. Occupant room scheme for thermal comfort analysis.

3.5 Evaluation Indices

3.5.1 Indoor Air Quality

To evaluate the air quality, it is necessary to monitor the SF_6 concentration in the occupants' breathing zone with time. After the experiment starts, the manikin starts exhaling contaminants in the indoor air. The system measures the value in ppm in different positions, including the five breathing zones of the exposed persons, the exhausted air, and the supply air, for a total of 7 different sets of data. The concentration of SF_6 in the supply air is then subtracted from the values measured in the breathing zones. This eliminates the background amount of SF_6, as the scope of this study is to evaluate the pollutants released to the environment by the manikin breathing. The experiments show two phases: in the first part the concentration of tracer gas starts to build up from zero as the manikin starts breathing, while in the second part, a steady state is reached.

The standard deviation of the contaminant concentration measured is also calculated. The following analysis is applicable only to the steady state conditions, it must be determined from which time step it is reached for the different heat gains. The equilibrium is assumed to be reached for every position and every configuration in the same moment for both heat gains. The equilibrium is determined considering the concentration at the exhausted, with a deviation standard of values in that period under 1 ppm for every case. To investigate the changes in the stability of the system regarding the concentration measured at the breathing zone of the occupants, the standard deviation of those values in the period assumed as the steady state is calculated and plotted.

The concentration of SF_6 measured in each position is compared to the one contained in the exhausted gas, to calculate the contaminant removal efficiency (CRE). CRE is determined according to the following formula that applies to the steady state:

$$\varepsilon = \frac{\langle c_{exh} \rangle}{\langle c_i \rangle} \tag{1}$$

where c_i means the concentration of pollutant in the position evaluated and the c_{exh} is the one at the exhausted air. The value c_i is taken considering an average value of the measurements after the steady state is reached. A value of CRE higher than the unit indicates that the system is working properly as the contaminant is being directed to the exhaust and not to the occupants' breathing zones.

The infection risk is evaluated as the probability for healthy occupants to get infected with the pollutants detected in their breathing zones. The calculation is conducted using the Wells-Riley formula [45], considering the quantum infection rate of the pollutant particle analysed. The method is recognized by the National Institute of Health. The formula is:

$$P_D = 1 - e^{-N_{quantum}} \tag{2}$$

where P_D is the airborne infection risk with the exposed person during the given exposure period estimated by the dilution-based estimation method proposed. $N_{quantum}$ is the inhaled quanta by the exposed person during the given exposure period, and it can be calculated as follows.

$$N_{quantum} = \int_0^T p_{exposed} C_{quantum}(t)dt \quad (3)$$

In the calculation, $p_{exposed}$ is the breathing rate of the exposed person (m^3/s), and $C_{quantum}$ is the airborne quantum concentration at the exposed position (quanta/m^3). $C_{quantum}$ can be calculated with the following formula using q, the quantum generation rate (quanta/s). In this analysis, q is assumed to be 5 quanta/s: this value is the quantum generation rate of a COVID-19 infector according to the REHVA COVID-19 guidance.

$$C_{quantum} = \frac{q}{D * p_{infector}} \quad (4)$$

In the formula, $p_{infector}$ is the breathing rate of the infector, whereas D represents the dilution ratio at the exposed position, calculated using the fixed concentration exhaled by the infector and the measured concentration, in each time step, at the breathing zones of the occupants. It is measured at every position as, according to the different ventilation system configurations, the pollutants move in different positions in the room.

$$D = \frac{C_{infector}}{C_{exposed}} \quad (5)$$

Putting the formulas together the final explicit equation is obtained.

$$P_D = 1 - e^{-\int_0^T \frac{q * p_{exposed}}{D(t) * p_{infector}} dt} \quad (6)$$

The change in the infection risk with time is then calculated for each healthy occupant of the room.

3.5.2 Thermal Comfort

At the end of the experiment, after collecting the data for the indoor air quality, the thermal comfort is analysed as well. In this case, all six positions have been analysed including the manikin, numbering the occupants from one to six as shown in the scheme in Figure 10. The parameters related to thermal comfort regard the temperature and the air velocity. The different heights considered for the measures are the ones impacting people's comfort as they reach sensitive parts of the body while occupants are seated:

- m is the ankles level.
- 0.6 m is the waist level.
- m is the head level.
- 1.7 m is the upper boundary of the occupied zone.

In addition, one thermometer at the height of 1.1 m measures the operative temperature, T_{op}, of the room.

The local internal air temperature is measured with the anemometer stations. To assure an adequate level of thermal comfort, it is necessary to investigate the temperature difference between the head and ankles. The EN ISO 7730 sets the limits to the temperature difference to satisfy the criteria belonging to different categories of comfort. Moreover, the room's Top is measured, and it is defined by ISO 7730:2005 as the uniform temperature of an imaginary black enclosure in which the occupant would exchange the same amount of heat by radiation plus convection as in the actual nonuniform environment.

$$T_{op} = \frac{h_r \cdot T_{mr} + h_c \cdot T_a}{h_c + h_r} \tag{7}$$

This leads to having only one unique value all over the room. The vertical gradient of temperature is then plotted for each occupant and configuration, comparing the results with the limit value accepted.

With the indoor temperature data, it is possible to determine the heat removal efficiency, defined as follows.

$$HRE = \frac{T_{ex} - T_{su}}{T_{ave(0.1-1.1)} - T_{su}} \tag{8}$$

The value obtained represents the effectiveness of the ventilation system in removing the excessive heat from the space. The value $T_{ave(0.1\text{-}1.1)}$ represents the average air temperature from the height of 0.1 m to 1.1 m, T_{ex} is the temperature at the exhaust terminal and T_{su} is the supply air temperature. A higher value than the unit indicates that the system is efficiently cooling the occupied environment.

The air velocity is measured at the same time as the indoor temperature. The EN ISO 7730, as for the temperature, defines limits for the velocity gradient at different heights, as well as for the velocity value itself. The velocity vertical gradient is plotted for every occupant in each configuration.

The draught rate is used to evaluate possible local discomfort due to the velocity of air, which may cause unwanted cooling of the human body despite favourable thermal conditions. It is one of the most common causes of discomfort and it leads to the sensation of movement of air. The draught rate is calculated based on the local air temperature, the local mean air velocity, and the turbulence intensity according to the following formula defined by the International Standard EN ISO 7730:2005.

$$DR = (34 - t_{a,l})(v - 0.05)^{0.62}(0.37 \cdot v \cdot Tu + 3.14) \tag{9}$$

Where ta,l is the local mean air temperature, v is the local mean air speed from 0.05 m/s to 0.5 m/s, and Tu is the local turbulence intensity from 10% to 60%. This last value is defined as follows.

$$Tu = \frac{u_{SD}}{\hat{u}} \cdot 100 \tag{10}$$

Where u_{SD} is the fluctuation of velocity and v is the mean air velocity. When the air velocity is under 0.05 m/s the draught rate is assumed to be zero, while in case the air velocity exceeds the upper limit of 0.5 m/s, it is supposed to be 100%. The draught rate values are directly calculated by the measurement stations, and they are reported in tables for each occupant and configuration at the different heights of the room investigated.

4 Indoor Air Quality

4.1 Contaminant concentration distribution

The changing of the concentration of SF_6 with time has been presented for all cases. Its value rises with time as the experiment begins and the manikin starts breathing, reaching the steady state. This balance is reached quicker with a high heat gain set-up (HHG) than with a lower heat gain (LHG), because of a higher airflow rate. The steady state is reached in the LHG and HHG configuration respectively at the time step 7 and time step 5, so after 51 min and 24 min. The results for the different systems are shown in the following graphs, in Figure 11-16.

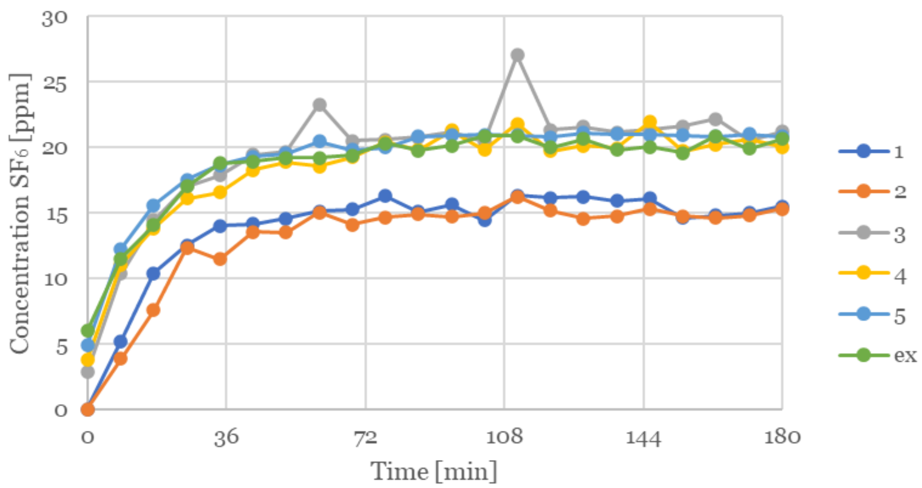

Figure 11. Tracer gas concentration in time in the corridor lift case with LHG (36 W/m^2).

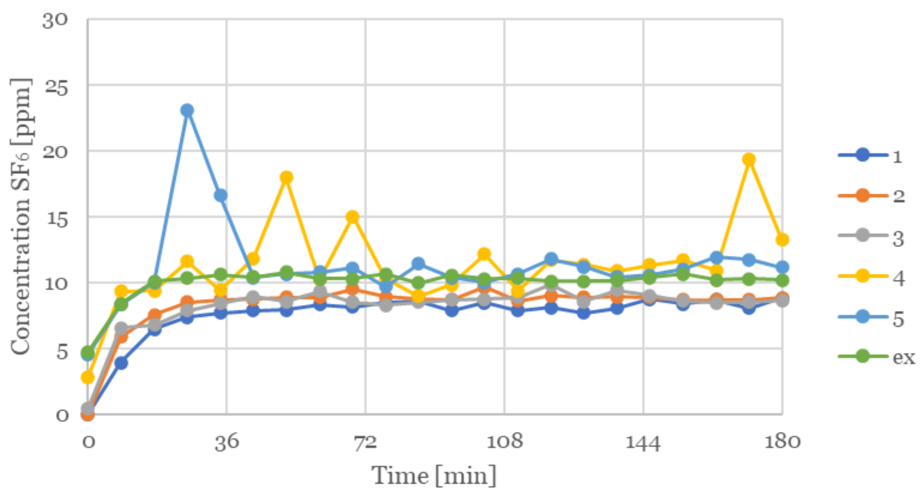

Figure 12. Tracer gas concentration in time in the corridor lift case with HHG (67 W/m^2).

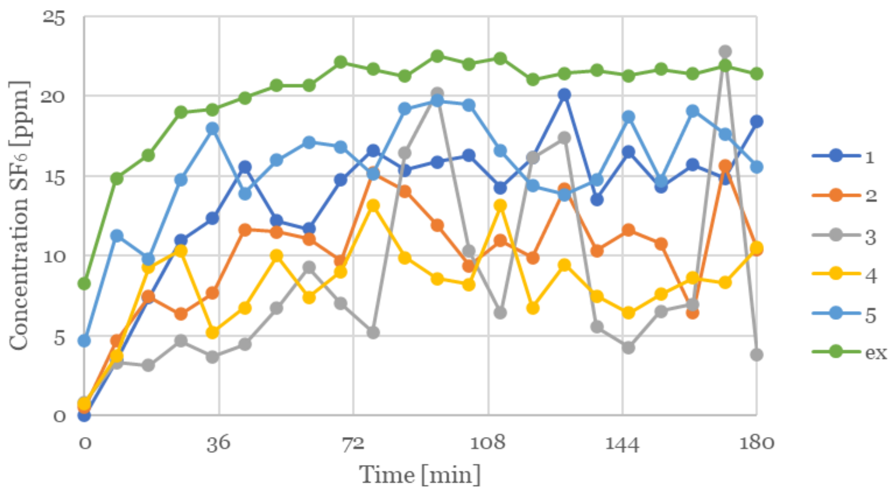

Figure 13. Tracer gas concentration in time in the window case with LHG (36 W/m^2).

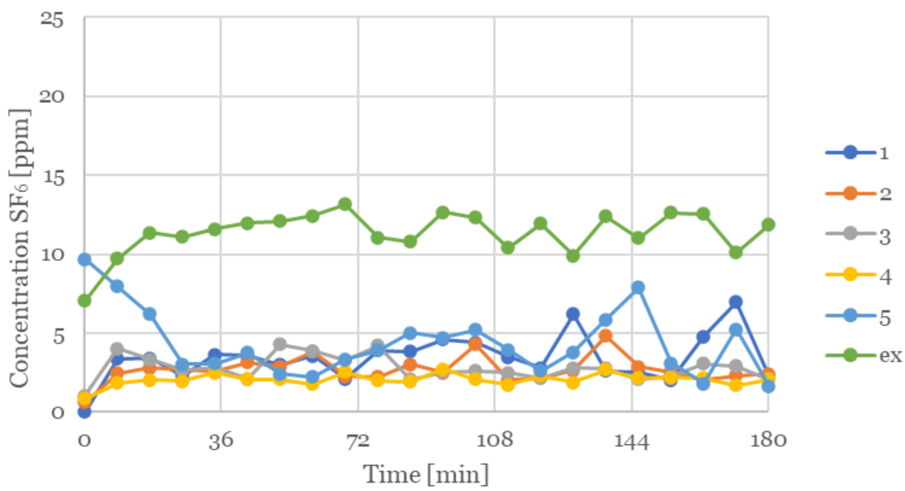

Figure 14. Tracer gas concentration in time in the window case with HHG (67 W/m^2).

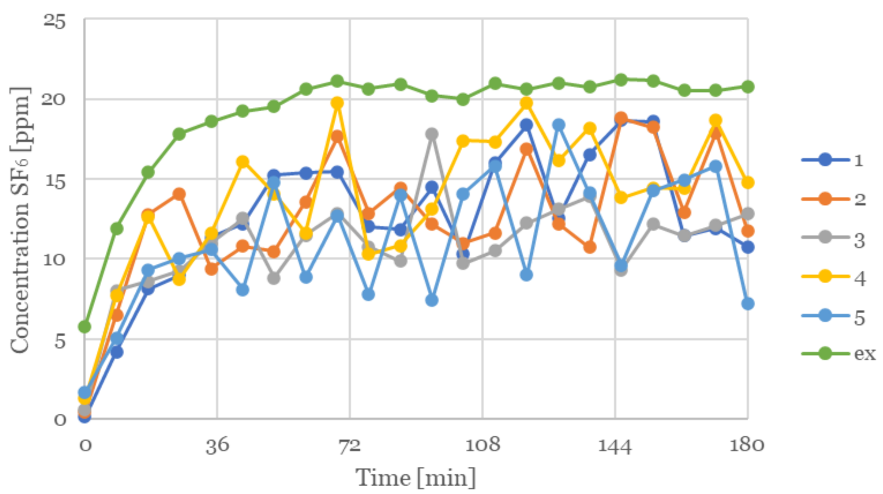

Figure 15. Tracer gas concentration in time in the corridor case with LHG (36 W/m^2).

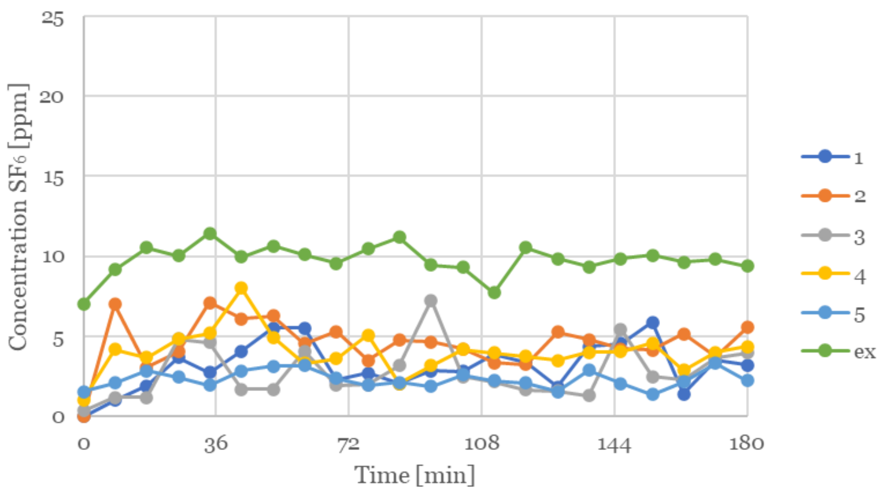

Figure 16. Tracer gas concentration in time in the corridor case with HHG (67 W/m^2).

In every case, the amount of pollutant varies depending on the heat gain, in general reaching lower values with the HHG configuration. The experiment setup considers the cooling season, and therefore a higher heat gain means more cooling power is required. This results in a higher supply air flow rate in the HHG case. The pollutant concentration measured is also sensitive to the perforated duct position and to the occupant's position in the room. This last variable affects the proximity to the general exhausting, to the heat gains such as the simulated window, and to the infected person. The exhaust air concentration reaches the value of 22.5 ppm with an LHG with the supply air on the window side, with similar maximum values around 21 ppm with the other two locations of the perforated duct. Instead, the maximum concentrations of SF_6 measured in the exhaust in the HHG are around 12 ppm (window case). In the HHG configuration, the pollutant concentration has similar values for the different occupants, suggesting better air distribution all over the room.

Not only the heat gain but also the inlet's position affects the contaminant concentration measured. The case with the perforated duct lifted on the corridor side wall leads to higher pollutant levels in positions 3, 4 and 5, as shown in Figures 11 and 12. Those positions represent the occupants seated on the side of the room close to the general exhaust, located in the corner of the ceiling. These high concentrations could be explained by the fact that the airflow coming from the other occupants, next to the door, is directed towards the general exhaust. The flows of pollutants interfere with the occupied zone nearby, showing a non-optimal air distribution. In those positions, the concentration of SF_6 measured is higher in the breathing zone than the one in the exhausted air. This part of the room is disadvantaged with both heat gains studied, while instead in positions 1 and 2 the pollutants level is always inferior to the exhausted one. With both heat gain levels, the position registering higher ppm values is number 5, reaching up to 21.1 ppm (LHG) and 23.1 ppm (LHG). However, this last high value is registered only in one measurement, while all the other points under an HHG are around 11 ppm. This position represents the occupant on the left side of the manikin, in the corner of the table. That location directly faces the corner of the room in which the general exhaust is collocated. Table 4 shows the number of measurements, over the total of 22 par positions, in which the concentration of the pollutants in the breathing zones 3, 4, and 5 are higher than the one in the exhausted.

Table 4. Number of measurements with higher SF_6 in the breathing zone than in the exhausted (corridor lift case).

Heat gain (W/m^2)	Positions		
	3	4	5
36	18 (81.8%)	8 (36.4%)	18 (81.8%)
67	0	14 (63.6%)	14 (63.6%)

The pollutants are higher in the breathing zone than in the exhaust for more than half of the measurements. This part of the room is with the lifted perforated duct the most critical for the air distribution. In particular, point 5 registers too high values of contaminant for 81.8% with the LHG and 63.6% with the HHG. Position 5 is then the worst one for this configuration, as the tracer gas tends to accumulate there.

In the window case, the uniformity of the concentration in the different positions is reached only with a higher heat gain, as reported in Figures 13 and 14. Instead, in the LHG case the measurements differ a lot. In particular, position 5 results in having higher values of SF_6 in the LHG compared to the other occupants' locations. This shows again that too much exhausting flow is directed in that breathing zone. The maximum SF_6 concentration is registered in that position, with a value of 9.66 ppm. The maximum pollutant measured with an HHG instead is 22.79 ppm at position 3. Moreover, the deviation standard of the data with the LHG is higher, showing more flow instability. In both heat gain configurations, only in one measurement did the breathing zone register higher contaminants than the exhaust (consisting of 4.5% of the measurements). This happens in the LHG in position 3 and in the HHG in position 5.

The results of the window case are similar to the ones shown with the perforated duct on the floor but on the other side of the room, next to the corridor. In this last case, the pollutant concentration in the breathing zones is always lower than the exhaust as shown in Figures 15 and 16. While supplying the clean air from the corridor side at the floor level, the distribution of the pollutants is more uniform, and lower values are measured also in positions 3, 4 and 5. The maximum level of SF_6 in the breathing zone is always found in position 4, with values of 19.77 ppm (LHG) and 8.00 ppm (HHG).

The maximum pollution reached in each position is different depending on the supply air location. Figures 17 and 18 show the maximum ppm values measured in each position, respectively for the LHG and the HHG case. This underlines the air distribution effect in the different configurations.

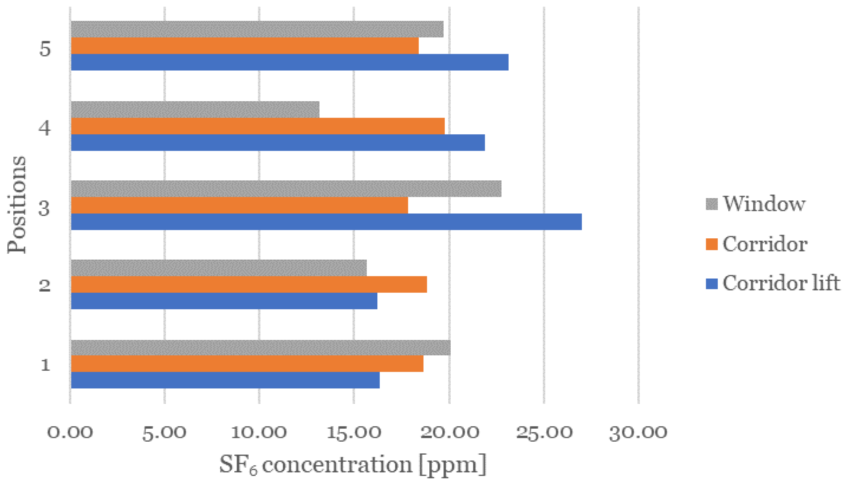

Figure 17. Maximum pollutant concentrations comparison with an LHG (36 W/m^2).

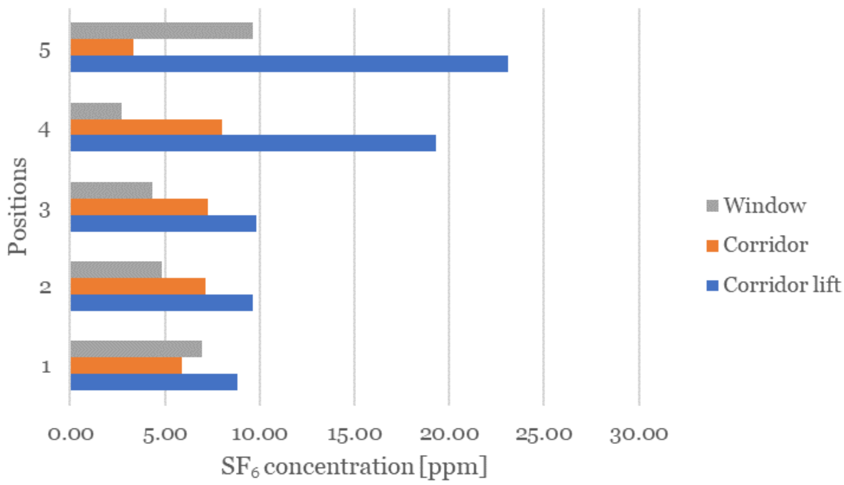

Figure 18. Maximum pollutant concentrations comparison with an HHG (67 W/m^2).

All the cases show how the side of the room close to the general exhausting is disadvantaged from an air distribution point of view. All the systems efficiently eliminate pollutants close to the door but spread those on the other side. The only case that always registers lower values is the corridor side, at the floor level, while the worst is the corridor lift case. This suggests that the air distribution is better achieved while supplying from the floor level, having the general exhaust placed on the ceiling and the local exhaust at the height of 2 m. The two cases with the perforated duct at the floor level reach lower maximum concentrations than the one lifted. With the perforated duct placed on the floor, the clean air is allowed to reach the breathing zone before being exhausted. Instead, while supplying from a higher level (1.7 m), which is exactly the upper limit of the breathing zone, the fresh air is not provided to the occupants as it reaches the exhaust before. Worst performances are registered then while the supply from the floor comes from the window side. This is the side of the room in which the general exhaust is placed, in the corner. It is then possible that the clean air reaches the exhaust before the breathing zone, following the thermal plume of the window. Considering the maximum concentration of pollutants reached, the configuration supplying air from the floor has better performance. The lifted configuration registers concentration of SF_6 even doubled compared to the other two cases.

Moreover, the different configurations reach a different uniformity of contaminants around the room. To compare it, Tables 5 and 6 report the average contaminant concentration measured in the steady state in the different breathing zones. The standard deviation between the positions is also calculated.

Table 5. Average contaminant concentration (ppm) comparison in the different locations and configurations with LHG (36 W/m^2).

System	Positions						Standard Deviation
	1	2	3	4	5	EX	
corridor lift	15.44	14.84	21.53	20.14	20.68	20.08	2.80
corridor	14.35	13.96	11.81	15.29	12.43	20.67	1.27
window	15.40	11.43	10.31	9.03	16.78	21.56	2.99

Table 6. Average contaminant concentration (ppm) comparison in the different locations and configurations with HHG (67 W/m^2).

System	Positions						Standard Deviation
	1	2	3	4	5	EX	
corridor lift	8.21	8.89	8.77	11.97	11.19	10.35	1.49
corridor	3.49	4.78	2.98	4.16	2.34	9.92	0.86
window	3.69	2.81	2.81	2.15	3.87	11.71	0.63

With an LHG, the uniformity in the different configurations is more similar. The perforated duct placed on the floor at the corridor side allows closer concentrations of the pollutants in the different points of the room, with a deviation standard of 1.27 ppm. The corridor lifted and the window case, instead, have a wider variation indicating more unstable distribution around the room, reaching 2.80 ppm and 2.99 ppm. Those two last configurations are the ones more sensitive to the general exhaust location, with the corner of position 5 being more difficult to clean. With an HHG, instead, there is a difference between supplying from the floor level and from a lifted position. The floor-level cases are better at creating a uniform air distribution, with a variation standard inferior to 1 ppm. The different occupants would have similar breathing zones, with close pollutant concentration values. The corridor lift case, instead, has a worse air distribution. The configurations at the floor level are then better for the values of maximum contaminant reached and for its uniformity in the breathing zones. In particular, the corridor case reaches better uniformities in both configurations of heat gains.

4.2 Standard Deviation

To analyse the fluctuation of the data collected, the deviation standard of the value of the SF_6 concentration, once the system reaches equilibrium, is calculated. The range of values measured is shown in Figures 19 and 20 for the corridor lift case, in Figures 21 and 22 for the window case, and in Figures 23 and 24 for the corridor case.

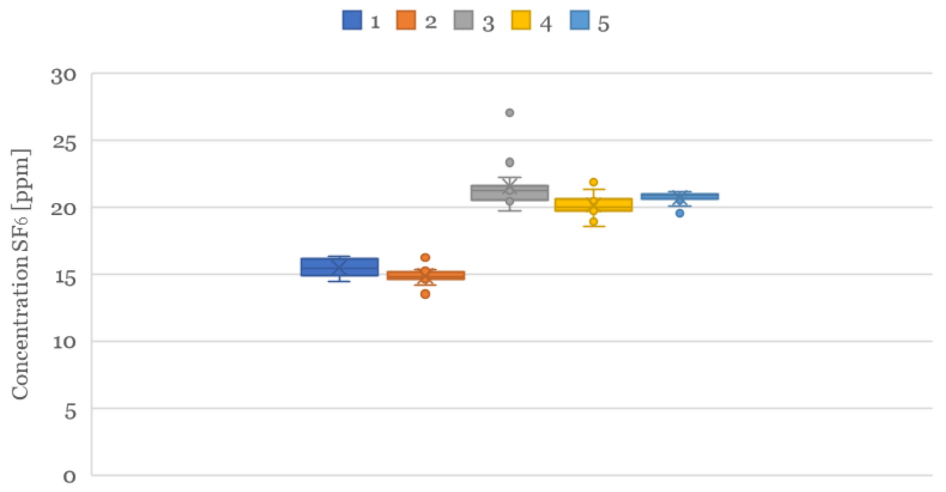

Figure 19. Standard deviation of contaminant concentration values in the corridor lift case with LHG (36 W/m^2).

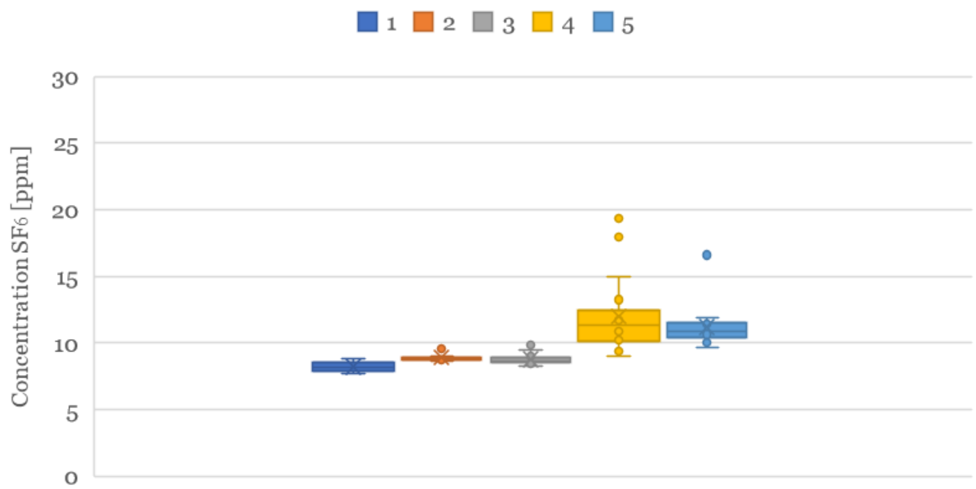

Figure 20. Standard deviation of contaminant concentration values in the corridor lift case with HHG (67 W/m^2).

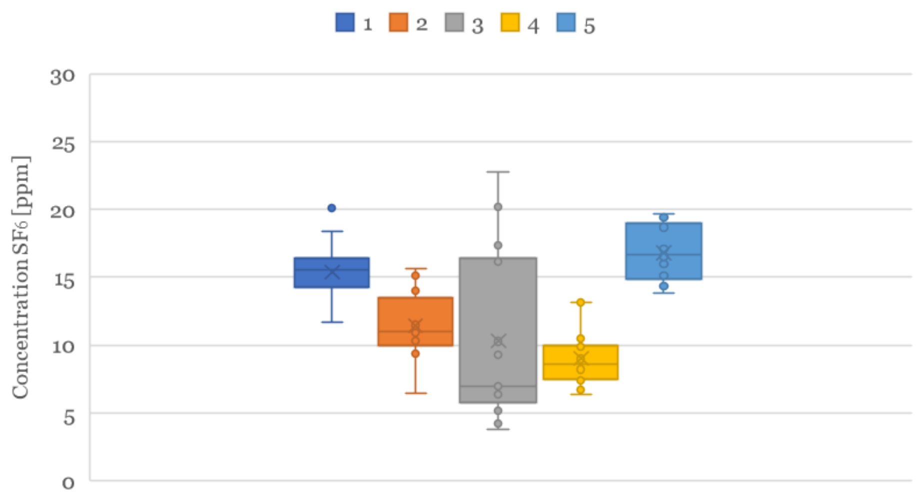

Figure 21. Standard deviation of contaminant concentration values in the window case with LHG (36 W/m^2).

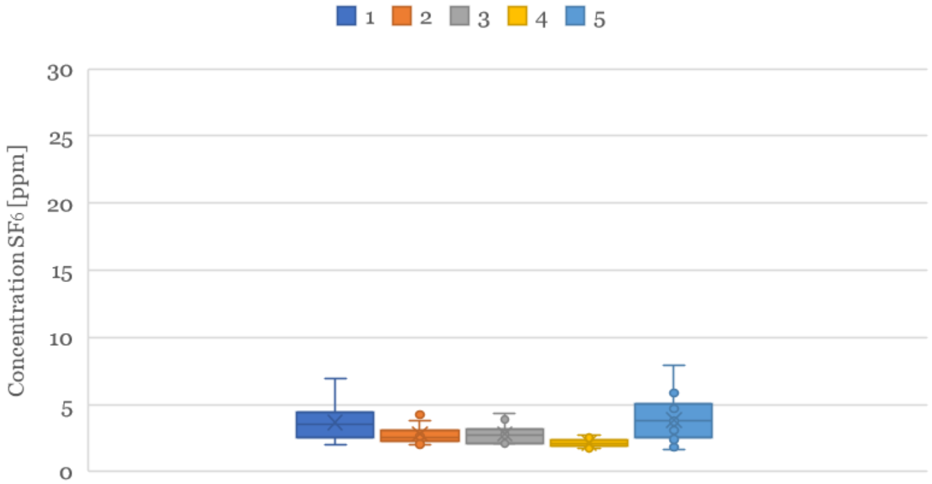

Figure 22. Standard deviation of contaminant concentration values in the window case with HHG (67 W/m^2).

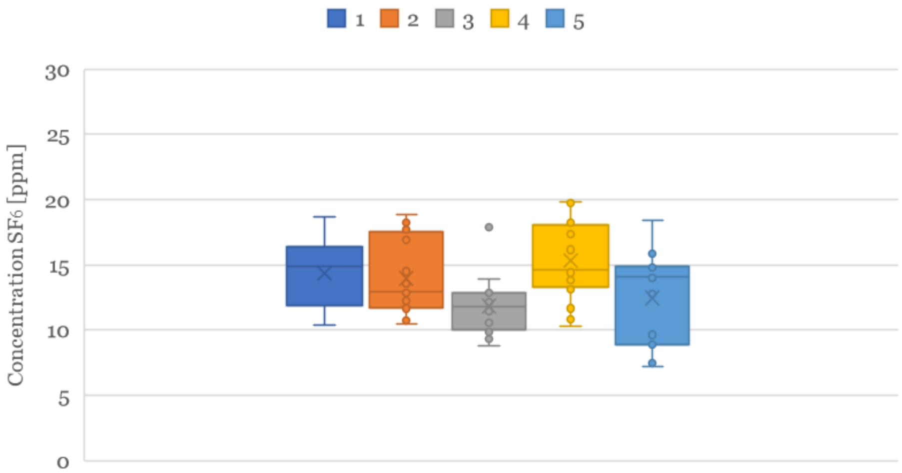

Figure 23. Standard deviation of contaminant concentration values in the corridor case with LHG (36 W/m^2).

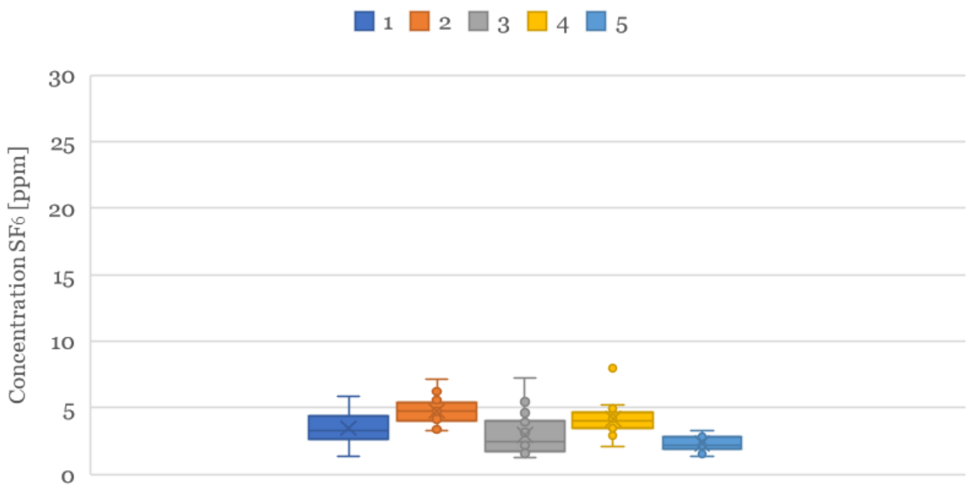

Figure 24. Standard deviation of contaminant concentration values in the corridor case with HHG (67 W/m^2).

The data shows the stability of the pollutant concentrations measured after the steady state for each position. The different heat gain has generally an impact on the variation of data. The increasing of the supply air rate allows to reach a better equilibrium, and therefore more stable measurements. However, the case of the corridor lift seems to be less sensitive to this variable. The measurements of the contaminant are more stable in this case, having low variation also in the LHG configuration. For the window and corridor cases instead, there is a higher instability in the concentration of SF_6 detected at the occupant breathing zone, especially with LHG (36 W/m^2). That shows that with HHG (67 W/m^2) the air flux is more guided, resulting in having a more fixed value of air composition in each part of the room.

For the lifted case, the most critical part of the room was close to the general exhaust, around positions 3, 4 and 5. Here, the highest concentrations of SF_6 were measured. This part of the room is also the one showing more variation of data. The highest standard deviations are measured in point 3 with 1.63 ppm (LHG) and point 4 with 2.75 ppm (HHG).

In comparison, the configurations supplied from the floor have instead higher instability of the measurements done. For the window case, the most critical positions were as well 3, 4 and 5. The highest standard deviations are measured there too, with maximum values of 5.90 ppm in position 3 (LHG) and 1.54 ppm in position 5 (HHG). The corridor case was less sensitive to the general exhaust location, registering a similar amount of pollutants all over the room. Anyway, the higher instabilities of data are still in that corner for this case too. The highest standard deviations calculated are in position 5 with 3.43 ppm (LHG) and in position 3 with 1.55 ppm (HHG).

The configuration of the lifted perforated duct placed on the corridor side reaches higher concentrations of pollutants but shows more stability in the measurements. The cases that supply from the floor have higher standard deviations of the data collected in each position. On the other hand, the window and corridor cases were allowed to maintain the contaminant concentrations lower. Also, the variations are particularly high only in the LHG. All the configurations show that the measurements were less stable on the same side of the room, close to the general exhaust. This seems to be then the most delicate position in every configuration for the air distribution.

4.3 Contaminant Removal Efficiency

The contaminant removal efficiency (CRE) in every position for every different ventilation system configuration is represented in the following graphs. This value is calculated after the steady state is reached.

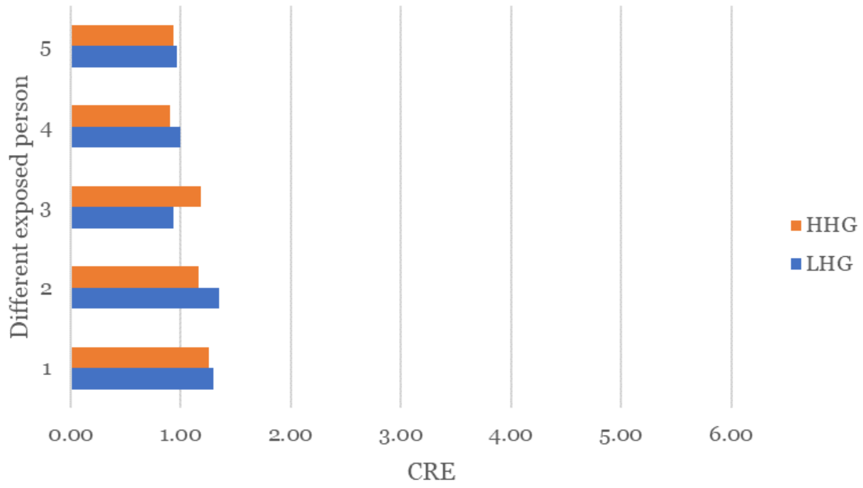

Figure 25. CRE comparison between LHG (36 W/m^2) and HHG (67 W/m^2) for the corridor lift case.

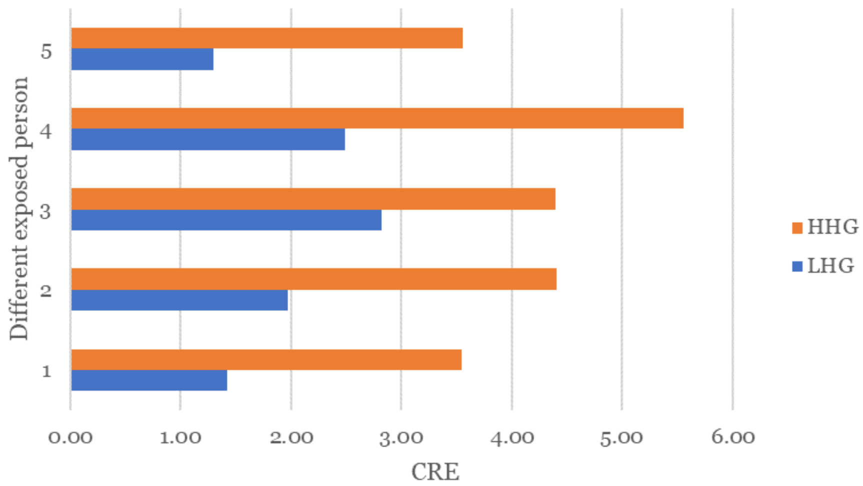

Figure 26. CRE comparison between LHG (36 W/m^2) and HHG (67 W/m^2) for the window case.

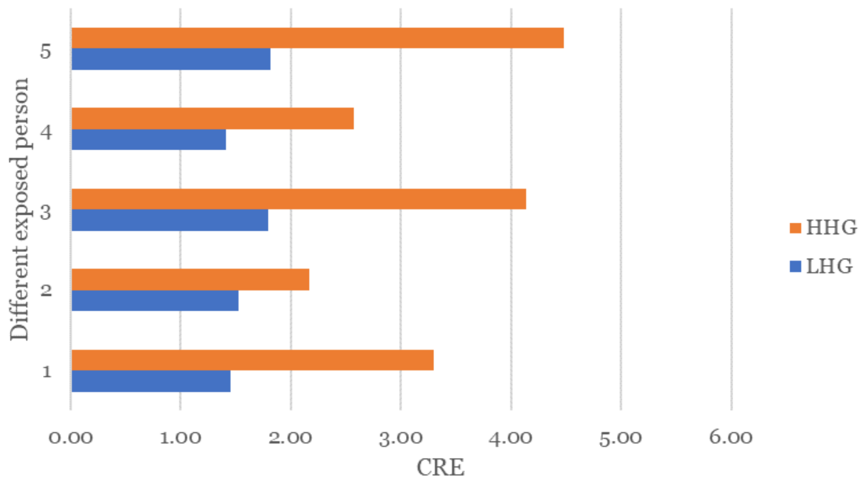

Figure 27. CRE comparison between LHG (36 W/m^2) and HHG (67 W/m^2) for the corridor case.

The measure of the pollutant concentration gives insights into the air distribution of the room, while the efficiency of the system is assessed by the contaminant removal efficiency. This value is measured after the steady state, considering after that in each position an average concentration of contaminant for the rest of the experiment's length. An average value is considered also for the exhausted. From these calculations, it is clear how the supply air position and the supply air flow rate affect the contaminant's removal. Moreover, the values of CRE are different for the different occupants of the room.

The case with the perforated duct placed on the corridor side lifted is less sensitive to airflow rate changes. The values of CRE do not differ significantly from HHG to LHG, as shown in Figure 25. The increasing airflow rate is not beneficial for contaminant removal. Apart from position 3, the other breathing zones present a higher CRE with a lower airflow rate, with a maximum relative difference in position 2 of 14%. Position 3 is the only one benefiting from a higher airflow rate, reaching a CRE of 0.94, a value of 1.18 with an increase of 26%. This could suggest that to clean the indoor air it is not economically and energetically convenient with this ventilation system to increase the air flow rate. The lower CRE with an HHG underlines that the higher air flow rate creates a disturbance in the air distribution. The higher airflow rate is used to cool the environment under a higher cooling load required, so this system would work better when it is not the only cooling system.

Instead, when the air is supplied from the floor level, the CRE is always largely higher than 1. In particular, higher CRE values are reached with an HHG for every position, as shown in Figure 26. This shows that while air is supplied from the floor, a higher air flow rate positively impacts the capacity to remove pollutants from the room. In particular, when the perforated duct is placed next to the window the impact of the airflow rate changing is higher than with the other supply configurations and allows it to reach the highest CRE values. The CRE increase due to higher airflow rate reaches a maximum of 63.5% in position 5, with similar percentages in the other positions. The minimum impact is in position 3, with an increase of 39.5% of CRE. The high difference between LHG and HHG in the window case could be explained by the fact that the heat gain change is experimented varying the simulated window temperature and heating foil on the floor. The supply system is then in a favourable position, with an increased movement of clean air due to the heat from the window panels and heating foil.

The other configuration with the perforated duct on the floor, on the corridor side, shows as well a CRE improvement with the increased air flow rate. The CRE values in all positions for the corridor case are shown in Figure 27. The maximum percentage increase is 59.4%, reached in the position 5. The effect of the airflow rate on the increased CRE is summarized in Table 7, for the cases supplying from the floor level. It is reported the relative CRE difference and the percentage of increasing. However, it has to be noted that this shows the relative difference in pollutant concentration from the breathing zone to the exhaust. This only underlines the removal effects, but the important indicator for the health and the infection risk is the absolute value of contaminant concentration.

Table 7. CRE increasing from LHG to HHG.

Positions	1	2	3	4	5
Window	2.13	2.44	1.58	3.07	2.26
	60.0%	55.3%	35.9%	55.2%	63.5%
Corridor	1.85	0.64	2.34	1.16	2.66
	56.1%	29.5%	56.5%	45.1%	59.4%

The other parameter affecting the CRE measured is the position in the room of the occupant considered. The CRE in the corridor lift HHG case is slightly lower than the unit only in positions 3, 4 and 5. This is coherent with the previous analysis of the concentration of pollutants, which pointed out those as the most disadvantaged positions for the lifted supply system. The minimum CRE values are 0.90, in position 4, and 0.94 in position 5 with a HHG. These results indicate that the air distribution is not optimal in that part of the room, close to the general exhaust. In the other positions, instead, the CRE is always safely over the unit, reaching a maximum of 1.35 in position 2 with LHG.

On the other hand, supplying from the window side at the floor level allows us to reach higher CRE values in every breathing zone simulated. A value of CRE equal to 5.56 is measured in position 4. This occupant's breathing zone is on the corridor side of the room, suggesting that the air distribution is efficient in providing clean air and also far from the supply inlets. Good results are also reached, both for the window and the corridor case, in position 5. This one was indicated before as the most unfavourable one as placed in the same corner of the room as the general exhaust. The results show that the air movement created in the room is appropriate for the application. The minimum CRE with the supply inlets placed on the window side at the floor level is 1.42 (LHG) in position 1, higher than the maximum value reached in the corridor lift case. The higher airflow rate improves the CRE, but the effects of the different positions of the occupants around the room are not affected.

This last aspect is present in the corridor case as well. The high values reached all over the room suggest how the ventilation system is effectively providing fresh air far from the supply inlets too. The maximum CRE calculated is on the other side of the room, close to the window. There, position 5 registers the higher CRE values for both heat gains, respectively 1.82 (LHG) and 4.48 (HHG).

The values of CRE calculated show that supplying from the floor level results in the most efficient way to remove the contaminants. In this configuration, the perforated duct is combined with local exhaust located above the breathing zone (at 2 m). A possible explanation for this higher CRE could be that the pollutants emitted are carried up, reaching the exhaust with the help of thermal plumes instead of spreading in the occupied zone. Moreover, the window and corridor cases also showed lower pollutant concentrations. Not only then they keep the contaminant at a lower level, but they also remove it more efficiently. To values of CRE while supplying from the floor level are often more than doubled compared to the lifted configuration. To underline the difference, Tables 8 and 9 summarise the increases in the percentage of CRE while changing the supply air system from the lifted configuration to the floor one.

Table 8. CRE increases from the corridor lifted configuration to the floor level configurations with an LHG (36 W/m^2).

Positions	1	2	3	4	5
Window	8.4%	31.3%	66.7%	60.0%	25.4%
Corridor	10.3%	11.6%	47.8%	29.3%	46.7%

Table 9. CRE increases from the corridor lifted configuration to the floor level configurations with an HHG (67 W/m^2).

Positions	1	2	3	4	5
Window	64.5%	73.6%	73.2%	83.8%	73.6%
Corridor	61.8%	46.4%	71.5%	65.0%	79.0%

In every position, and with every heat gain studied, the configurations from the floor level perform better. The difference in CRE reaches up to 66.7% (LHG) and 83.8% (HHG). The largest improvements are in positions 3, 4 and 5 as they are, as stated before, the most unfavoured positions for the lifted perforated duct.

4.4 Infection Risk

The infection risk was evaluated for all the cases, with the result in percentage shown in Figures 28-33.

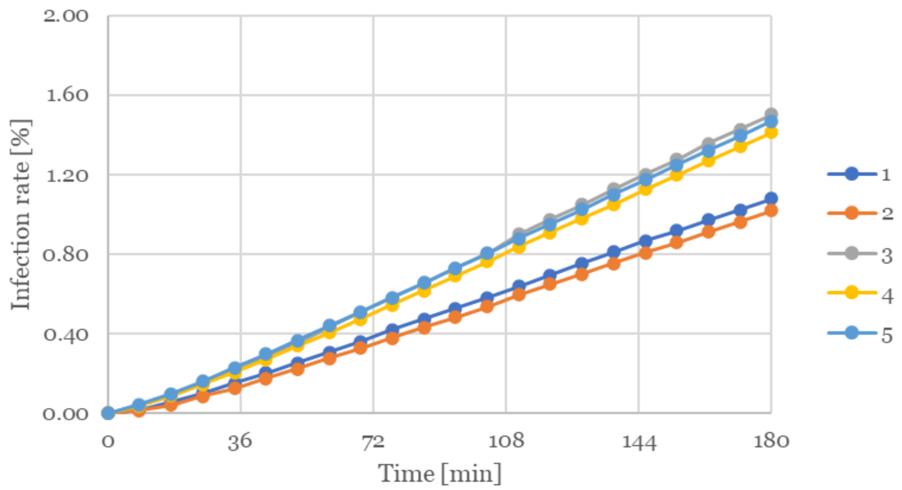

Figure 28. Infection risk increasing with time in corridor lift case with LHG (36 W/m^2).

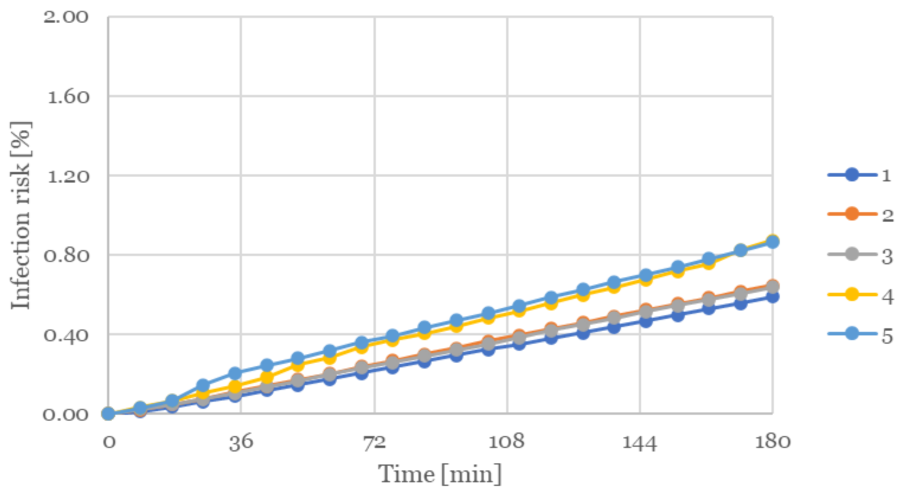

Figure 29. Infection risk increasing with time in corridor lift case with HHG (67 W/m^2).

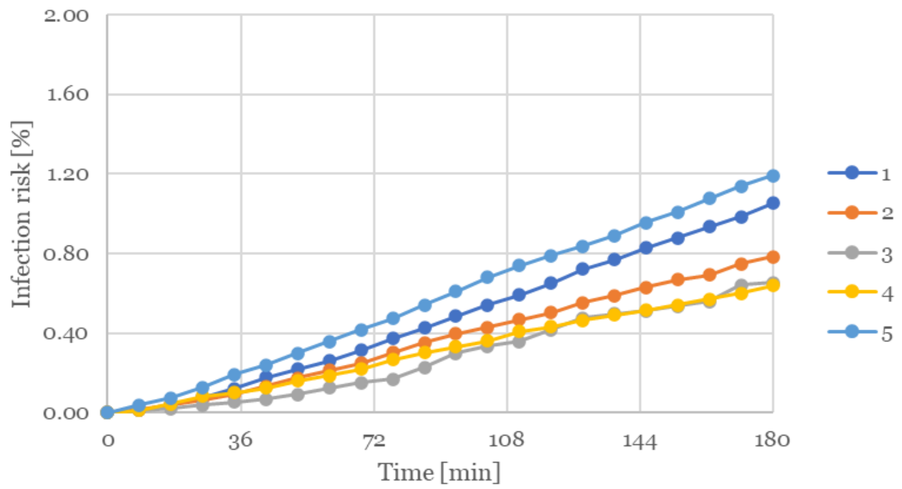

Figure 30. Infection risk increasing with time in window case with LHG (36 W/m^2).

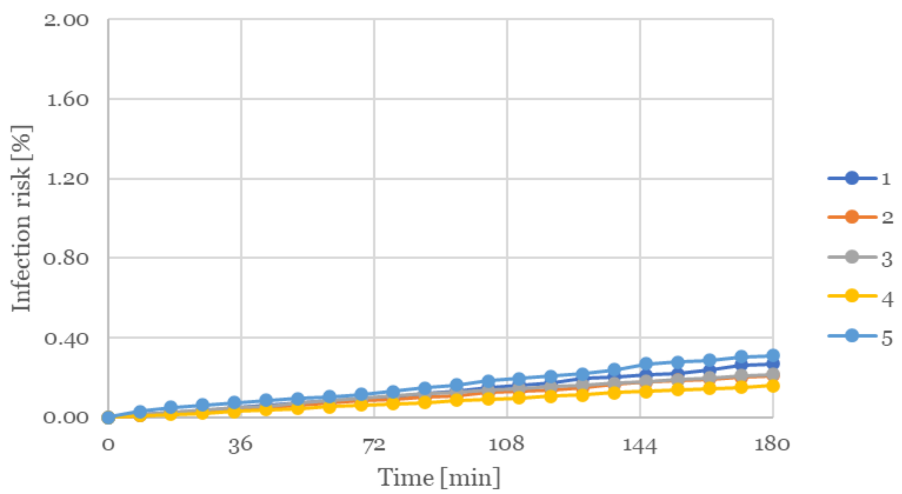

Figure 31. Infection risk increasing with time in window case with HHG (67 W/m^2).

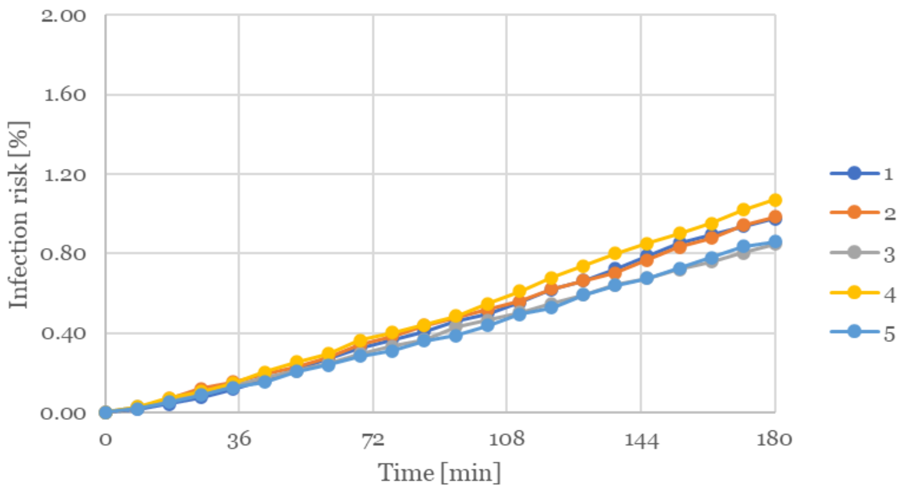

Figure 32. Infection risk increasing with time in corridor case with LHG (36 W/m^2).

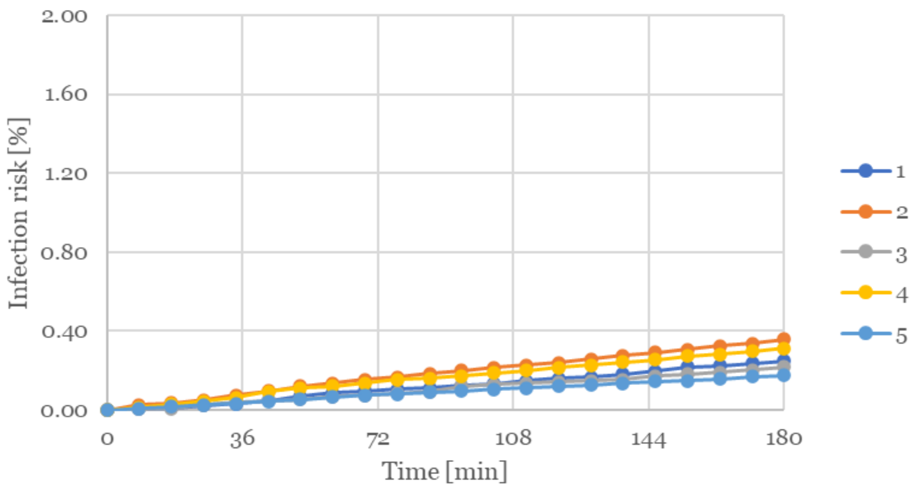

Figure 33. Infection risk increasing with time in corridor case with HHG (67 W/m^2).

The implications on the health of the different configurations are determined by calculating the infection risk. The infection risk starts with zero and increases with time as the manikin exhales SF_6 in the environment. Moreover, the infection risk is a function of the time of exposure, which was 180 minutes. In every configuration, the increase is constant with time, with a linear path.

The airflow rate has a strong impact, as the maximum infection risk values occur always in the LHG case, with a lower air supply flow. This is coherent with the previously measured contaminant concentrations. In the case of a perforated duct lifted on the corridor side, the maximum is 1.5%, as shown in Figure 28, while with an HHG it reaches 0.87%, as in Figure 29. Supplying from the floor level on the window side the infection risk reaches 1.19% with a LHG, in Figure 30, and 0.31% with a HHG, in Figure 31. On the corridor side, the values are similar: respectively 1.07% in Figure 32, and 0.36% in Figure 33. The impact of the higher airflow rate is constant while supplying air from the floor level, ranging from a 73% to a 79% relative reduction of the infection risk. While the perforated duct is on the corridor wall lifted, the air supplied impacts less but still a reduction ranging from 36% to 57% is registered.

The pollutants measured were different across the room, with higher pollutant concentrations measured in the breathing zones close to the general exhaust. The effect of the different positions in the room on the infection risk is shown in Figure 34 (LHG) and Figure 35 (HHG).

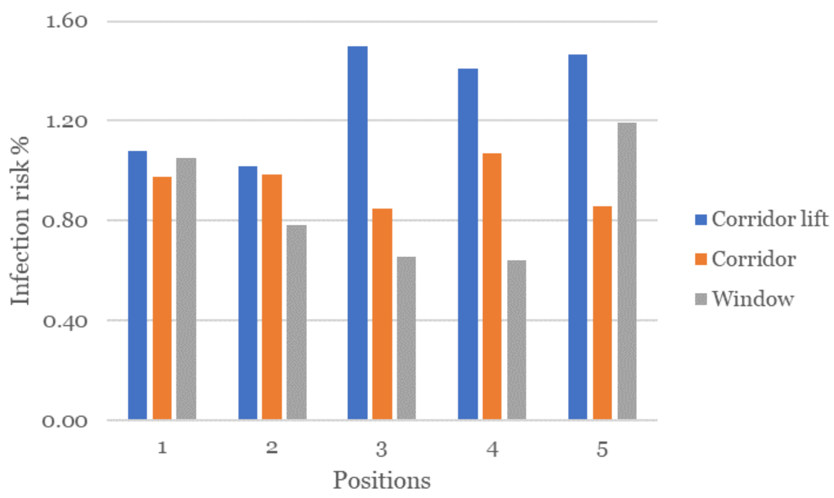

Figure 34. Infection rate in the different positions after 180 minutes with LHG (36 W/m^2).

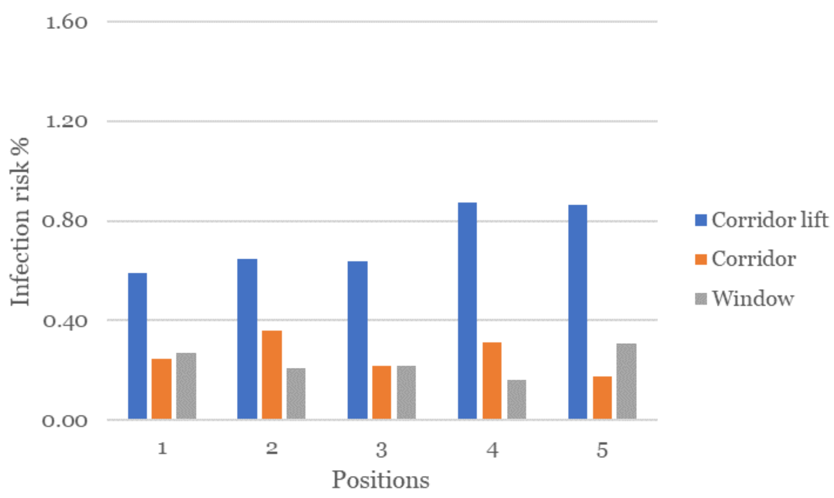

Figure 35. Infection rate in the different positions after 180 minutes with HHG (67 W/m^2).

As noted from the analysis of the pollutant concentration, positions 3, 4 and 5 are the most critical for the air distribution for the corridor lift case, showing higher values of infection risk as well. In general, supplying from a lifted position results in a higher infection risk in every position of the room. This difference with the other configuration is even more remarkable under the heat gain of 67 W/m^2. The configurations that supply from the floor level show less variability from one position to another.

As the infection risk under a steady state increases linearly, it can be extrapolated the infection risk after 8 hours of permanence in the room, which is reported in Table 10.

Table 10. Infection risk estimated after 8 hours.

Infection risk % after 8 hours						
System	Heat gain W/m^2	Positions				
		1	2	3	4	5
Window	36	2.86	2.15	1.83	1.74	3.23
	67	0.72	0.57	0.58	0.43	0.82
Corridor	36	2.69	2.62	2.28	2.88	2.33
	67	0.67	0.94	0.58	0.83	0.47
Corridor lift	36	2.94	2.78	4.08	3.82	3.96
	67	1.59	1.75	1.72	2.29	2.29

Based on REHVA COVID-19 guidance, for the category II ventilation buildings and according to ISO 17772-1:2017 and EN 16798-1:2019, the infection risk should be kept under 5% within 8 hours of occupancy offices [28]. This requirement is fulfilled in all the cases studied and in all the configurations of heat gains, reaching a maximum of 4.08% with the lift system and LHG. All the configurations studied are adequate in mitigating indoor airborne transmission, with better performances while supplying from the floor level.

5 Thermal Comfort

After collecting the data on the indoor air quality, also the temperature and the air velocity are measured, as they are the parameters related to indoor comfort. It is necessary to ensure that the air quality requirements are satisfied without compromising occupants' comfort.

5.1 Temperature

The operative temperature of the room is reported in Table 11.

Table 11. Operative temperature in all cases.

System	Heat gain [W/m^2]	T_{op} [°C]
Window	36	24.3
	67	23.8
Corridor	36	24.7
	67	23.9
Corridor lift	36	23.5
	67	23.4

The measured operative temperatures in all cases studied are in the target range, as the indoor temperature limits are respected. This guarantees the global comfort requirements. The following parameters measured, instead, allow to establish eventual local discomfort. The following figures reported the vertical temperature gradient in every occupied zone with different configurations. The corridor lifted case temperatures are reported in Figures 36 and 37, the window case in Figures 38 and 39, and the corridor case in Figures 40 and 41.

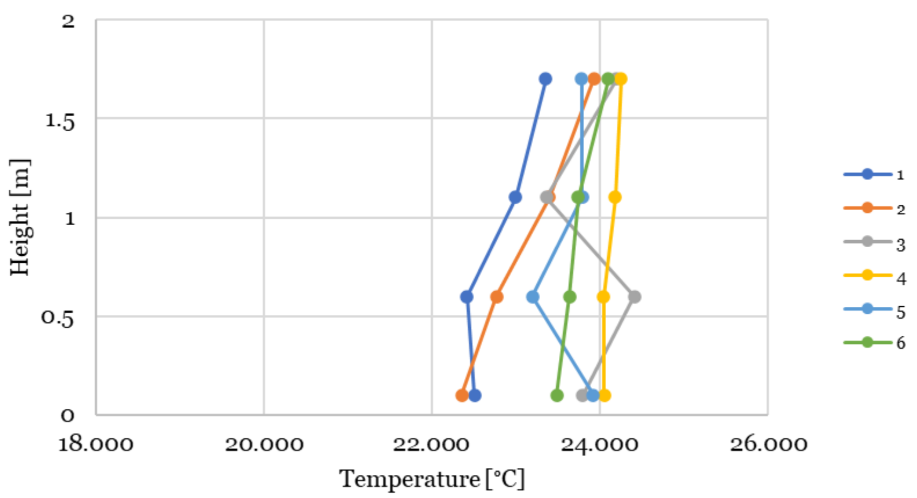

Figure 36. Vertical temperature gradient in the corridor lift case with LHG (36 W/m^2).

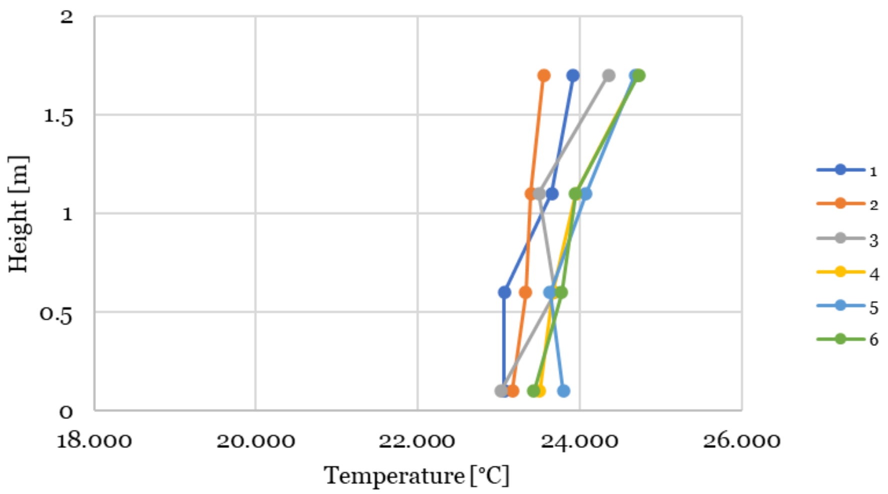

Figure 37. Vertical temperature gradient in the corridor lift case with HHG (67 W/m^2).

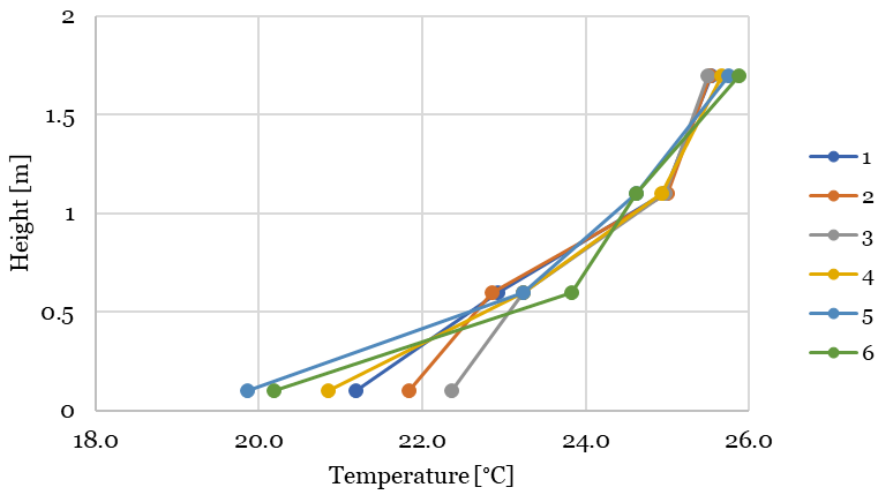

Figure 38. Vertical temperature gradient in the window case with LHG (36 W/m^2).

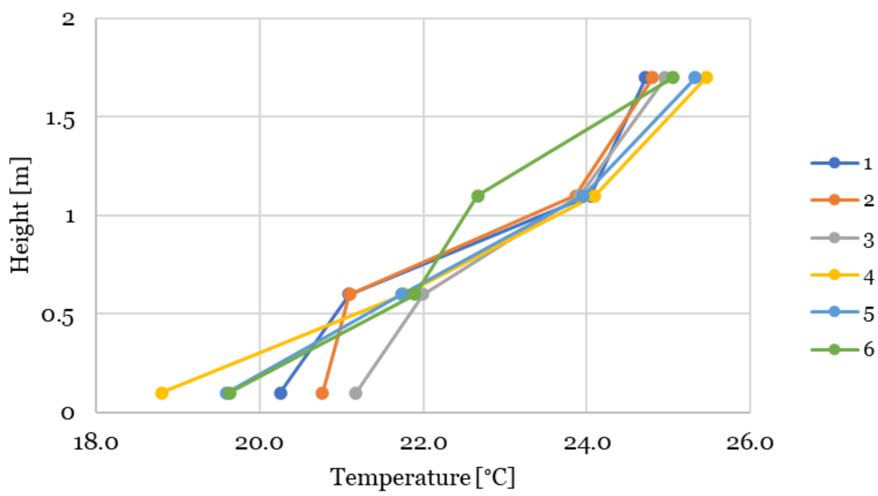

Figure 39. Vertical temperature gradient in the window case with HHG (67 W/m^2).

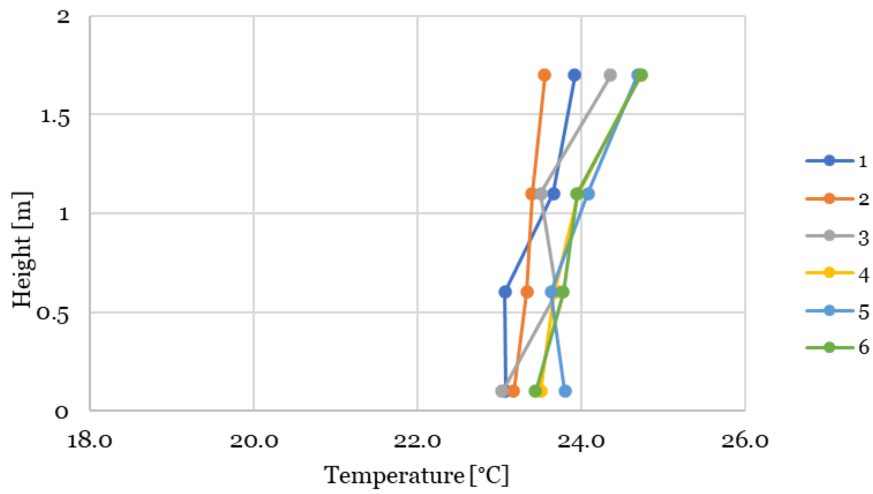

Figure 40. Vertical temperature gradient in the corridor case with LHG (36 W/m^2).

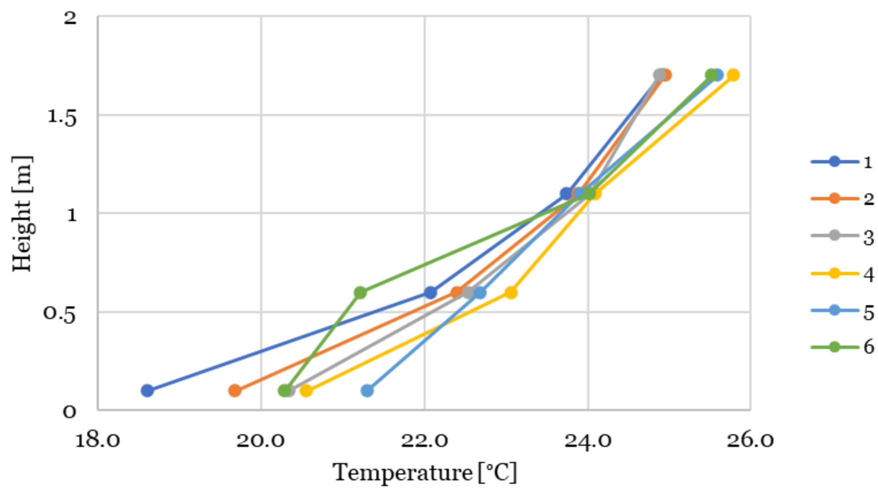

Figure 41. Vertical temperature gradient in the corridor case with HHG (67 W/m^2).

More in detail, to maintain an adequate level of comfort, the difference between the temperature at the head height and the temperature at the ankle level must be controlled. That difference is reported for every configuration studied in Table 12.

Table 12. Temperature difference between head height and ankle height.

$T_{1.1m}$ - $T_{0.1m}$ [°C]

System	Heat gain [W/m^2]	Position					
		1	2	3	4	5	6
Window	36	3.77	3.16	2.62	4.07	4.75	4.43
	67	3.79	3.10	2.74	5.29	4.36	3.03
Corridor	36	5.81	5.97	5.76	3.48	3.33	3.99
	67	5.14	5.27	4.54	5.23	4.30	5.23
Corridor lift	36	0.85	0.38	1.33	1.21	0.89	1.29
	67	0.49	1.04	-0.43	0.14	-0.12	0.25

The EN ISO 7730:2005 sets the limits to the vertical temperature difference to satisfy the criteria belonging to different categories of comfort. The higher comfort standards are set for the category S1, while the requisites decrease to the category S2 and S3. The highest standard is considered only for particular applications in which an individual indoor environment is required. The category S2, which assures a good indoor environment, sets the limits to the common applications. In this particular case of a meeting room, the intermediate standard S2 is applied.

The results show that from a thermal comfort point of view, not all the requirements for the category of comfort S2 are met. For the intermediate category, the vertical air temperature difference between the borders of the occupied zone should not be more than 3°C. This value is exceeded while supplying from the floor in almost every position. The positions exceeding the limit, for every configuration, are underlined in red in Table 12. Supplying from the floor level creates a high vertical temperature difference: from the corridor side, the degree difference reaches a maximum of 5.97°C, which is almost double the limit. In particular, the discomfort is in general perceived when the temperature at an upper occupied zone is higher than the one at lower heights, as in these cases. This could mean that the system is not suitable for this cooling demand, even in the LHG case. This is because the flow of the cold supply air required is too high, causing a strong indoor vertical temperature stratification. Moreover, a higher temperature gradient is perceived in the positions closer to the perforated duct. The ventilation system, with the configuration that places the perforated duct at the floor level, is not suitable to provide the cooling load alone.

On the other hand, while supplying from above the occupied zone, this problem is not faced as the maximum degree difference is 1.33°C. Placing the perforated duct at the height of 1.7 m allows it to meet the requisites for category S1 of comfort as well, as it sets the limit up to a 2°C difference. The data show that the configurations that allow reaching the higher CRE are also the ones presenting the highest temperature difference in the occupied zone. The results underline the difficulties in meeting the ventilation efficiency requirements without compromising the occupants' thermal comfort.

5.2 *Heat Removal Efficiency*

The value of $T_{ave(0.1-1.1)}$ is calculated to determine the heat removal efficiency (HRE). The values are reported in Table 13 for each configuration, with the T_{sup} and T_{ex} used in Table 14.

Table 13. Tave(0.1-1.1) for every configuration.

$T_{ave(0.1\text{-}1.1)}$ °C

System	Heat gain [W/m²]	Position 1	2	3	4	5	6
Window	36	23.07	23.42	23.66	22.89	22.24	22.40
	67	22.15	22.32	22.55	21.45	21.78	21.15
Corridor	36	22.44	22.38	22.46	23.69	23.75	23.51
	67	21.17	21.78	22.18	22.32	22.60	22.15
Corridor lift	36	23.36	23.28	23.27	23.73	23.94	23.69
	67	22.75	22.87	23.58	24.12	23.85	23.61

Table 14. Supply and exhaust temperature for each case.

System	Heat gain [W/m²]	T_{sup}°C	T_{ex}°C
Window	36	15.47	24.15
	67	15.55	24.00
Corridor	36	15.45	25.23
	67	15.47	25.00
Corridor lift	36	15.47	24.00
	67	15.55	24.00

The HRE values obtained according to the previous definition are presented in Table 15.

Table 15. HRE values for every configuration.

HRE							
System	Heat gain [W/m²]	Position					
		1	2	3	4	5	6
Window	36	1.14	1.09	1.06	1.17	1.28	1.25
	67	1.28	1.25	1.21	1.43	1.36	1.51
Corridor	36	1.40	1.41	1.40	1.19	1.18	1.21
	67	1.67	1.51	1.42	1.39	1.34	1.43
Corridor lift	36	1.08	1.09	1.09	1.03	1.01	1.04
	67	1.17	1.15	1.05	0.99	1.02	1.05

An HRE value higher than the unit indicates that there is a temperature gradient between the occupied zone and the exhausted air, meaning that the heat is being removed successfully. Every configuration analysed results efficient in removing the heat, as the HRE calculated exceeds or reaches the unit.

The results show that the heat gain, and then the air flow rate, have an impact on the HRE with every supply system. While placing the perforated duct at the floor level, the system is even more sensitive to the change in the supply air flow rate. The case of the corridor lifted, instead, shows less dependence on this parameter as previously underlined studying the CRE. For the floor level systems, the effect of increased airflow rate is always beneficial for the HRE. The higher values of HRE are reached with an HHG, showing the relative difference between the two cases in Table 16. As specified for the CRE, these values only underline the heat removal efficiency by showing the relative difference between the breathing zone and the exhausting. The important values for comfort are then the absolute values.

Table 16. Increasing HRE due to increased airflow rate.

Positions	1	2	3	4	5	6
Window	0.14	0.16	0.15	0.26	0.07	0.26
	10.8%	12.5%	12.3%	18.3%	5.5%	17.0%
Corridor	0.27	0.10	0.03	0.20	0.16	0.21
	16.2%	6.5%	1.8%	14.7%	11.9%	14.9%

The configurations allowing to reach higher HRE are the ones supplying from the floor level. In both the corridor and the window case, the heat is more efficiently removed for the occupants in the room close to the supply air. In those positions is easier to provide cooled air, even in the window case despite the close heat gain represented by the panels. Positions 4, 5 and 6 present higher HRE than in the corridor case. For positions 1, 2 and 3, as they are located close to the corridor wall, it is the opposite. In the window case, due to its position, more thermal plume of the window is removed directly by the general exhaust with HHG. The flux of clean air from that side helps the air to go up carried by the heat of the window panels. The maximum value of HRE reached, with an LHG, in the window case is 1.28 in position 4 while for the corridor case is equal to 1.41 in position 2. Under an HHG, the window case reaches an HRE of 1.51 in position 5 while the corridor configuration has a value of 1.67 in position 1. The values are similar in each position that is on the same side of the room, depending on the placement of the supply inlet and the heat gain level.

5.3 *Air Velocity*

Another important factor for the occupants' comfort is the air velocity. Its vertical gradient is reported for the corridor lift case in Figures 42 and 43, for the window case in Figures 44 and 45 and for the corridor one in Figures 46 and 47.

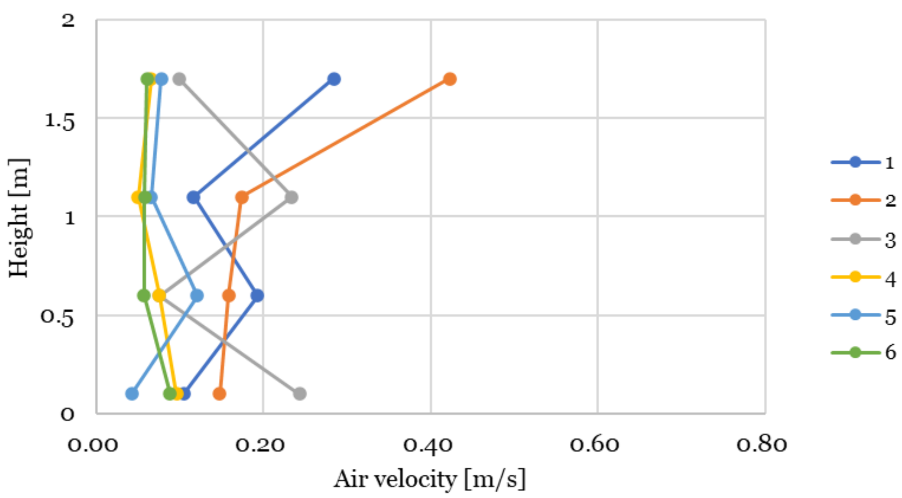

Figure 42. Vertical air velocity distribution for corridor lift case with LHG (36 W/m^2).

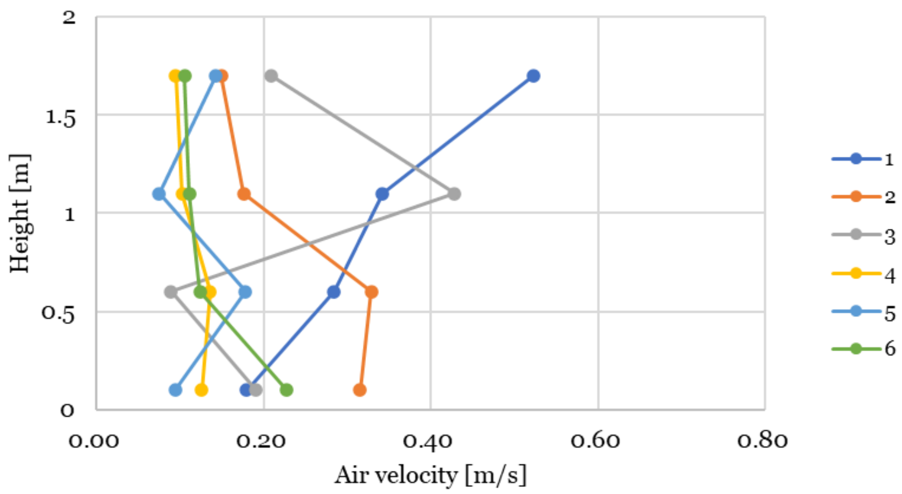

Figure 43. Vertical air velocity distribution for corridor lift case with HHG (67 W/m^2).

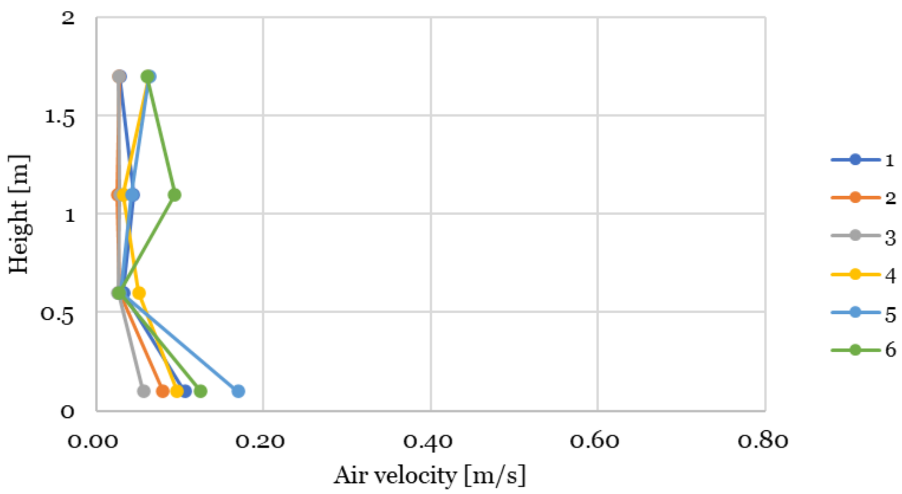

Figure 44. Vertical air velocity distribution for window case with LHG (36 W/m^2).

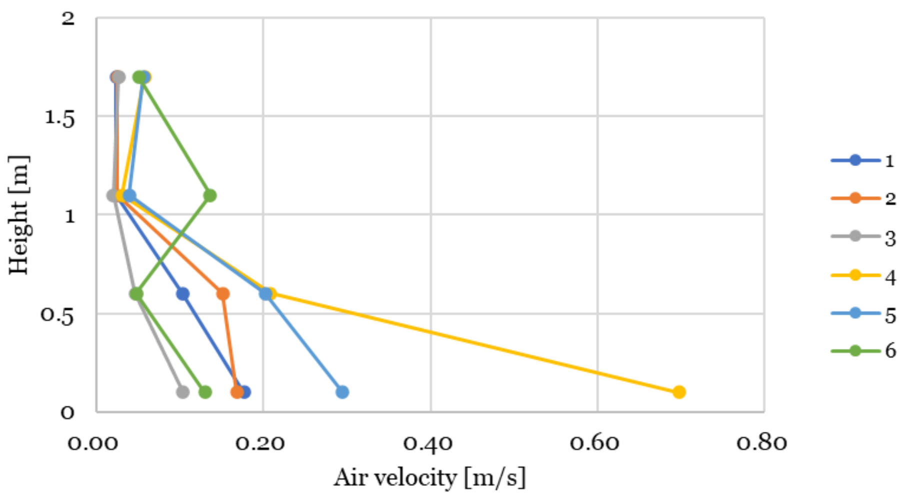

Figure 45. Vertical air velocity distribution for window case with HHG (67 W/m^2).

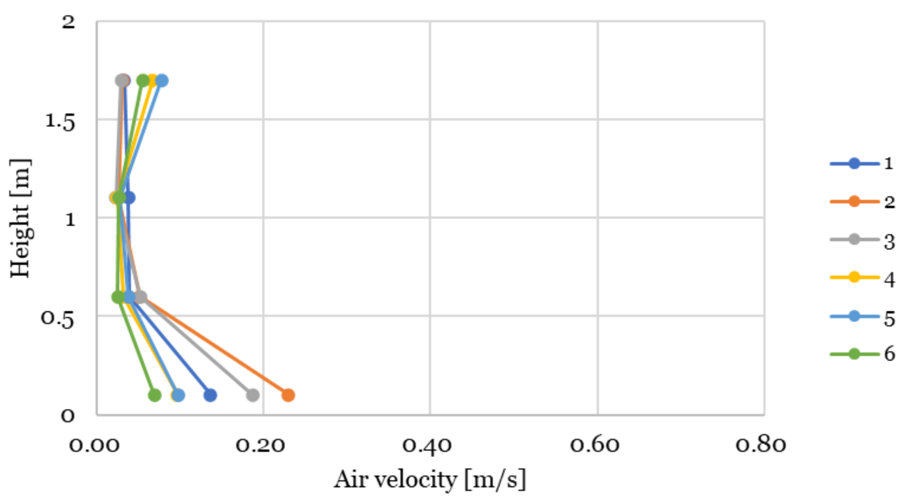

Figure 46. Vertical air velocity distribution for corridor case with LHG (36 W/m^2).

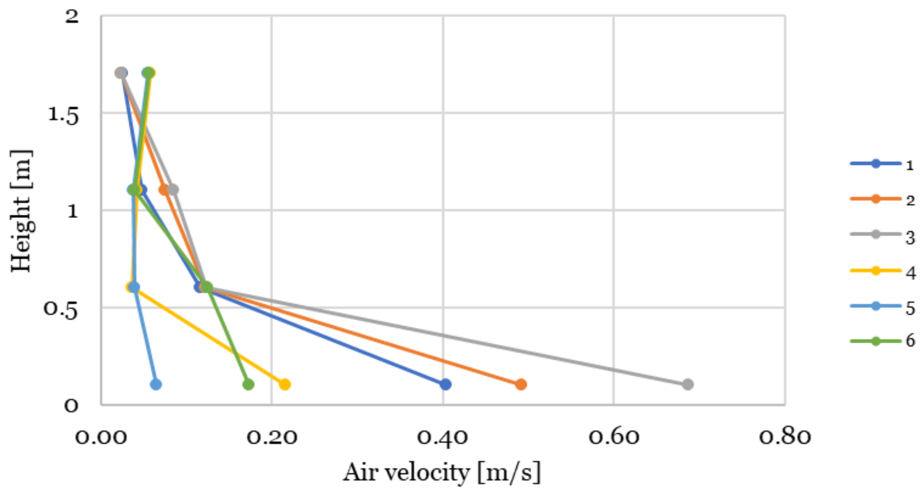

Figure 47. Vertical air velocity distribution for corridor case with HHG (67 W/m^2).

The three different classes of thermal comfort apply also to the air velocity [46]. The standard of the highest category is met only if the air velocity is lower than 0.14 m/s in summer, whereas the upper limit for the S3 level of comfort is 0.2 m/s. The applied category, the S2, only allows air velocity values up to 0.17 m/s. Based on the results, in some cases, the limit is not respected. The higher velocities are as expected under the configurations with higher heat gain. The increased cooling load requires a higher air flow rate that results in generally higher velocities in all the cases.

Table 17. Air velocity values for every height, occupant, and heat gain in the corridor lift case.

Velocity [m/s], Corridor lift

Heat gain [W/m^2]	Height [43]	Position					
		1	2	3	4	5	6
36	0.1	0.104	0.148	0.243	0.097	0.042	0.088
	0.6	0.193	0.158	0.076	0.076	0.121	0.057
	1.1	0.117	0.174	0.234	0.050	0.066	0.058

	1.7	0.284	0.422	0.099	0.066	0.078	0.061
67	0.1	0.179	0.315	0.191	0.126	0.094	0.227
	0.6	0.284	0.329	0.089	0.135	0.178	0.123
	1.1	0.342	0.176	0.428	0.102	0.074	0.111
	1.7	0.523	0.149	0.208	0.095	0.142	0.105

In Table 17, the values that are over the S2 limit in the corridor lift case are underlined in red. The lifted configuration is the most critical one. The limit of the air velocity is often not respected in positions 1, 2 and 3. These occupants are the ones close to the corridor wall, where the perforated duct is placed. The higher values are registered at the perforated duct height, of 1.7 m. This indicated that the comfort with that configuration cannot be guaranteed next to the air injection. This problem also occurs with the lower airflow rate, reaching an air velocity of 0.422 m/s in position 2, whereas in the HHG configuration, it reaches 0.523 m/s in position 1. The values calculated also do not respect the category of comfort S3. The air velocity values are instead adequate for the S2 category on the other side of the room, next to the window. In those positions, 4, 5 and 6, the air velocity is higher at the floor level than at 1.7 m. This indicates this supply configuration can be suitable only if not placed close to the breathing zones and allows the air distribution to decrease the airflow speed.

Table 18. Air velocity values for every height, occupant, and heat gain in the window case.

Velocity [m/s], Window							
Heat gain [W/m^2]	Height [43]	Position					
		1	2	3	4	5	6
36	0.1	0.106	0.080	0.057	0.096	0.170	0.125
	0.6	0.032	0.027	0.026	0.051	0.029	0.028
	1.1	0.044	0.025	0.028	0.033	0.042	0.094
	1.7	0.028	0.027	0.026	0.063	0.063	0.061
67	0.1	0.176	0.168	0.104	0.697	0.295	0.130
	0.6	0.103	0.151	0.047	0.209	0.202	0.048
	1.1	0.023	0.024	0.020	0.031	0.039	0.137
	1.7	0.024	0.025	0.027	0.058	0.057	0.051

Table 19. Air velocity values for every height, occupant, and heat gain in the corridor case.

Velocity [m/s], Corridor							
Heat gain [W/m^2]	Height [43]	Position					
		1	2	3	4	5	6
36	0.1	0.136	0.230	0.187	0.097	0.097	0.069
	0.6	0.038	0.051	0.052	0.032	0.038	0.025
	1.1	0.038	0.026	0.023	0.024	0.027	0.026
	1.7	0.033	0.031	0.029	0.067	0.078	0.055
67	0.1	0.404	0.492	0.687	0.215	0.066	0.174
	0.6	0.116	0.122	0.123	0.037	0.039	0.125
	1.1	0.049	0.075	0.086	0.042	0.039	0.040
	1.7	0.025	0.025	0.024	0.059	0.055	0.057

Instead, while supplying from the floor, the S2 level of comfort is respected with an LHG. The values over the limit of S2 are underlined in red in Table 18 for the window case and in Table

19 for the corridor case. As the perforated duct is placed at the floor level, the ankles' height (o.1 m) is the most critical one. Supplying from the window side allows us to stay within limits also in the level of this measurement. While supplying instead from the corridor side, the air velocities close to the perforated duct register values slightly over the limit at 0.1 m (0.230 m/s in position 2 and 0.187 m/s in position 3). The fact that the supply from the window side respects the limit with an LHG compared to the corridor case, suggests that the window close to the supply air helps the airflow to go up. This way the air moves from the ankle level of the occupied zone.

In the HHG case, instead, the increased airflow rate creates a too-high air velocity at the floor level. With both air terminal device placements, the higher values are registered close to the perforated duct. In the window case, the maximum is reached in position 4 with a velocity of 0.697 m/s, while in the corridor case, it reaches 0.687 m/s in position 3. The two configurations have similar results, and it indicates that the supply from the floor is adequate with a cooling load limit. The problem could be solved by not providing all the cooling load required from the supply air, but adding additional cooling systems when the demand is higher. As suggested in other similar studies [28], ceiling cooling panels could be a suitable solution.

5.4 *Draught Rate*

In the following figures, the values of the draught rate measured are reported for every occupant, configuration, and height in the room.

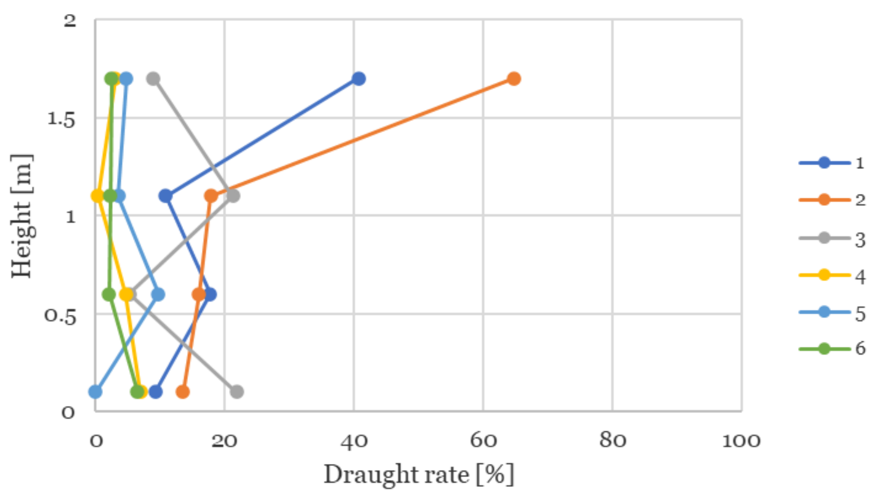

Figure 48. Draught rate at every height and position in the corridor lift case with LHG (36 W/m^2).

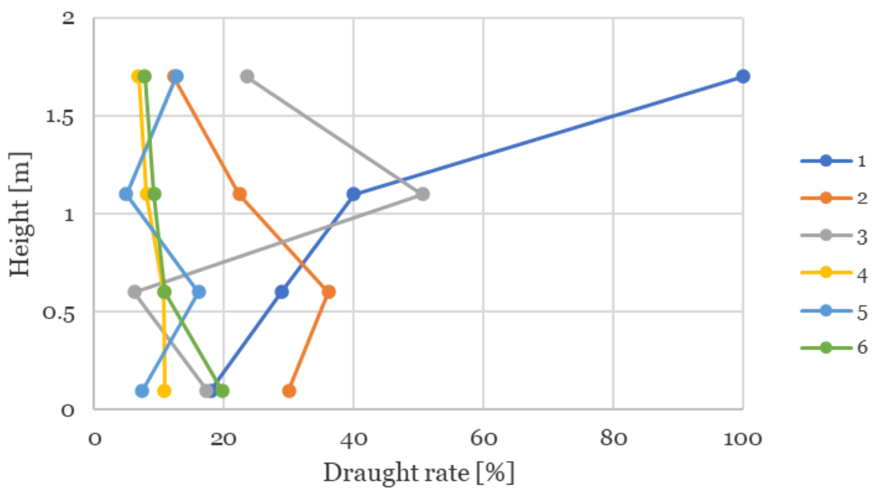

Figure 49. Draught rate at every height and position in the corridor lift case with HHG (67 W/m²).

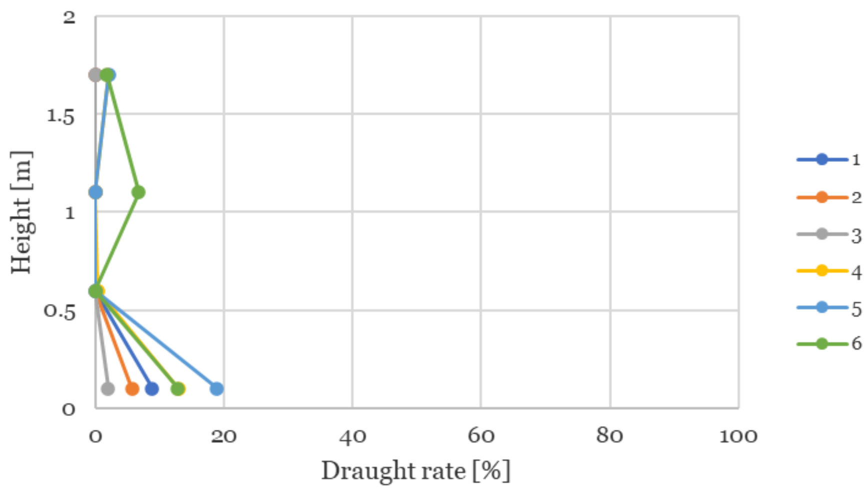

Figure 50. Draught rate at every height and position in the window case with LHG (36 W/m²).

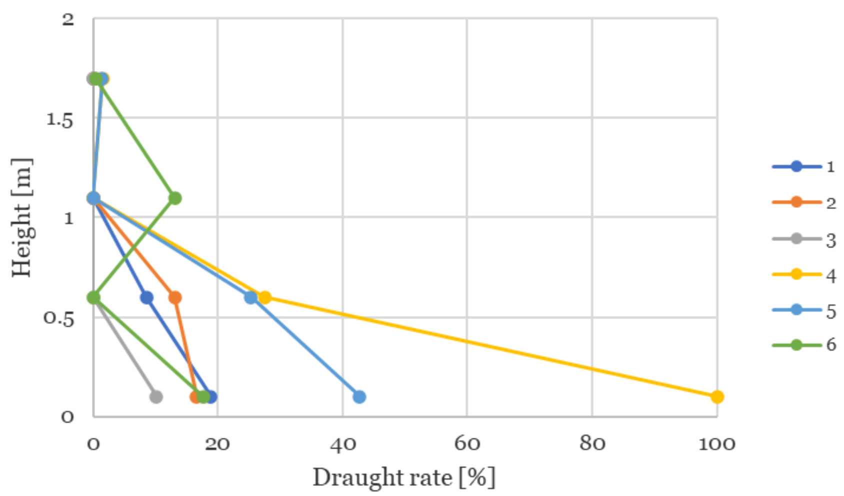

Figure 51. Draught rate at every height and position in the window case with HHG (67 W/m²).

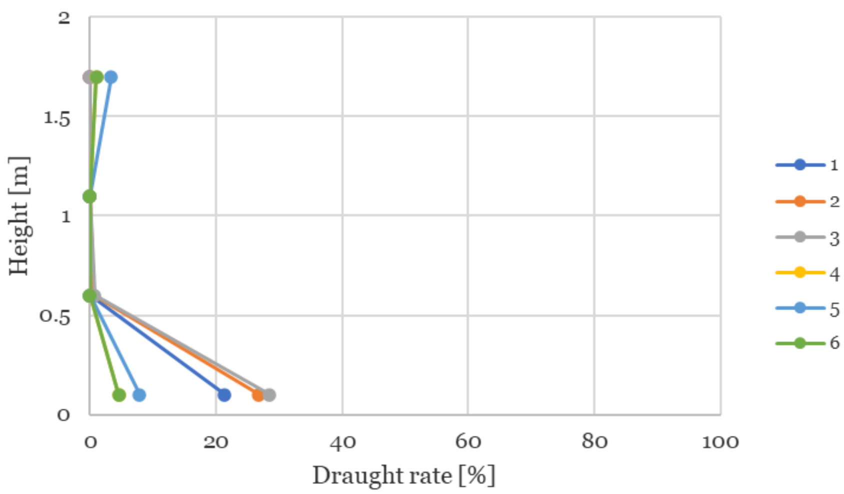

Figure 52. Draught rate at every height and position in the corridor case with LHG (36 W/m^2).

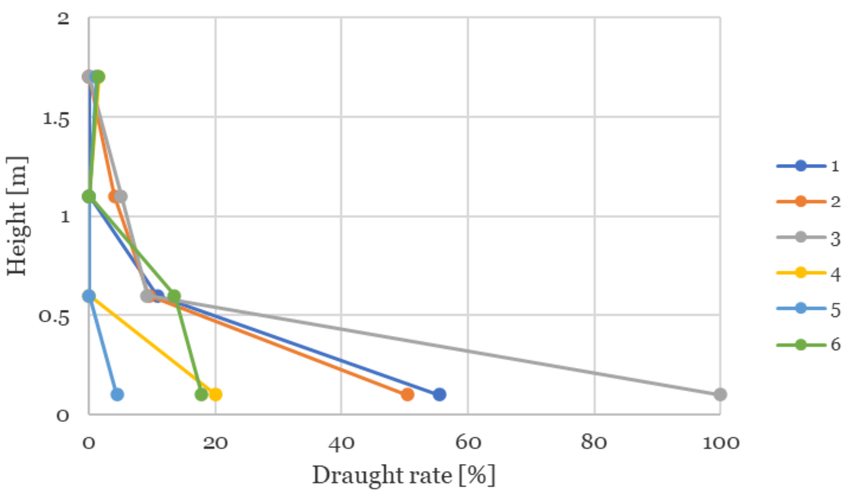

Figure 53. Draught rate at every height and position in the corridor case with HHG (67 W/m^2).

The results show that in some positions, with certain configurations, the draught rate reaches remarkably high values. To satisfy indoor comfort standards, the draft rate should be under 15% for the S2 category [17]. The draught risk is affected by temperature and air velocity, so the results are coherent with the ones previously discussed. The air velocity does not meet the requirements for category S2 as well as the vertical temperature difference. It is shown that the position more sensitive to changes is the one close to where the air supply system is placed, not meeting the requirements of those occupants.

For the lifted system the higher values of air velocities are reached at 1.7 m of height. The high air velocity values lead to a too-high draught risk, which does not meet the category S2 requirements for comfort. The increase of the heat gain has a worsening effect, as the airflow rate increases, and causes more difficulties in keeping a comfortable environment. However, with the lifted configuration the draught risk exceeds the limit already with an LHG, as shown in Figure 48. In positions 1 and 2, close to the perforated duct, the draught risk at 1.7 m is 40.7% and 64.8% with an LHG. Close to the corridor side, the draught risk is still higher at lower heights too. With an HHG, reported in Figure 49, the draught rates increase in particular in the middle of the breathing zones. At the height of 0.6 m and 1.1 m, the values reach 28.8% and 40.0% in position 1, and 36.1% and 22.3% in position 2. On the other hand, the draught rates registered on the window side positions, far from the supply inlets, are always safely under the limit. This indicates the lifted perforated duct is a suitable configuration only while placed far from the workstations.

Coherently to what was stated before, in the cases of supply air from the floor level, the air velocity is always higher closer to the ground. From both positions, supplying from the window or the corridor side, a higher heat gain increases the air velocity. The air velocity level reached at the ankles' height is causing local discomfort to the occupants. These aspects reflect on the draught risk in the occupied zones, reported in Figure 50 and Figure 51 for the window case, and in Figure 52 and Figure 53 for the corridor one. Supplying from the window side, the limit of the S2 category is respected for the LHG case. Only one value at 0.1 m exceeds the maximum, in position 5, reaching 18.9%. Instead, placing the perforated duct on the corridor side does not allow it to meet the comfort criteria also with the LHG. In particular, close to the supply, the draught rate reaches 0.1 m in positions 1, 2 and 3 with values of 21.4%, 26.8%, and 28.5% respectively. Instead, far from the supply the draught rate is around 7%.

The configurations with the perforated duct at the floor level present instead too high values of draught rate with an HHG. In particular, at the height of 0.1 m, the draught rate reaches the maximum. The highest are registered close to the supply. From the window side, in position 4 the draught rate is 100%, while in positions 5 and 6 respectively 42.7% and 17.6%. In those positions, the values are over the limit also in the upper part of the breathing zone. From the corridor side, instead, only the ankle levels represent a problem in meeting the comfort criteria. The values of the draught rate at 0.1 m are particularly high, with values of 55.4%, 50.4%, and 100% respectively in positions 1, 2, and 3. However, the window case is the only supply configuration respecting the draught risk limit. This, at least for the LHG case, shows that it could be an appropriate ventilation configuration without an excessive cooling load.

The results demonstrate that meeting the comfort criteria and thermal comfort is still a challenge in the built environment. However, all the different configurations had positive outcomes with safe infection risk calculated, even for a longer permanence of 8 hours. This shows that applying local exhaust in a meeting room is a promising improvement in contrast to indoor airborne transmission. The need to investigate more how to meet, in addition, the thermal comfort requirement as well represents a future field of study. From the result, it is underlined that thermal comfort can often be achieved only far from the supply air inlets, in particular, while supplying from the floor level. It would be then advisable in the not fully occupied room to place the seatings far from the supply unit. This is at least while high cooling loads are required, implying a high supply airflow rate.

6 Discussion

The results obtained underline the importance of advanced air distribution. Changing the perforated duct location impacts the infection risk and the occupants' thermal comfort. Moreover, the indoor air pattern is particularly sensitive to the heat gain level as well. It was underlined how the techniques to improve the IAQ usually have negative outcomes on thermal comfort. The experiments were carried out in a small meeting room, registering a high level of draught risk next to the supply air inlets, around 1 m away from the closest occupant. While supplying from the floor level, the vertical temperature gradient was over the limit. However, all the configurations showed an appropriate level of infection risk for the length of the experiment of 3 hours. Its value stayed below the limit even with an assumed time of permanence in the room of 8 hours.

The study points out that the use of local exhaust systems could be extended to common spaces like offices and meeting rooms too. The infection risk limits were safely respected in every configuration studied for every occupant of the volume. These findings could suggest, among the other studies in this field, to further investigate the local ventilation approach. Being able to guarantee a safe indoor environment could help prevent indoor airborne transmission. Moreover, assuring clean air in each breathing zone improves the occupants' quality of work, removing not only viruses but also irritating or annoying pollutants. The results show that it is possible to guarantee the required IAQ using a personalised exhausting. This is important in the scenario of decreasing the building's energy consumption. The IAQ has to be guaranteed, but this is often in contrast with lowering the energy consumption and costs. The study conducted for this thesis has the aim to contribute to moving towards a shift in the ventilation approach. The local approach, compared to the traditional mixing systems, could allow for a severe decrease in energy consumption and has been proven to be effective.

The analysis presented has several limitations that do not make the obtained results universally applicable. It could be that it is not possible to apply the results in every kind of meeting room, and that leaves the necessity for further studies in this field. First of all, the ventilation system presented is analysed only in the cooling season, while studies show different effects between heating and cooling conditions [35]. Moreover, the performance of the different diffusers has an impact [36], while here with the goal of showing the effect of different supply inlet placements, only the perforated duct was used. In addition, the position of the contaminant source has an effect [36], [28], while in this study it was kept fixed. The conduction of experiments with multiple pollutant source locations could help to go further from the specific case [30], [34]. The different heat gain distributions as well were not investigated, keeping those fixed. Also, the standard meeting room conditions were considered, assuming a steady state with average breathing conditions. Possible people's movements were not included, even though they could interfere with the air distribution and with the pollutant's path to the exhausts [47]. In addition, the infection risk was calculated considering the REHVA COVID-19 guidance, with the aim to only investigate the capacity of removal from the ventilation system. The condition of the environment for other specific pathogens was not considered, not considering the air temperature and the humidity as the important factors for the pathogens to survive in the surrounding environment [3]. In more specific needs, just as when during the pandemic the focus was on COVID-19, humidity and other factors could be controlled. In fact, different pathogens survive better in different conditions, making that an important parameter for the occupants' health as well [13].

From the results obtained, several ideas for further improvement of the studied set-up are suggested. One of the problems underlined was the worsening effect on thermal comfort with the increasing heat gain and airflow rate. This factor has an opposite effect, instead, on the CRE and then on the infection risk. On the other hand, the infection risk limits are largely respected even with lower heat gain and lower airflow rate: this suggests that the investigated system could be suitable for a not-high demanding cooling load. In case of additional cooling power requirements, instead of increasing the air supply flow rate provided, the use of additional cooling technologies could be implemented. For example, the use of a ceiling cooling panel could be considered [36]. This change could positively affect the vertical temperature gradient, decreasing the temperature difference from the head to the ankle level for the occupants. Moreover, reducing the air required from the perforated duct allows the injection at lower velocities decreasing the draught risk. Other than that, the extra demand for cooling could be provided by PV systems, allowing the single person to control their own micro-environment which is proven to increase the perceived thermal comfort [4], [28]. In addition, to overcome the main disadvantages of supplying from the floor level, the extracted warm air from the occupied zone could be directed to the feet level, reducing the temperature difference [48].

The thesis would suggest further studies in the application of local exhaust in common public indoor spaces such as meeting rooms, offices, and schools. The air distribution has to be perfected as the thermal comfort is still not optimal. Further research could then find a between configuration and try to analyse the experiment room while supplying from different parts of the room. This could help in reducing the draught risk, not having all the supply airflow required supplied close to one occupant. Secondly, the idea of creating micro-environments in the indoor space is supported by an energy-saving mindset, for economic reasons and as a contrast to global change. Here in this thesis, only the effects of the ventilation are investigated, without further analysis of the energy performance of the ventilation which represents an interesting follow-up for this work. Moreover, for the reasons stated above, conducting experiments with different air supply technologies could be interesting as well, analysing different coupling with the local exhausting. For this aim, it could be useful to integrate the Computational Fluid Dynamics model, CFD, to validate the experimental results and to further investigate more configurations. With the aid of simulating tools, different layouts of the room as well as different parameters could be analysed.

7 Conclusions

In this thesis, an analysis of the performance of local exhaust coupled with three different perforated duct configurations was analysed. The study investigated the ventilation efficiency in removing the contaminants as well as determining the infection risk for the different occupants. Moreover, the study focused on thermal comfort, to assure that an effective pollutant capture does not imply a reduction in occupants' local thermal comfort. The experiments were conducted in a test chamber at Aalto University, assumed to be a meeting room. According to the final use of the room, a time of permanence of 3 hours was considered for the length of the experiments. The occupants of the meeting room were supposed to be 6, with one of them infected represented by a breathing manikin. The tracer gas SF_6 was used to simulate a pathogen, exhaled from the manikin.

The results show thermal comfort conditions and effective pollutant removal are often in contrast. Supplying from the floor-level, results in being the best option presenting the lower infection risks, far from the limit suggested. On the other hand, supplying from a height of 1.7 m allows better temperature distribution. Still, the lifted supply respects the infection risk limits, reaching a maximum of 4.08% after 8 hours and 1.5% after 3 hours. Every configuration allows for maintenance of the set-up temperature, and also a good HRE. However, the local comfort is not always guaranteed. While supplying from the floor level, the vertical temperature gradient in the breathing zone is too high, up to 5.97°C in the corridor case and 5.29°C in the window case. With the perforated duct lifted, instead, the limit was always respected without a strong thermal stratification. Every configuration instead, registered high air velocity, in particular close to the perforated duct. Supplying from the floor, the air velocity is very high at the ankle level reaching 0.697 m/s while close to the window and 0.687 m/s while close to the corridor. Also, in the corridor lift case, the air velocity is too high, but at 1.7 m height, where it reaches 0.523 m/s. The high values of air velocities have a strong effect on the draught rate, which even reaches 100%. The performances with an HHG were worse, while often the thermal comfort could be guaranteed at an LHG. This suggests a need for improved configuration while the cooling load required is higher. Moreover, in a not fully occupied room, it is better that the occupants sit far from the supply unit.

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Symbols

ε^c	Contaminant Removal Efficiency
c_i	Local concentration of pollutant in the position "i" [ppm]
c_{exh}	Concentration of pollutant in the exhausted air [ppm]
P_D	Infection risk [%]
$N_{quantum}$	Inhaled quanta by the exposed person [quanta/s]
$p_{exposed}$	The breathing rate of the exposed person [m^3/s]
$C_{quantum}$	Airborne quantum concentration at the exposed person [quanta/m^3]
q	Quantum generation rate [quanta/s]
$p_{infector}$	Breathing rate of the infected person [m^3/s]
D	Dilution ratio
$c_{infector}$	Pollutant concentration at the infector [ppm]
$c_{exposed}$	Pollutant concentration at the exposed [ppm]
T_{op}	Operative temperature [°C]
h_r	Radiative coefficient [W/(m^2,K)]
h_c	Convective coefficient [W/(m^2,K)]
T_a	Indoor air temperature [°C]
T_{ex}	Exhausted air temperature [°C]
T_{su}	Supply air temperature [°C]
$T_{ave(0.1\text{-}1.1)}$	Average temperature between the heights of 0.1 m and 1.1 m [°C]
$T_{a,l}$	Mean local air temperature [°C]
v	Mean local airspeed [m/s]
T_u	Local air turbulence intensity [%]
u_{sd}	Fluctuation of velocity value
$\hat{u}$	Mean air velocity [m/s]

Figures index

Tables Index

Ringraziamenti

Questa laurea è per me un grande traguardo e sono estremamente felice ed orgogliosa. Sono contenta di come ho affrontato le difficoltà incontrate e di essermi sempre messa in gioco. Ringrazio di non aver mollato quando all'inizio ingegneria mi sembrava uno scoglio troppo grande e avevo la sensazione di non capire nulla. Posso sicuramente dirmi fiera di me e di essere pronta per le nuove sfide che mi attendono.

Vorrei cominciare ringraziando i professori Risto Kosonen e Marco Perino, e la dottoressa Weixin Zaho per il supporto nella realizzazione della tesi.

Sono particolarmente grata di aver avuto la possibilità di andare in Erasmus, vivendo un anno in Finlandia. È stata un'esperienza intensa, con tanti momenti importanti che sono sicura hanno contribuito a formare la persona che sono e che sarò. Ho trovato un paese sorprendentemente accogliente e calmo, pieno di persone con tanta voglia di fare e di stare assieme condividendo. Sono stata nei paesaggi più belli che abbia mai visto, quando ero più fortunata con l'aurora boreale. Sono contenta per ogni persona che ho incontrato in quel lungo, ma che allo stesso tempo è volato via veloce, anno, sicura che di ognuna di loro mi porterò dietro qualcosa.

Oltre al mio impegno personale però, ho avuto la fortuna di essere sempre ampiamente supportata (non solo economicamente!). Ringrazio tanto la mia famiglia, per supportarmi sempre e credere tanto in me. Nonostante sono sicura non sia sempre semplice per loro avermi lontana, mi hanno sempre affiancata nel mio desiderio di esplorare il mondo e ascoltato curiosi i miei racconti.

Ringrazio anche i miei amici a Bologna, in particolare Chievo e TC, poiché nonostante sia poco a casa negli ultimi anni, sono sempre pronti a rivedermi a braccia aperte ogni volta che torno, facendomi sentire come se non fossi mai andata via.

Infine, ringrazio Thilo per essere il compagno di viaggio che sognavo, nonché la persona più buona che conosco. Sono sicura, in tua compagnia, di essere la migliore versione di me. Non vedo l'ora di cominciare questa nuova avventura insieme! Ich liebe dich <3.

Kiitos!

Bologna, 16/09/2024

Martina