Experimental investigation of ventilation performance of corner placed stratum ventilation in an office environment

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Abstract

Energy use in buildings account for about one third of the total global energy supply and contributes as much as 30% of the anthropogenic greenhouse gas emissions. It is estimated that energy use in buildings will increase to 67% by 2030. The need for better thermal comfort and air quality in indoor environments is the leading cause for high energy use in buildings. Heating, ventilation and air conditioning systems take up about 50% of the total energy use in buildings which is about 10-20% of the national energy use in most developed countries. The development and adoption of sustainable ventilation systems is a viable solution to mitigate climate change and curtail carbon emissions.

The experimental study was conducted in a room resembling a modern office in a laboratory environment. The study involved investigating the ability of the system to provide cooling and heating. Concentration decay tracer gas technique using Sulphur hexafluoride (SF$_6$) gas was used to determine the local air change index and air change efficiency in the room. Low-velocity omni-directional thermistor anemometer type CTA88 were used to measure the air velocity and temperature in the room. Smoke was used to visualise the flow patterns created in the room. The climate chamber was used to mimic climatic conditions in winter. Fifteen cases were investigated with five air flow rates set points (30, 40, 50, 60 and 70 l/s) at three supply air temperatures, i.e., 17.6 °C, 21.0 °C and 25.3 °C.

The results of the local air change index and air change efficiency for the nominal supply temperature of 17.6 °C showed that the system had strong characteristics of a mixing ventilation system. At the supply air temperature of 21.0 °C, the performance of the system deteriorated slightly to below that of a mixing ventilation system and could not satisfactorily provide heating at supply temperature of 25.3 °C. Better performance of the system at all supply air temperature setpoints was observed at lower airflow rates. At all supply air temperature setpoints, relatively higher degree of temperature stratification was observed at lower supply. The draught rate levels decreased with increase in supply air temperature and height. The location of the air inlet terminals in relation to the workstations had significant effect on the performance of the system. The stratum ventilation system did not work efficiently because the air streams were heavily mixed before reaching the occupants.

Keywords: Stratum ventilation, local air change index, air change efficiency, ventilation effectiveness, draught rate, percentage dissatisfied, heat removal effectiveness
Nomenclature

C1_SV  Case 1 of stratum ventilation configuration at 17.6 °C and 30 l/s
C2_SV  Case 2 of stratum ventilation configuration at 17.6 °C and 40 l/s
C3_SV  Case 3 of stratum ventilation configuration at 17.6 °C and 50 l/s
C4_SV  Case 4 of stratum ventilation configuration at 17.6 °C and 60 l/s
C5_SV  Case 5 of stratum ventilation configuration at 17.6 °C and 70 l/s
C6_SV  Case 6 of stratum ventilation configuration at 21 °C and 30 l/s
C7_SV  Case 7 of stratum ventilation configuration at 21 °C and 40 l/s
C8_SV  Case 8 of stratum ventilation configuration at 21 °C and 50 l/s
C9_SV  Case 9 of stratum ventilation configuration at 21 °C and 60 l/s
C10_SV Case 10 of stratum ventilation configuration at 21 °C and 70 l/s
C11_SV Case 11 of stratum ventilation configuration at 25.3 °C and 30 l/s
C12_SV Case 12 of stratum ventilation configuration at 25.3 °C and 40 l/s
C13_SV Case 13 of stratum ventilation configuration at 25.3 °C and 50 l/s
C14_SV Case 14 of stratum ventilation configuration at 25.3 °C and 60 l/s
C15_SV Case 15 of stratum ventilation configuration at 25.3 °C and 70 l/s
DAT    Dimensionless air temperature
SV     Stratum ventilation
DV     Displacement ventilation
MV     Mixing ventilation
IJV    Impinging jet ventilation
IAQ    Indoor air quality
ε_p^a  Local air change index
ε_a    Air change efficiency
ε_p^c  Local air quality index
ε_c    Contaminant removal effectiveness
HRE    Heat removal effectiveness for the cooling cases
ε_T    Temperature effectiveness for the heating cases
Ar_i   Archimedes number
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$\tau_{p\text{mean}}$</td>
<td>Local mean age of air</td>
</tr>
<tr>
<td>$\tau_{\text{mean}}$</td>
<td>Mean age of air</td>
</tr>
<tr>
<td>DR</td>
<td>Draught rate [%]</td>
</tr>
<tr>
<td>PD</td>
<td>Percentage Dissatisfied [%]</td>
</tr>
<tr>
<td>PPD</td>
<td>Predicted Percentage Dissatisfied [%]</td>
</tr>
<tr>
<td>PMV</td>
<td>Predicted Mean Vote</td>
</tr>
<tr>
<td>$I_p$</td>
<td>Turbulence intensity</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravitational acceleration [m/s²]</td>
</tr>
<tr>
<td>$T_a$</td>
<td>Local air temperature at a point [°C]</td>
</tr>
<tr>
<td>$T_i$</td>
<td>Inlet air temperature [°C]</td>
</tr>
<tr>
<td>$T_o$</td>
<td>Outlet air temperature [°C]</td>
</tr>
<tr>
<td>$T_{0.1,0.6,1.1}$</td>
<td>Arithmetic mean air temperature at heights 0.1, 0.6 and 1.1 m [°C]</td>
</tr>
<tr>
<td>$T_r$</td>
<td>Mean air temperature in centre of room at 1.7 m from floor [K]</td>
</tr>
<tr>
<td>$A_e$</td>
<td>Inlet supply opening area [m²]</td>
</tr>
<tr>
<td>$u_{in}$</td>
<td>Nominal inlet air velocity [m/s]</td>
</tr>
<tr>
<td>$u_{rms}$</td>
<td>Root mean square of the turbulent velocity fluctuations [m/s]</td>
</tr>
<tr>
<td>$u_{\text{mean}}$</td>
<td>Mean velocity [m/s]</td>
</tr>
<tr>
<td>C</td>
<td>Tracer gas concentration [ppm]</td>
</tr>
<tr>
<td>$C_s$</td>
<td>Concentration of the tracer gas in the supply air [ppm]</td>
</tr>
<tr>
<td>$C_0$</td>
<td>Initial concentration of the tracer gas [ppm]</td>
</tr>
<tr>
<td>n</td>
<td>Air change rate [h⁻¹]</td>
</tr>
<tr>
<td>t</td>
<td>Time [h]</td>
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1 Introduction

1.1 Background and research focus

The generation and use of energy is the largest contributor of anthropogenic CO₂ emissions [1]. About 80% of the total global energy supply comes from fossil fuels which account for over 65% of the total global greenhouse gas emissions [2], [3]. It is predicted that economic development, globalisation and population growth will be among factors that will increase energy use worldwide in future [4]. Industry, transportation and building sectors number among significant end-users of the global energy supply. Recently, the building sector has surpassed the transportation sector in terms of energy use due to increase in population, enhancement of the building services and the requirements of higher comfort levels [5]. Other contributing factors to higher building energy use include size and location of the building, weather conditions, architectural design, building energy systems and economic standard of occupants. It is projected that by 2030, energy use in buildings will increase to 67% [5].

The quest to achieve and maintain acceptable indoor environment has led to a significant amount of energy use in both residential and commercial buildings. Heating, cooling and other building energy services account for about one third of the total worldwide energy use and contribute as much as 30% to the anthropogenic CO₂ emissions. In colder climates like in most European countries, higher energy use occurs with buildings taking up to 40% of the total energy supply and contributing as much as 36% of the total CO₂ emission [6]–[8]. Research has shown that about half of the total building energy use pertains the creation of comfortable conditions and good indoor air quality (IAQ). In developed countries, the heating, ventilation and air conditioning (HVAC) systems take up about 50% of the total building energy use which is estimated to be 10-20% of the national energy use [4], [5], [9]. For climate protection and the reduction of global CO₂ emissions the economical use of energy resources is of outstanding significance.

Ventilation systems play a key role in creating an acceptable microclimate in the indoor environment. In modern day society, people especially infants and the elderly spend more than 90% of the time in artificial indoor environment [10], [11]. Acceptable thermal environment and air quality in indoor spaces such as dwellings, workplaces or in automobiles have been
linked to higher productivity and general well-being of the occupants. Bako-Biro et al. found that adequate ventilation rates significantly improved thermal comfort and indoor air quality which in turn enhanced the performance of pupils in schools [12]. On the contrary, poor indoor environment has been associated with problems such as the ‘sick building syndrome’ [11]. Indoor air pollution which is greatly influenced by particulate matter has been shown to increase with insufficient ventilation rates and air flow patterns that cause stagnation zones within the occupied zone of the indoor environment. The ability of particles to significantly affect the indoor air quality is dependent on the airborne particle concentration, size distribution and chemical or biological composition [13]. Lung cancer, asthma, different cardiovascular and cardiopulmonary diseases are attributed to people’s exposure to particles in indoor environments [14].

The building sector presents potential for energy saving. There are many energy saving measures that can be explored within the sector such as conducting energy efficiency awareness for occupants, improving the building energy management and incorporating energy efficient technology [15]. Improving energy efficiency in buildings can lead to reduction in primary energy demand and greenhouse gas emissions [16]. It has been proved that improving the energy performance of buildings results in reduced energy demand for building operations while upholding the health and comfort of the occupants [15].

One area of potential energy saving in the building sector involves the operation of ventilation and air conditioning systems. Achieving thermal comfort and good health of the building occupants with minimised use of energy is the essence of HVAC systems [17]. This research focuses on ventilation efficiency and energy use in buildings. The adoption of sustainable ventilation methods is a viable solution to mitigate climate change and curtail carbon emissions. It has been demonstrated that the use of advanced ventilation methods like stratum ventilation (SV) and displacement ventilation (DV) systems in specific configurations can reduce the carbon emissions up to 31.7% and 23.3%, respectively [18]. Increasing the ventilation effectiveness significantly reduces occupants' exposure to particles in the indoor environment [19]. Enhancement of ventilation i.e., increasing the air change rate is an efficient measure to additionally reduce the pollutant load in indoor spaces [20].
1.2 Overall research aim and specific research objectives

The overall aim of the study is to evaluate the influence of supply air temperature and supply airflow rates on the ventilation effectiveness of a corner-placed stratum ventilation system in an office. The goal of the study is to evaluate how reduction in energy use while maintaining acceptable indoor environment can be achieved using new technology. The specific objectives are:

- To conduct experimental study involving tracer gas technique to determine different ventilation effectiveness indices: local air change index and air change efficiency.
- To carry out measurements of the air velocity and temperature in the test room in order to determine the thermal comfort conditions.
- To conduct flow visualization to ascertain the airflow pattern in the test room.
- To evaluate critically the effect of supply air flow rates and supply air temperature on the airflow patterns created in an office.
- Finally, to analyse critically the effect of the airflow rates and supply air temperature on the overall ventilation effectiveness of the SV system.

1.3 Significance of the study

This research forms part of the ongoing search for better ventilation systems across the world. The study is aimed at ascertaining the performance of the SV system at supply temperature lower than 21 °C; the recommended minimum temperature for better performance and at room temperatures lower than 25 °C. In part this research answers the call for further investigation into the effects of the different variable parameters such as types of supply air terminal, supply, exhaust locations and number to determine the optimal configuration for the SV system. The study also sought to verify the suitability of SV system for heating applications.
2 Literature review

The objectives of the study are to determine the ventilation indices for the proposed combination of ventilation systems to evaluate its performance in terms of ventilation efficiency. Mainly, the effect of the supply air flow rates and temperature on the ventilation effectiveness of the SV system was evaluated. A properly designed and installed ventilation system ensures good distribution and control of air movements to achieve acceptable thermal comfort and good air quality in the occupied zone [21].

The field of ventilation has been receiving increasing attention from many researchers due to its influence on the general well-being and productivity of people. Many scholars have delved into the study of different ventilation systems. In Asia, SV system has been receiving much attention recently.

2.1 Stratum ventilation system

The SV system was proposed by Lin as a response to the requirements of some governments in East Asia of operating indoor spaces at elevated temperatures in order to conserve energy [22]–[24]. The new recommended indoor air temperatures in the Republic of Korea (26-28 °C), Chinese mainland (26 °C), Hong Kong (25.5 °C), Taiwan (27 °C) and Japan (28 °C) for summers [25]. Since the conventional ventilation systems are incapable of efficiently providing thermal neutrality in warm conditions, the SV system was devised to serve that purpose. The ventilation system is aimed at coping with higher room temperature and air movement and has been found suitable for cooling small to medium rooms [26]. Lin et al. stated that with a properly designed supply air velocity and volume, location of diffusers and exhausts, the SV system has potential to maintain better thermal comfort with a smaller vertical temperature difference, lower energy use and better IAQ in the breathing zone [23]. In addition, the comparison of the mean air temperatures in the occupied zone confirmed that SV systems offered the highest cooling efficiency, followed by DV and then mixing ventilation (MV) systems [27].

The SV system draws the strengths of the personalised ventilation systems. Personalized or task ventilation systems have been ranked as the most energy efficient and provide the best air
quality in the breathing zone. However, such systems are inadequate because only limited ductwork can be installed in the occupied zone to avoid obstructions. Besides the limited ductwork, task ventilation systems cannot adequately cater for the mobile occupants within the occupied zone [28]. The SV system which supply fresh air directly into the occupied zone was proposed to overcome the shortcomings of the task ventilation system while retaining their benefits of better indoor air quality and energy performance [28]. For example, owing to its low nonlinearity and fast response, the SV system can be used to offer differentiated air velocity, temperature and PMV distributions to cater for individual occupant preferences in shared spaces [29], [30].

The underlying operation principle of the SV system is supply of fresher air directly into the occupants breathing zone (i.e., between 0.9 m and 1.4 m from the floor surface). To achieve this purpose, air supply inlets are placed at the side wall of the room at locations slightly above the head height of the sitting person. The recommended air inlet height is 1.3 m from the floor which corresponds to head level of sitting sedentary worker [30]. As a result of typical discharge height, the air speed increases along the height, but a reverse temperature gradient (cool head and warm ankle) is formed in the occupied zone. Consequently, lower CO$_2$ concentration exist in the occupied zone than in the upper part of the room. The cooling effect which is strongest at the head level is due to both lower temperature and air movements of the supplied air [25].

Many benefits are realised from the direct supply of air into the occupied zone such as shorter supply air path, younger mean age of air, higher ventilation effectiveness and better IAQ in the breathing zone. Other advantages include smaller capacity required, smaller system size, smaller space requirement, lower initial costs, lower energy use and smaller carbon footprint compared with the MV, DV, impinging jet ventilation system (IJV) and Task ventilation systems for a particular application [26], [31]. It is recommended that the supply air path should not be longer than 9 m to achieve better performance [26]. Ventilation inefficiencies resulting from short cut ventilation phenomenon are minimized in this ventilation type. Main characteristics of the SV system include: reverse temperature gradient in the occupied zone; higher air speed at the head–chest level for equal air supply volume; higher supply air temperature; higher room air temperature and higher evaporating temperature for the associated
refrigeration plant, thus higher coefficient of performance (COP) for the refrigeration machine(s) [22], [23].

The energy saving potential of the SV system lies in its use of low airflow rates because only the zones requiring cooling are serviced (head level). Additionally, energy is also saved by avoiding overcooling of the lower part of the room [32], [33]. The fan power is a function of airflow rate and the efficiency of the fan used in a ventilation system. Thus, lower airflow rate and higher efficiency of the fans lead to lower energy use [15]. The cooling effect is obtained from the influence of both the low supply air temperature and air movement in the occupied zone [25], [28]. When the annual energy use of the SV system was compared with the MV and DV systems, substantial amount of savings were realised at 44% and 25%, respectively [32]. Lin et al. attributed this energy saving due to the reduction in ventilation and transmission loads coupled with increased COP of chillers used in SV systems [32].

According to the proposed performance evaluation and design guidelines for the SV system, the recommended room temperatures are between 25.5 °C and 27 °C reliant on the activity level and clothing insulation value. For better performance, recommended supply air temperature of 21 °C can be utilized as the preliminary value. Depending on the level of thermal comfort, supply air temperatures of between 20 °C and 23 °C can also be used [26]. To minimise the risk of draught and cross contamination, the supply air velocity and the location of the air supply and exhaust terminal devices must be optimized to break the boundary layer around the occupant’s body. The location of the exhaust air terminal can be at elevation either below or above the supply air terminal [33]. Since this ventilation system is still at infancy stage its suitability for heating applications is still being researched and established studies have indicated that the system has not been found suitable for such use [26]. Typical configuration of SV system is shown in Fig.1, a study conducted by Fong et al. to evaluate thermal comfort conditions in a classroom using three different ventilation types.
A study by Fong et al. to evaluate the thermal conditions in a classroom using three ventilation methods showed that stratum ventilation could provide satisfactory thermal comfort level to room temperature up to 27 °C [33]. The study also illustrated that SV system used less energy due to less ventilation load. SV achieved energy saving of 12% and 9% compared with the MV and DV systems, respectively. Furthermore, energy use by three ventilation systems was examined for an office, classroom and retail shop in Hong Kong. The results revealed that the year-round energy use by the SV system was lower than that for MV and DV systems [33]. To ascertain the thermal and ventilation performance of the SV system, Tian et al. experimentally investigated the influence of air speed, temperature and CO₂ concentration in a stratum ventilated office. The results of the study indicated that the values of the predicted mean vote (PMV), predicted percentage of dissatisfied (PPD) and percentage dissatisfied due to draught (PD) conformed to the requirements of ISO 7730, and ASHRAE 55-2010 standards [25]. The supply air temperature of 21 °C was found to provide better thermal comfort than air supplied at 19 °C. The ventilation effectiveness was close to 1.5 and the ventilation system was expected to provide better IAQ in an efficient way [26], [34].

SV system is still a novel ventilation system. Further research is needed to investigate the effects of the different parameters such as the supply air temperature, air supply velocity and location of the air supply terminals to determine the optimal configuration of the ventilation system. The current study investigated the influence of supply airflow rate and supply air temperature on ventilation effectiveness of the SV system with inlet air terminals placed in the
corners of a medium-sized office. The rectangular air inlet terminals are placed in the corners to examine whether the system would perform as effective as the corner-placed IJV system such as the ability to be used in different architectural spaces and larger penetration distance into the ventilated space [35]. This would enable the system to be used in large spaces while using few air inlets. The study also sought to verify the suitability of the SV system for use at lower room temperatures and supply air temperatures than those used in previous studies.

2.2 Parameters affecting the ventilation process

Many parameters have been shown to affect the ventilation effectiveness and energy performance of any ventilation system. Factors such as shape of the air supply device, air discharge height, supply flow rate and supply air temperature. Others include heat loads and location of exhaust terminals. The influence of each parameter on the ventilation process is presented as espoused in previous studies regarding some of the ventilation systems.

2.2.1 Type of air inlet device

A good air distribution system conserves energy and is key in creating a healthy and comfortable indoor environment for occupants [36]. The type of air inlet has significant effect on the generated airflow pattern in the ventilated space. The resulting airflow pattern in turn determines the condition of the indoor environment [37]. Lee et al. conducted a study to evaluate the impact of inlet types used in MV system. They concluded that the air inlet type is an essential physical determinant to the distribution of the airborne contaminant concentrations. Different contaminant concentration patterns result from the airflow patterns that are generated by each inlet type [38]. Chen et al. studied the effect of several parameters on the performance of the IJV system and discovered that the shape of the supply device was fundamental in determining the flow pattern on the floor [36]. In a related study, Chen et al. found out that a square-shaped air supply device created lower overall draught discomfort than rectangular and semi-elliptic shapes in IJV systems [37].

It can be claimed that the airflow pattern created by the inlet device has direct connection to the ventilation effectiveness, especially in relation to the occupied zone. For instance, studies
in traditional displacement ventilation and under floor air distribution systems showed that the diffuser type has significant impact on the ventilation efficiency and energy performance of the systems [39]–[41]. In a computational fluid dynamics (CFD) study to compare ventilation effectiveness indices in isolation rooms, it was concluded that the type of diffuser has notable influence on the airflow patterns in the room and determine the risk of infection [42].

Lin et al. conducted a numerical study to validate the CFD model for use in SV system. The results of the study indicated that the location of inlet air supply and exhaust opening was instrumental in bringing air of younger mean age and displacing of contaminants from the occupied zone [28]. In the same study, it was asserted that the performance of the SV system in terms of thermal comfort, dispersion of CO₂ and indoor air quality was higher than that of the conventional MV and DV systems. In addition, the results of a study by Tian et al. that looked at the diffusion of CO₂, formaldehyde and toluene showed that the flow pattern formed by the SV system was capable of providing good IAQ in the breathing zone [43]. In another study, Tian et al. discovered that the particle concentrations in a stratum ventilated room, especially in the breathing zone were less than those under DV system [44].

Yao and Lin used the velocity and temperature distributions to investigate the performance of the fabric diffusers in SV system. The experimental results showed that fabric diffusers can be used as air terminals for SV system and performed better under higher airflow rates. Compared with double deflection grilles, fabric diffusers provided better air diffusion and a more comfortable thermal environment characterised by relatively uniform velocity and temperature distribution [45]. Moreover, Yao and Lin conducted an experimental and numerical study to explore the impact of air terminal on the performance of the SV system. They discovered that the air supply terminal had significant effect on the airflow pattern and recommended a circular diffuser for better performance of the SV system [45]. Additionally, an investigation into the air-borne infection performance of the SV system disclosed that the flow patterns created by different ventilation methods have great influence on the particle concentration. The risk of inhaling pathogenic particles was lower under SV system than that under DV system because of lower particle concentrations in the breathing zone in the former [46].
2.2.2 Discharge height and location of air inlets

The effect of the air discharge height to a greater extent has given rise to the different ventilation systems in existence today. For instance, the discharge height for MV system is either on the ceiling or near the ceiling, while for the DV and IJV systems it is near the floor; and the same lies in the range 0.8 m to 1.4 m for SV. Chen et al. indicated that the discharge height had a significant influence on the jet decay velocity and the airflow pattern in IJV system [36], [47]. A computational investigation on the factors influencing thermal comfort in an IJV system room showed that low discharge height and shape of air supply device had major impact on the flow pattern in the vicinity of the supply device because of the footprint of the impinging jet. Consequently, the created flow pattern affected the draught level in the occupied zone [37].

In addition to the discharge height, the location of the air supply inlet has profound impact on the overall thermal comfort in the ventilated space. The temperature distribution and pattern are predominantly determined by the location of the air supply terminals in the room [48]. Chung and Hsu showed that different distributions of thermal comfort factors in the same ventilated space can result due to the locations of the air inlet terminal [49]. In the same study, it was alleged that the ventilation efficiency might be dominantly influenced by the location of the diffuser than the air change rate [49]. Khan et al. numerically investigated the effects of the relative locations of inlets and outlets in MV systems. The results disclosed that the ceiling supply inlet can provide a uniform concentration distribution in mixing ventilated room [38]. However, McCarry’s study revealed otherwise; it proposed that the ceiling-mounted supply air inlet leads to poor circulation at the desk in partitioned areas [50]. Villafruela et al. stated that the position of the air inlets and outlets has significant influence on the quality of the ventilation [51]. This viewpoint was reinforced by the results of an experimental study to evaluate air distribution in mechanically ventilated residential rooms. It was discovered that the ventilation effectiveness depended on the location of the supply and extract air terminals and on the difference between the supply air and room air temperature [52].

The guiding principle in the location of the supply terminal is that it must be in a place where fresh air can be delivered to all parts of the occupied zone. For this cause, some researchers recommend that the air supply devices should be placed close to the centre of the room [48]. Lin advised that diffusers should be located away from the occupants’ locations to minimize the draught risks [48]. Baumann et al. recommended that occupants should at least be 1.0 m to
1.5 m away from the air supply grilles [53]. Loudermilk reiterated that no occupant should be located within the radius of the diffuser for air velocities exceeding 0.25 m/s and temperatures of more than 0.6 °C lower than the room temperature [54]. Fanger et al. summed it up by recommending that optimal supply air velocity and temperature conditions must be established to reduce thermal discomfort due to draught. The results of their study illustrated that the optimal supply air conditions depend on the distance between the occupant and the air diffuser [55].

2.2.3 Supply air flow rates

The air supply flow rate and temperature has substantial impact on the air diffusion, thermal comfort and indoor air quality in the ventilated space [56]. Lee et al. in a study to evaluate air distribution effectiveness with stratified air distribution systems found out that the discharge height, number of diffusers, supply air temperature, and total flow rate have major effect on the air distribution effectiveness [57]. Higher air supply flow rates are associated with increased draught risk even when used in conjunction with higher temperatures in IJV system [47], [58]. Varodoumpun and Navvab analysed the impact of terminal configuration in impinging jet ventilated room using the response surface methodology and discovered that the supply airflow rate was the most important factor influencing draught, followed by the shape of supply device and discharge height [35].

On the other hand, elevated room temperatures in stratum ventilated spaces facilitate the use of high air speeds without increasing the draught risk [26]. In a study to evaluate the effect of the interaction of human body and room airflow on thermal comfort under SV system, it was observed that elevating the air change rate from 7 to 15 air changes per hour (ACH) varied the downstream airflow pattern dramatically, from an uprising flow induced by body heat to a jet-dominated flow [59]. To avoid a larger vertical temperature difference, low supply velocities must be not be used in SV system. Low air supply velocity reduces the air jet length making it flow downwards more quickly to the lower parts of the ventilated space, hence, causing local thermal discomfort due to lower temperature at the ankle level [45]. The revision of the ISO 7730 incorporated the theory by Fountain and Arens’ that higher air speeds can be used to offset increased air temperature [60]. Arens’ also stated that for rooms with air temperatures higher than 22.5 °C, there is low risk of draught and high preference for more air movements [61]. The application of higher air speeds in the range of 0.6 m/s for room air temperatures
greater than 25 °C conforms to ASHRAE 55-2010 [62]. Increasing the airflow rate at constant supply air temperature improved both the heat removal and local contaminant removal efficiencies in a stratum ventilated office [63].

It can be contended that the airflow rate is directly linked to energy use. Higher airflow rates are associated with higher energy use and vice versa. The ability of the SV system to have higher energy performance is due to relatively low airflow rates used than in MV and DV systems[28], [33]. Increasing the supply flow rates enhances mixing and leads to higher ventilation efficiency in mixing ventilation system [64]. However, the use of constant flow rates may be the cause of many problems such as draught and existence of stagnation zones in the ventilated space. Fallenus et al. studied the effect of pulsating inflow on the ventilation efficiency in MV systems. The study concluded that the same effect of enhanced mixing that can be achieved by increasing the flow rate can be attained by applying a pulsating inflow [64]. Furthermore, Kabanshi et al. proposed the use of intermittent air supply systems as they create unsteady flows similar to that of natural wind which improve cooling and reduce draught risk [65].

### 2.2.4 Supply air temperature

The supply air temperature has significant effect on the room air in any ventilation system. The supply air temperature enables ventilation systems to be used either for cooling or heating purposes; the supply air temperature is lower than the room air temperature for cooling purposes and higher than the room air temperature in heating applications. The air supply temperature has huge influence on the thermal comfort and IAQ in the indoor environment. Chen et al. studied the influence of air supply parameters on indoor air diffusion. They discovered that the flow rate and air supply temperature had significant influence on the diffusion of air, thermal comfort and IAQ [56]. Temperature stratification in spaces ventilated by IJV systems has been linked to the influence of supply air temperature [37].

Besides, Tian et al. examined the impact of supply air temperature on the mean local age of air and thermal comfort in a stratum ventilated office. The results showed that when the supply air temperature was increased from 19 °C to 21 °C, the corresponding mean occupied zone temperature rose from 24.5 °C to 26.5 °C [24]. Improvements in the inhaled air quality for the
occupants were also noted when supply air temperature was increased from 19 °C to 21 °C as evidenced by shorter mean age of air from 475 s to 443 s [24]. In a similar experimental study of the SV system, Tian et al. found that the supply air temperature of 21 °C provided better thermal comfort than when air is supplied at 19 °C [25]. In addition, Cheng et al. considered the effect of the air supply flow rate, supply air temperature and the room temperature on thermal comfort in a stratum ventilated atmosphere using a test chamber. The results indicated that at the room temperature of 27 °C, increasing the air change rate from 7 to 17 ACH had minimal influence on thermal sensation and draught [66]. This was attributed to high preference for air movements at higher room temperatures. Additionally, to reduce draught risk, they recommended that the supply air temperature should not be below 20 °C [66]. Lin et al. reiterates that since fresh air is directly supplied to the breathing zone in SV systems, the air temperature gradient should be low and the supply air temperature above 20 °C to prevent thermal discomfort [28].

2.2.5 Heat loads

Heat sources in the ventilated space affect the penetration of the incoming air flow by the creation of convection and buoyancy effects. Depending on the momentum of the incoming air jet, the distance the jet penetrates the room is greatly determined by heat sources. The presence of furnishings, internal heat sources and people limits the penetration distance of the airflow in the ventilated space [17], [65]. Thus, high momentum jets are used in IJV systems to allow for further penetration past heat loads as compared to relatively low momentum jets of the DV systems [37], [67], [68]. In a quest to overcome the negative effect of heat loads on ventilation effectiveness, Rees and Haves combined the DV system with chilled ceiling to provide both ventilation and cooling for larger sensible loads [69]. In a similar study, Cehlin et al. proposed the use of active chilled beams to provide adequate thermal comfort and better air quality in open plan offices characterised by large heat loads without incurring the draught risk. It was noted that the existence of large heat loads demand use of high flow rates which have been associated with higher draught risks in conventional ventilation systems [7]. In the same study, the effect of the heat load distribution was found to be a major parameter in the creation of indoor airflow and indoor climate in the ventilated space [7].
3 Theory

Ventilation systems are essential in ensuring good IAQ and thermal comfort. The ventilation process is to a greater extent influenced by the flow rate and air flow pattern created by the ventilation system in any ventilated space [70]. When a ventilation system creates improper air flow patterns, insufficient ventilation can result even if high flow rates are used. Consequently, a lot of energy is used for low quality ventilation. Therefore, an appropriate amount of conditioned air coupled with effective air distribution system are essential for creating comfortable conditions, removing contaminants and reducing ventilation systems operational costs [70].

The ventilation effectiveness of a system must be evaluated to ascertain its suitability for use in any space. Different ventilation indices are available for the assessment of the ventilation effectiveness of the ventilation system. Ventilation effectiveness indices are universal so that they can be used in any ventilation system while at the same time retaining the mutual comparability in their values [51]. Ventilation effectiveness indices are important in indicating the existence of unwanted occurrences such as short-cut ventilation. Short-cut ventilation, mainly determined by the air distribution system and geometry of the ventilated space, is a phenomenon where much of the supplied air reaches the extract air terminal without passing through the occupied zone [70].

There are broadly two ways to evaluate the ventilation effectiveness of the ventilation system. The first method involves assessing the ability of the system to remove internally generated air-borne contaminants by comparing the concentration in the exhaust air and the mean concentration in the ventilated space. The second method which is more general measures the ventilation effectiveness by the system’s ability to exchange the air in the ventilated space [70]. The choice of the ventilation index to be used depends on the specific objective of the ventilation process which can include: To achieve a certain thermal comfort, to renew the air in the room, to remove a contaminant or to minimise the infection risk [19]. In this study the second approach is adopted.
3.1 Ventilation effectiveness indices

Ventilation effectiveness indices give good indications on the level of thermal comfort, uniformity, and ventilation effectiveness for the room [71]. Mundt et al. categorized the ventilation effectiveness indices into two classes [70]:

a) Indices representing the ability of the system to exchange the air in the ventilated space. This group consists of the air change efficiency ($\varepsilon_a$) and the local air change index ($\varepsilon_{a,p}$)

b) Indices demonstrating the ability of the system to remove air-borne contaminants from the ventilated space. This category comprises the contaminant removal effectiveness ($\varepsilon_c$) and the local air quality index ($\varepsilon_{c,p}$).

For the purpose of this study, the local air change index and the air change efficiency were used to evaluate the ventilation effectiveness of the SV system. Therefore, only the theory pertaining to these two indices is presented in this report.

3.1.1 Air change efficiency ($\varepsilon_a$)

The air change efficiency indicates how fast the air in a ventilated space is exchanged for ‘new’ supply air in comparison with the theoretically fastest rate with the same ventilation flow. This index is considered as a measure of how efficiently the supplied air is distributed in the ventilated space. The index quantifies the ability of the ventilation system to renew the air in the ventilated space and is useful at the design stage when both the location and type of the contaminant are unknown [51], [70]. Mathematically, the air change efficiency can be determined from Equation (1).

$$
\varepsilon_a = \frac{\tau_n}{2 \times \tau_{mean}} \times 100 \ [\%]
$$

(1)

where $\varepsilon_a$ is the air change efficiency, $\tau_n$ is the nominal time constant (h); $\tau_{mean}$ is the mean age of air in the ventilated space (h) [70].

The nominal time constant, $\tau_n$ is the shortest time needed to replace the air volume of the enclosed space V using a certain ventilation flowrate $q_v$ [51]. Equation (2) gives the nominal time constant.
\[ \tau_n = \frac{V}{q_v} \]  

(2)

where, \( V \) is the room volume (\( m^3 \)) and \( q_v \) is the ventilation rate (\( m^3/h \)).

The mean age of air, \( \tau_{mean} \) is the mean value of the local age of air of all the air in the ventilated space. When the air in the ventilated space is perfectly mixed, the nominal time constant equals the space mean age of air to give the reference value of 50% for the air change efficiency. Air change efficiencies of less than 50% may indicate the existence of stagnation zones and occurrence of the short-cut ventilation phenomenon. It is recommended that the ventilation installations be rearranged if the air change efficiency is below 40%. Ideal piston flow has air change efficiency equal to 100%, while MV system has \( \varepsilon^a = 50\% \), and the \( \varepsilon^a \) for the DV system lies between 50 and 100% [70].

The air change efficiency depends only on the room airflow pattern. For instance, in a stationary, isothermal flow and with a high Reynolds number, the air change efficiency depends on the air inlets and outlets and on the geometry of the room [51]. The air change efficiency can be determined by the tracer gas technique.

### 3.1.2 Local air change index (\( \varepsilon^a_p \))

The local air change index is a measure of how fast the supply air reaches a certain point in a ventilated space. It can be expressed as the ratio of the nominal time constant and the local mean age of air. Equation (3) can be used to calculate the local air change index.

\[ \varepsilon^a_p = \frac{\tau_n}{\tau_p} \]  

(3)

where \( \varepsilon^a_p \) is the local air change index, \( \tau_n \) is the nominal time constant (h), and \( \tau_p \) is the local mean age of air (h) [70].

The local mean age of air indicates the average time that the supplied air reaches a certain point \( P \), in the ventilated space. It is regarded as a measure of local air quality with longer mean age of air indicating poor local air quality. The local air change index is inversely proportion to the local mean age of air. Thus, the shorter the local mean age of air, the higher the local air change index and the better the air quality [70]. For perfectly mixed air, the local air change index is
approximately equal to unity in all positions in the space because the local mean age of air equals the nominal time constant. Tracer gas methods can be applied in the determination of the local air change index [70]. When the concentration decay method is used, the local mean age of air at a point is obtained by using Equation (4)

$$\tau_{p\text{mean}} = \frac{1}{C_p(0)} \int_0^\infty C_p(t) dt,$$ \hspace{1cm} (4)

Where $C_p(0)$ is the initial concentration of tracer gas at time 0, and $C_p(t)$ is the concentration of the tracer gas at time $t$ [72]. Using a similar approach, the mean age of air for the entire space, $\tau_{\text{mean}}$ is calculated using Equation (5):

$$\tau_{\text{mean}} = \frac{\int_0^\infty tC_0(t) dt}{\int_0^\infty C_0(t) dt}$$ \hspace{1cm} (5)

### 3.2 Heat removal effectiveness

The efficiency of the system to remove heat is assessed using the heat removal effectiveness parameter (HRE). HRE parameter has been used by many researchers to assess their systems’ performance [7], [72]. It is determined using Equation (6) for cases investigating the cooling ability of the system.

$$HRE = \frac{(T_o-T_l)}{(T_{0.1,0.6,1.1}-T_l)}$$ \hspace{1cm} (6)

where, $T_o$ is the exhaust air temperature, $T_l$ is the supply air temperature and $T_{0.1,0.6,1.1}$ is the arithmetic mean air temperature at heights 0.1, 0.6 and 1.1 m.

### 3.3 Temperature effectiveness

For cases involving the analysis of the system’s ability to provide heating, the heat adding capacity of the system is assessed using the temperature effectiveness parameter and is defined as shown in Equation (7)
\[ \varepsilon_T = \frac{(T_i - T_o)}{(T_i - T_{0.1,0.6,1.1})} \]  

(7)

where, \( T_o \) is the exhaust air temperature, \( T_i \) is the supply air temperature and \( T_{0.1,0.6,1.1} \) is the arithmetic mean air temperature at heights 0.1, 0.6 and 1.1 m.

If \( \varepsilon_T > 1 \), it indicates that the temperature in the occupied zone is higher than at the outlet. If \( \varepsilon_T < 1 \), shows that the temperature in the occupied zone is lower than the temperature at the outlet which indicates underutilization of the heat from the ventilation system. The optimal condition is represented by \( \varepsilon_T = 1 \).

3.4 Dimensionless air temperature

To compare the vertical air temperature profile between different cases, the dimensionless air temperature (DAT) is used [7], [72]. The DAT is defined as shown in Equation (8)

\[ DAT = \frac{T_a - T_i}{T_o - T_i} \]  

(8)

where, \( T_a \) is local air temperature at a point [°C], \( T_i \) is the inlet air temperature [°C] and \( T_o \) is the outlet air temperature [°C].

Furthermore, the interaction of the buoyant and momentum forces in the airflows created by the system was analysed using the Archimedes number. Since the performance of the stratified ventilation system is greatly influenced by the relationship of the buoyant and momentum forces. The inlet Archimedes number (Ar\(_i\)) has been prevalently utilized to examine the relative significance of the buoyant and inertia forces in building airflows [72]–[74]. The number which combines the supply air velocity and room air temperature difference is given by Equation (9):

\[ Ar_i = g \times \frac{(T_r - T_i)}{T_r} \times \frac{\sqrt{A_e}}{(u_{in})^2} \]  

(9)

where, \( g \) is the gravitational acceleration [m/s\(^2\)], \( T_r \) [K] is the mean air temperature in the centre of the room at 1.7 m from the floor, \( T_i \) [K] is the mean supply air temperature, \( A_e \) is the inlet supply opening area [m\(^2\)], and \( u_{in} \) is the nominal inlet air velocity [m/s].
3.5 Thermal comfort evaluation

There are several indices that can be utilised to assess the thermal comfort in an indoor environment. Indices such as the PMV and PPD illustrate the occupants’ perception of the thermal environment. Dissatisfaction about the thermal comfort in the indoor environment can result from unwanted cooling or heating of one part of the body called local thermal discomfort. According to ISO 7730 [75], local thermal discomfort can be caused by high vertical temperature difference between the head and ankles, radiant temperature asymmetry, too warm or too cold floor and draught. Sedentary workers with light activity are more sensitive to local discomfort than those at high activity level. In this study, two indoor climate indices: Draught rate (DR) and percentage dissatisfied (PD) are used to assess the performance of the ventilation system.

3.5.1 Draught rate

High air movements have been associated with increased risk of draught. To quantify draught, the DR index which indicates the discomfort due to undesirable cooling of the occupant’s body is used. The index represents percentage of dissatisfaction due to draught and is a function of the air temperature, air velocity and turbulent intensity in the occupied space. The index is determined from Equation (10):

\[
DR = (34 - T_a)(u_a - 0.05)^{0.62}(0.37 \times u_a \times T_u + 3.14)
\]

For \( u_a < 0.05 \text{m/s} \) use \( u_a = 0.05 \text{m/s} \)

For \( DR > 100\% \) use \( DR=100\% \)

where, \( T_a \) is the local temperature, \( u_a \) is the mean air velocity and \( T_u \) is the local turbulent intensity [72], [75].

In the study the turbulence intensity, \( I_p \) was obtained by employing Equation (11)

\[
I_p = \frac{u_{rms}}{u_{mean}} \times 100 [\%]
\]
where, $u_{rms}$ is the root mean square of the turbulent velocity fluctuations, and $u_{mean}$ is the mean velocity.

### 3.5.2 Percentage dissatisfied

When a large vertical temperance difference occurs between the head and ankles, the human body experiences thermal discomfort. ISO 7730 [75] recommends a vertical temperature difference less than 3°C for occupants to experience comfortable conditions. The PD is a function of the vertical temperature difference and is calculated from Equation (12):

$$PD = \frac{100}{1 + \exp(5.76 - 0.856\Delta T_{0.1-1.1})}$$  \hspace{1cm} (12)

where, $\Delta T_{0.1-1.1}$ is the vertical temperature difference between the ankle level (0.1m) and the neck level of a seated person (1.1m) used in the study [75].

Equation (12) should only be used when $\Delta T_{0.1-1.1} < 8 \degree C$ because it is derived from original data using logistic regression analysis. The thermal comfort in an environment can be categorised in any of the three categories A, B, or C. ISO 7730 summarises the comfort conditions in relation to selected thermal condition indices as shown in Table 1.

---

**Table 1 Categories of thermal environments [75]**

<table>
<thead>
<tr>
<th>Category</th>
<th>Thermal state of the body as a whole</th>
<th>Local Discomfort</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>PPD%</td>
<td>PMV</td>
</tr>
<tr>
<td>A</td>
<td>&lt; 6</td>
<td>0.2 &lt; PMV &lt; 0.2</td>
</tr>
<tr>
<td>B</td>
<td>&lt; 10</td>
<td>0.5 &lt; PMV &lt; 0.5</td>
</tr>
<tr>
<td>C</td>
<td>&lt; 15</td>
<td>0.7 &lt; PMV &lt; 0.7</td>
</tr>
</tbody>
</table>

---
3.6 Tracer gas method

The tracer gas techniques have been used for over three decades to assess the ventilation effectiveness in mechanically ventilated buildings and the infiltration rate in naturally ventilated buildings.

Sherman defined a tracer gas as “an idealized substance used to tag volumes of air so as to be able to infer their bulk movement” [76]. Several gases can be used as tracer gases including nitrous oxide (N$_2$O), Sulphur Hexaflouride (SF$_6$), halogenated hydrocarbons such as hexafluorobenzene (C$_6$F$_6$) and perfluorocarbons (PFC). Carbon dioxide (CO$_2$) can also be used as tracer gas [20]. There are important features that a tracer gas must possess, namely: Easily detectable, chemically stable and inert, non-toxic and low concentration in the ambient air. Other features include high availability, almost same density as air, low environmental burden, low cost, and no adsorption by the walls, ceiling, or furnishings in the room [20], [77].

Nevertheless, certain undesirable features of some tracer gases limit their use. For instance, the use of CO$_2$ as a tracer gas is limited by its smaller range of application and significant background concentrations in the ambient air (about 400 ppm). The toxic nature of N$_2$O has restricted its use only to unoccupied spaces. Other limitations of N$_2$O pertain to its easy adsorption to room surfaces at concentrations below 1000 ppm, and high solubility in water which can lead to overestimation of the air change rate in very airtight rooms [20]. Halogenated hydrocarbon-based tracer gases which are mostly used in constant injection method using passive sampler technique have their own shortcomings. They are limited by their affinity to attach to surfaces and temperature-dependent emission rate [78].

Although the density of SF$_6$ gas is about five times higher than that of air, the difference does not cause any distortions in the air change rate measurements conducted using SF$_6$ gas in practice. Moreover, SF$_6$ gas has relatively higher global warming potential owing to low degradation rates in the atmosphere. As a matter of caution, SF$_6$ should be used in sparing concentrations because it belongs to the climatic relevant greenhouse gases category [20], [79]. Notwithstanding its limitations, SF$_6$ gas is the most frequently used tracer gas globally due to its high stability and high decomposition temperature of around 550 °C. The concentration of SF$_6$ gas in the ambient air is low, estimated at 1 ppm. SF$_6$ can be measured with high accuracy over a wider concentration range and is suitable for use in occupied buildings [20].
Three tracer gas techniques exist: Constant injection method, Constant concentration method and the Concentration decay method. In this study, the concentration decay method using SF$_6$ as a tracer gas was used. A detailed discussion is presented on the decay method; only an outline of the other methods is presented because they lie outside the scope of the study.

3.6.1 Constant injection method

In this method, a defined quantity of tracer gas is constantly released into the space of interest for over a period. With the passage of time, the tracer gas concentration increases and reaches an equilibrium concentration at which air samples are taken and the tracer gas concentration determined from the samples.

3.6.2 Constant concentration method

In the constant concentration method, the tracer gas is continuously released in the room air and is thoroughly mixed until a predefined concentration is reached. Automated dosing and control systems keep the tracer gas concentration constant at the predefined concentration level throughout the entire measurement process. At constant tracer gas concentration conditions, the air supply is proportional to the tracer gas supply rate. The air supply rate is calculated from the ratio of the tracer gas supply to the tracer gas concentration. Given the room volume, the air change rate can be calculated from this ratio. This method offers the ability to detect even short-term changes of air supply. However, the huge costs of the technical equipment involved in the method limits its use in indoor air quality examinations [20].

3.6.3 Concentration decay method

The simplest and most frequently used tracer gas technique is the concentration decay method. The technique requires only the measurements of the relative tracer gas concentrations rather than the absolute concentrations, thus less quantity of tracer gas is required [77]. In the concentration decay method, a specific amount of tracer gas is injected into the room for a short
period of time. The tracer gas is thoroughly mixed with the help of fans and the decline in the concentration measured at regular time intervals. When complete mixing is achieved, the decay in the concentration of the tracer gas is exponential. The faster the rate at which the air/tracer gas mixture is replaced with fresh air, the higher the air change rate. Typical exponential decay curve of the tracer gas concentration with time obtained when using the concentration decay method is shown in Fig.2.

![Graph showing exponential decay curve](image)

*Figure 2 Relationship between tracer gas concentration and time [20]*

The time after which the air change cycle is completed is referred to as the nominal time constant, $\tau_n$. The minimum decay time requirement for the decay method ranges from 5 to 10 hours or more. However, this requirement is hardly fulfilled for applications with air change rates lower than 0.2 h$^{-1}$ [20].

The decay method offers more flexibility as higher air change rates of the order 10 ACH or even higher if data loggers are incorporated in the measurements can be evaluated [20]. If the ventilation rate of a certain space is constant and the air in the space is well mixed, the concentration, $C$, of the tracer gas is assumed the same in all the parts of the space and will decay exponentially with time and can be evaluated using Equation (13). Caution must be taken to use automated control systems to ensure that the mixing fans are only operational in the first few minutes after injecting the tracer gas to avoid interfering with the airflow pattern created by the ventilation system under study.
\[ C = C_s + (C_0 - C_s) \cdot e^{-n \cdot t} \]  \hspace{1cm} (13)

where \( C \) is the tracer gas concentration, \( C_s \) is concentration of the tracer gas in the supply air, \( C_0 \) is initial concentration directly after injection and mixing of the tracer gas, \( n \) is the air change rate and \( t \) is time [20].

But practically, there is negligible amount of tracer gas in the supply air except when \( \text{CO}_2 \) is used as a tracer gas. Equation (13) then reduces to the form in Equation (14) for gases with negligible concentration in the supply air.

\[ C = C_0 \cdot e^{-n \cdot t} \]  \hspace{1cm} (14)

In places where the ventilation rate fluctuates a lot with time, may be due to weather conditions, several repeated measurements are conducted to get an average value of ACH with acceptable accuracy.

Uncertainties associated with tracer gas techniques include significant leakage of tracer gas, problems due to condensation, chemical reaction or solution in water of the tracer gas. Improper mixing of the tracer gas in the space under investigation also greatly contributes to the errors in the measurements. Errors due to improper mixing of the tracer gas can be minimised by multi-point injection strategies [77]. For high accuracy with the tracer gas measurements, Buggenhout et al. emphasised the significance of proper mixing and recommended the outlet as the best overall position for tracer gas sampling point [80].
4 Method

4.1 Materials

The following materials were used in the study: INNOVA 1302 gas monitor and INNOVA 1303 multi-channel sampling unit operated by INNOVA 7260 software, SF₆ gas, two axial fans, balloon, thermocouples, smoke generator, cameras, omnidirectional thermistor anemometer type CTA88, two thermal mannequins (100 W each), two metallic cylinders and two 75 W lamps.

4.2 Experimental set up and procedure

The study was conducted in one of the test rooms in a laboratory at the University of Gävle. The mock-up office closely resembled the features of the modern office with one exterior wall and three interior walls. The dimensions of the test room were 7.2 m × 4.1m × 2.7 m, (L × W × H). The composition of the wall from inside to outside was: 15 mm wood sheet, 35 mm air gap, 15 mm wood sheet, 190 mm insulation and 5 mm wood sheet. The floor and main ceiling were insulated by a 150 mm thick layer of mineral wool and covered by a layer of plastic sheet to minimise air infiltration. The test room had a suspended ceiling made of 600 mm × 600 mm fiberglass tiles which were hanging 310 mm below the main ceiling. It had three windows located on the northern wall of the test room built in direct connection with a climate chamber.

The mock-up office mimicked a shared office with two workstations. The workstations which were placed about 3.6 m from the air inlet terminals comprised a table, chair and a seated thermal mannequin each. Each thermal mannequin was made of galvanized tube of 0.32 m diameter covered with fabric to emit the same radiation as an ordinary human being. It had same area as the human body and produced 100 W. The computer at each workstation was simulated by a 75 W lamp placed inside metallic cylinder open at one end. The total internal heat generated was 350W, i.e., \{(2 × 100 W) + (2 × 75 W)\}. Figure 3 shows the experimental set up of the mock-up office.
Fig. 3 clearly shows the positions of the workstations and the heat sources in relation to the air supply terminals. It also indicates the positions of the measurements for temperature and velocity along with the tracer gas sampling points. The tracer gas sampling points are denoted by letter T along with the position number, e.g., T-1, while the points for temperature and velocity measurement are denoted by P such as P-1.

The air distribution system had one main inlet which subsequently divided into two final inlets in the test room. Rectangular straight grill air inlet terminals with dimensions 175 mm × 127 mm each placed at 1.3 m from floor surface to centreline of the grilles. The air inlet terminals were placed at the corners of the wall adjoining the south wall. There was only one exhaust terminal located in the ceiling near the northern wall of the room. Figure 4 illustrates the details of the air supply terminals.
Fifteen cases were investigated involving three nominal supply air temperature setpoints: 17.6°C, 21.0 °C and 25.3 °C. At each supply air temperature, five different airflow rates were investigated, i.e., 30, 40, 50, 60 and 70 l/s. The details of the inputs to the different cases are shown in Table 2.
Table 2 Conditions for different cases

<table>
<thead>
<tr>
<th>Case</th>
<th>Nominal Supply flow rate [L/s]</th>
<th>Inlet Temperature [°C]</th>
<th>Exhaust air temperature [°C]</th>
<th>Occupant [W]</th>
<th>Equipment [W]</th>
<th>( U_{in} ) [m/s]</th>
<th>( Ar_i [\times 10^{-3}] )</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1_SV</td>
<td>2 \times 15</td>
<td>17.7</td>
<td>23.6</td>
<td>2 \times 100</td>
<td>2 \times 75</td>
<td>0.75</td>
<td>49</td>
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<td>22.6</td>
<td>2 \times 100</td>
<td>2 \times 75</td>
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<td>22.6</td>
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<tr>
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<td>22.4</td>
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<td>2 \times 75</td>
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<td>13.5</td>
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<td>1.01</td>
<td>19</td>
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<tr>
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<td>24.4</td>
<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.26</td>
<td>11.4</td>
</tr>
<tr>
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<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.52</td>
<td>7.1</td>
</tr>
<tr>
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<td>21.2</td>
<td>24.1</td>
<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.77</td>
<td>4.3</td>
</tr>
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<td>24.1</td>
<td>2 \times 100</td>
<td>2 \times 75</td>
<td>0.75</td>
<td>9.5</td>
</tr>
<tr>
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<td>24.1</td>
<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.01</td>
<td>4.5</td>
</tr>
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<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.26</td>
<td>2.3</td>
</tr>
<tr>
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<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.52</td>
<td>1.4</td>
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<tr>
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<td>2 \times 100</td>
<td>2 \times 75</td>
<td>1.77</td>
<td>30.6</td>
</tr>
</tbody>
</table>

Table 2 shows the nominal supply flow rate and temperature, the sources of internal heat generation, the effective velocity and the Archimedes number for different cases involved in the examination of the performance of the stratum ventilation system.

4.2.1 Tracer gas measurements

The concentration decay tracer gas method using SF\(_6\) gas as was used in the study. The tracer gas measurements were conducted using the INNOVA 1302 gas monitor and INNOVA 1303 multi-channel sampling unit being augmented by the INNOVA 7260 software installed on the laboratory computer. The INNOVA multi-channel sampling unit has six channels which enabled the measurement of the concentration of the tracer gas at six different positions in the room. The sampling points were strategically selected with one point located in the exhaust. The remaining sampling points were all placed at the breathing zone level i.e., height of 1.1 m from the floor surface. Positions T-1, T-2, and T-5 were placed in a straight line along the centreline of the room. Position T-4 was on the table in front of the thermal mannequin and position T-3 was placed on the right-hand side of one thermal mannequin. Position T-6 was
placed in the exhaust. Due to the limitation of the measuring equipment, no sampling was done near the other mannequin. However, this shortfall has no effect on the results since symmetry of the room was assumed. Before any tracer gas measurements, all the visible air passages were sealed and then the room was tested for any leakages; it was found to be acceptably airtight for tracer gas measurements. Figure 5 shows the configuration of the tracer gas sampling points in the test room.

The SF$_6$ gas was injected into the test room at a concentration of around 350 ppm and the automatic circulating movable fans were operated for the first three minutes to ensure thorough mixing. In order to enhance mixing, the SF$_6$ gas was injected at multiple points in the room at height about 1.8 m. Gas chromatography was used to measure the concentration of the gas in air samples. In each tracer gas test, air samples were collected via a pump connected to the gas chromatography unit. Measurements were performed two times for about four hours each session and the average deviation between the two measurements was less than 2%.

Figure 5 The arrangement of the tracer gas sampling points and the temperature/velocity measuring points in the test room.
4.2.2 Temperature and velocity measurements.

Temperature and velocity at selected points were measured using the low-velocity Omni-directional thermistor anemometer type CTA88. The thermistors were connected to a multi-channel logger and were read on the personal computer on which LabVIEW program version 2009 was installed. A total of seven different positions in the room were used for the measurement of the temperature and velocity using the thermistors. For positions represented by points P-1 to P-4, measurements were performed at four different heights, i.e., 0.1, 0.6, 1.1 and 1.7 m from the floor surface. Positions P3 and P4 are near the windows and have the most effect of the climate chamber. For locations P-6 and P-7, measurements were taken at heights 1.1 and 1.7 m from the floor surface. At point P-5, measurements were taken at 0.1, 0.3, 0.6, 0.8, 1.1, 1.4 and 1.7 m from the floor surface. Fig.5 shows the positions in the room at which the air temperature and velocity were measured. All the thermistors were calibrated in a low-velocity calibration unit before use to ensure accurate results. The sampling interval for all measurements was 600 s. The velocity was measured with an accuracy of ±0.05 m/s excluding directional error with the response time of 0.2 s to 90% of a step change. The uncertainty of temperature measurements was ±0.2 °C with the response time of 12 s to 90% of value in still air. The temperature and velocity measuring instruments used in the study were previously used by other researchers at the University of Gävle [7], [65], [72].

The surface temperatures for the wall, ceiling and window were measured using the T-type (copper-constantan) thermocouples connected to an Agilent 34970A data logger and computer. The same type of thermocouples was used to measure the temperature supply air in the main inlet and the two final air inlets. Calibration of the thermocouples and logger was carried out before and after the measurements to ensure that the accuracy was within the expected range.
5 Results and discussion

5.1 Thermal conditions and flow patterns

The measured velocity, temperature, draught rate and the dimensionless air temperature (DAT) are presented for only two positions: At the centre of the room and on the workstation being designated by P5 and P6, respectively. The two positions are chosen because they closely reflect the conditions in the occupied zone in the rest room. Detailed measurements result for rest of the positions are found in the various appendices of this report.

5.1.1 Flow patterns

The representative figures for the flow patterns created by the ventilation system at the three-nominal supply setpoints using the air supply flow rate of 50 l/s are shown in Figure 6. It is clear from Fig. 6a and 6b, that supplying at 17.6 °C, the airflow quickly tends towards the floor and this tendency decreases with increase in supply temperature. Such behaviour of air inflow has been associated with the SV system in other studies [26]. When supply air temperature was set at 21°C, part of the air inflow remains suspended at the intersection point of the air flows due to relatively lower density than the room air as shown in Fig. 6c and 6d. This led to the creation of stagnation areas in the test room. For the nominal supply temperature of 25.3 °C, as shown in Fig. 6e and 6f, at supply air temperature higher than the room air, much of the supply air remains in the breathing zone and heights above the breathing zone. Much of the air passes direct from the supply air terminals to the exhaust terminal, thus does not reach the lower zones of the room. The created air flow patterns cause short cut ventilation and stagnation points of the room leading to low ventilation effectiveness [70].
Figure 6 Flow visualization: a) air flow patterns at 50 l/s and 17.6 °C from Camera 1, b) air flow patterns at 50 l/s and 17.6 °C from Camera 2, c) air flow patterns at 50 l/s and 21 °C from Camera 1, d) air flow patterns at 50 l/s and 21 °C from Camera 2, e) air flow patterns at 50 l/s and 25.3 °C from Camera 1, f) air flow patterns at 50 l/s and 25.3 °C from Camera 2
5.1.2 Velocity conditions

The velocity profiles for measured velocities at positions P5 and P6 for the three temperature set points for supply air and air flow rate are presented in Figure 7.

Fig. 7 illustrates a general trend of decreasing velocity with increase in height under the condition of nominal supply air temperature of 17.6 °C and 21 °C. Highest velocities are
recorded at ankle level (0.1 m) and lowest at breathing zone level. This behaviour is attributed to the airflow pattern created in which the relatively cool supply air tends towards the floor while still retaining the high supply momentum. This trend contradicts the results reported by Shao et al. [30] where the velocity increased with increase in height in the occupied zone. As can be seen in Fig. 7, values of velocity measured at the breathing zone of a seated sedentary worker in the occupied zone, i.e., at 1.1 m from the ground are all below 0.2 m/s. Lower velocities (less than 0.15 m/s) are measured at the work stations which are represented by Point P6. The velocity conditions in the occupied zone at the breathing zone level all satisfy the design criteria of the maximum mean velocity recommended by ISO 7730 [75].

Position P5 recorded relatively high velocities under all conditions because of its central location because the air inflow reached after reinforcement following the coalescing of two air streams from the two air inlet terminals. A slight increase in velocity is observed at around 1.4 m from the ground and can be due to the influence of direct air inflow from the inlets. A similar trend of increase of room air velocity at heights slightly greater than the discharge height has been observed in other studies [81].

When the supply air temperature was increased to 21.0 °C, a slight increase in the velocity was noted at height 1.4 m. The slight increase in the velocities recorded at 1.4 m can be attributed to the high temperature of supply air which prevents the inflow air from immediately tending towards the floor, but momentarily remaining in the breathing zone level. At point P5, the velocity measured at 1.4 m is higher than that at 1.1 m because they are a combination of the velocity of the air rising from the floor after getting heated and the component of the velocity that came directly from the supply terminals without passing on the floor.

The velocity measured when the supply air temperature was 25.3 °C are shown in Fig 7c. Unlike the trend observed in Fig. 7a and 7b, velocity increases with increase in height in the Fig 7c. This is due to the higher supply air temperature which creates a flow pattern in which much of the air from supply air terminals remain at the breathing zone level being upheld by the relatively cool room air below.

The common feature in the velocity recorded for all conditions shown in Fig. 7 was that it increased with increase in airflow rate at each supply air temperature set point, which is expected. Highest velocities result from use of 70 l/s and the least for 30 l/s. However, very low velocities are recorded for Position 6 in all the conditions and can be attributed to the obstruction created by the table in front of the thermal mannequin since the measurement points
were placed on the table at the workstations. Further details of the measured velocities at all points in the room at nominal supply air temperature of 17.6 °C, 21.0 °C and 25.3 °C are contained appendices A1, A2 and A3, respectively.

5.1.3 Temperature conditions

Alongside the velocity, temperature was measured using the CTAs. Temperatures recorded for positions P5 and P6 under the nominal supply air temperature setpoints of 17.5 °C, 21.0 °C and 25.3 °C are shown in Figure 8.

Figure 8 Vertical temperature distribution in the test room: a) vertical temperature distribution at position P5 and P6 at nominal supply air temperature of 17.6 °C, b) vertical temperature distribution at position P5 and P6 at nominal supply air temperature of 21.0 °C, c) vertical temperature distribution at position P5 and P6 at nominal supply air temperature of 25.3 °C
Fig. 8a and 8b illustrate that the temperature increases with increase in height above 1.1 m from the floor surface when the supply air temperature set points were 17.6 °C and 21.0 °C. This is because of the effect of the relatively cool air supplied into the room which cools the air at height 0.6 m, hence lower temperatures are recorded at 0.6 m. Higher temperatures occur at heights greater than 1.1 m due to decrease in velocity which in turn reduces the convective cooling at those points. At the nominal supply air temperature of 17.6 °C, the maximum and minimum measured room air temperatures were 23.7 °C and 20.7 °C, respectively. The average exhaust air temperature was 22.3 °C. At the breathing level (1.1 m), slightly lower temperatures are recorded and this result is similar to that obtained in other studies [25]. The reduction in the vertical air temperature gradient with increase in distance from the air supply terminal has been reported [81]. Tendency towards stratification is observed at heights above 1.1 m as indicated in Fig. 8a and 8b with the lowest air flow rate exhibiting more stratification. Among the cases investigated, C1_SV recorded highest vertical temperature difference between 0.1 m and 1.1 m was about 0.75 °C while case C5_SV the least of nearly 0.1 °C. The temperature stratification was mainly due to effects of buoyancy and location of the exhaust air terminal. This smaller vertical temperature difference between head and ankle levels, and the existence of lower temperature in the breathing zone level than in the upper levels of the occupied zone in SV system have been reported [23], [26], [39], [81].

When the nominal supply air temperature was increased to 21.0 °C, the maximum and minimum room air temperatures also increased to 25.7 °C and 23.7 °C, respectively. The average exhaust air temperature was 24.7 °C. The stratification phenomenon is maintained, but smaller vertical air temperature difference occurs of 0.5 °C for Case C6_SV and about 0.25 °C for Case C10_SV. At high air flow rates using the nominal supply air temperature of 21°C, the temperature distribution is like a MV system which has nearly uniform temperature distribution [67].

However, when the nominal supply air temperature was 25.3 °C, a trend of increasing temperature from the ankle level to the head level was observed. The vertical air temperature difference between the 0.1 m and 1.1 m was highest for case with lowest air flow rate, i.e., Case C11_SV which had 1.25 °C and least with case C15_SV of less than 0.1 °C. This can be explained in part due to the decrease in velocity with increase in height and the higher supply air temperature. The maximum and minimum room air temperatures were 24.5 °C and 22 °C, respectively.
In the investigation of the system’s ability to provide cooling, the influence of air flow rates was observed; lower temperatures were measured at higher air flow rates. Thus, Cases C5_SV and C10_SV showed the lowest temperatures while cases C1_SV and C6_SV recorded the highest temperatures in the cooling mode as shown in Fig. 8a and 8b. Relatively larger vertical air temperature difference occurs at lower air flow rates due to lower air velocity and this behaviour in SV has been reported in other studies [45]. In assessing the ability of the system to provide heating in winter climatic conditions, the climate chamber was used. The climate chamber was set at -5 °C and the nominal supply air temperature raised to 25.3 °C. The opposite was observed where the temperature increased with increase in air flow rate with Case C15_SV recording highest temperature and C11_SV the least as indicated in Fig. 8c. The general trend was that greater vertical air temperature difference between 0.1 and 1.1 m occurred for the lowest airflow rate at all the nominal supply air temperature setpoints. The trend in temperature at point P6 is that it decreases with increase in height. This behaviour mimics the reverse temperature gradient attributed to the SV system [33], [81]. Detailed temperature measurement results for all the positions in the room at nominal supply air temperature of 17.6 °C, 21.0 °C, and 25.3 °C are contained in Appendices B1, B2 and B3, respectively.

5.1.4 Dimensionless air temperature

In order to compare the vertical air temperature profiles between different cases when assessing the ability of the system to provide cooling, the DAT is used. Positions P5 and P6 are presented as representative positions for the nominal supply air temperature of 17.6 °C and 21.0 °C only. DAT is not used for the heating mode analysis. Figure 9 shows the DAT profiles for the different cases at the stated two positions.
From Fig. 9a, it is illustrated that the DAT was higher for lower air flow rates and it decreased with increase in air flow rates. Higher vertical air temperature gradients when the supply air temperature is increased from 17.6 °C to 21.0 °C as shown in Figure 9. It can be deduced from Fig.9 that stratification increases at heights greater than 1.1 m because of the relatively higher temperatures experienced. Detailed results of the DAT for the setpoints of nominal supply air temperature of 17.6 °C and 21.0 °C are contained in appendix C1 and C2, respectively.

5.1.5 Draught rate conditions

The draught rate was used to assess the local discomfort due to air movements. Figure 10 shows the draught rate at positions P5 and P6 in the room under three conditions of nominal supply air temperature.
The draught rate percentage follows the trend of the magnitude of the velocity where it is highest at ankle level and lowest at 1.7 m height for nominal supply air temperature of 17.6 °C. Higher draught rate levels in SV systems have been associated with supply air temperatures below 21 °C [66]. When the nominal supply air temperature was at 17.6 °C, the maximum draught rate was 20% and occurred at ankle level in position P5 as shown in Fig.10a. This is due to the high velocity experienced at the position due to amalgamation of the two air streams from the inlet terminals. The draught is negligible at the workstations represented by positions P6 and is due to very low velocities experienced at the point. As contained in Table 1, the draught conditions prevailing in the room conform to ISO 7730 Category B of the thermal environments [75].

Figure 10 Draught levels in the test room: a) draught levels at position P5 and P6 at nominal supply air temperature of 17.6 °C, b) draught levels at position P5 and P6 at nominal supply air temperature of 21.0 °C, c) draught levels at position P5 and P6 at nominal supply air temperature of 25.3 °C
Increasing the nominal supply air temperature to 21.0 °C showed a slight decrease in draught rate, but follows the trend observed in Fig. 10a., i.e., it decreases with increase in height. However, a slight increase is observed at height of 1.4 m. The maximum draught rate which occurs at the ankle level is 16% which is lower than that obtained with the nominal supply air temperature of 17.6 °C as shown in Fig. 10b. This is because cooler heavier air possesses higher momentum at the same conditions of air flow rate than relatively warmer air which causes it quickly tends towards the floor. The warmer air momentarily remains at the breathing zone level and contributes to the draught being experienced at this level.

At the nominal supply air temperature of 25.3 °C an increase in the draught rate with increase in height was noted. This was as a result of the warmer air being introduced into the room tending upwards and being supported by the relatively cooler room air below it. The maximum draught rate obtained was about 12% which is the least among the nominal supply air temperature used as indicated in Fig. 10c. The draught rate levels decreased with increase in supply air temperature. All the cases tested under the three settings of the nominal supply air temperature satisfy Category B of the draught rate design requirements espoused by ISO 7730 [75]. Detailed draught rate for all cases and positions in the room under the setting of nominal supply air temperature of 17.6 °C, 21.0 °C and 25.3 °C are contained in appendices D1, D2 and D3, respectively.

5.2 Ventilation effectiveness

The ventilation effectiveness of the system was analysed by means of the heat removal effectiveness parameter, percentage dissatisfied, local air change index and the air change efficiency.

5.2.1 Heat removal effectiveness

The heat removal effectiveness parameter was used to assess the cooling ability of the system. Table 3 shows the average values of the heat removal efficiency for all the cases investigated at nominal supply air temperature of 17.6 °C and 21.0 °C setpoints.
Table 3 Average temperature effectiveness in locations P1-P7 for all cases at respective nominal supply air temperature setpoints: 17.6 °C and 21.0 °C.

<table>
<thead>
<tr>
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<th>Position</th>
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<th>HRE&lt;sup&gt;2&lt;/sup&gt;</th>
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<tr>
<td></td>
<td>P1</td>
<td>P2</td>
<td>P3</td>
</tr>
<tr>
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<td>1.20</td>
</tr>
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<td>C2_SV</td>
<td>1.19</td>
<td>1.20</td>
<td>1.20</td>
</tr>
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<td>1.18</td>
</tr>
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</tr>
<tr>
<td>C7_SV</td>
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<td>1.24</td>
</tr>
<tr>
<td>C8_SV</td>
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<td>1.17</td>
<td>1.20</td>
</tr>
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<td>1.07</td>
</tr>
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<td>0.98</td>
<td>1.00</td>
</tr>
</tbody>
</table>

<sup>1</sup> Calculated by using the arithmetic mean air temperature of the heights 0.1, 0.6 and 1.1 m.  
<sup>2</sup> Calculated by using the arithmetic mean air temperature of heights 1.1 m only.

From Table 3, it is illustrated that the heat removal efficiency decreases with increase in air flow rates for cooling mode cases. At supply air temperature of 17.6 °C, higher efficiencies are obtained for lower air flow rates represented by cases C1_SV and C2_SV which closely match the result obtained by other laboratory studies[72]. This result suggests that low air flow rates can be used to provide cooling and is in line with some features of stratum ventilated spaces reported by some researchers [33], [39].

There is slight increase in the heat removal effectiveness for cases C6_SV to C8_SV and decrease for the remaining cases when the nominal supply temperature is increased to 21.0 °C. This result indicates the potential of energy saving that can be achieved by supplying air at low flow rates at relatively higher temperature, energy is saved by expending less for cooling the supply air in summer conditions. The heat removal effectiveness obtained in cases C1_SV, C2_SV, C6_SV and C7_SV shown in Table 3 are very close to those reported for the corner displacement and corner impinging jet ventilations systems [72]. In cooling mode, cases C1_SV, C2_SV, C6_SV and C7_SV provided higher heat removal efficiency.

The assessment of the system to provide heating is reported in Table 4.
Cases from C11-SV to C15_SV in Table 4 give the ability of the system to provide heating in winter conditions, thus the temperature effectiveness gives the indication of heat addition into the room. From Table 4, a trend of increasing heat addition effectiveness is observed with increasing air flow rates. This implies that more energy will be expended to heat the supply air because of high air flow rates used. However, all other points in the room except positions P6 and P7 which represent the workstations have the heat addition parameter less than 1. This indicates underutilization of the heat in the ventilation system and signals problems like short cut ventilation. Underutilization of the supply heat may result from short cut ventilation as indicated in Fig 6e and 6f. This result obtained is lower than the ventilation effectiveness obtained for both mixing and stratum ventilation systems in other studies [81]. This point of view is strengthened by the results of the air change efficiency shown in Table 6 which are predominantly below 50%. This illustrates that the system in heating mode performed less than the MV system. Therefore, this result reiterates findings of other researchers on the suitability of the stratum ventilation to provide heating in winter climatic conditions [26].

5.2.2 Percentage dissatisfied

The local discomfort due to the vertical temperature difference at the selected points in the room was analysed using the PD. Table 5 shows the average PD values for all cases at respective nominal supply air temperatures. The general trend of decreasing PD with increase
in air flow rates is observed in all the cases. This is because convective cooling is enhanced by increased air movements due to high velocities associated with high airflow rates. The high velocities can break the thermal boundary layer on the human body to bring favourable thermal sensation [25], [28]. This is because temperature stratification decreased as the air flow rates were increased as can be seen in Fig.8. From Table 5 Case C1_SV has highest PD while Case C5_SV the least. The PD obtained in all the cases conform to requirements of Category A of ISO 7730 on thermal environments [75].

Table 5 Average values of PD for all cases at respective nominal supply air temperature setpoints: 17.6 °C, 21.0 °C and 25.3 °C

<table>
<thead>
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<td>1.21</td>
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</table>

When the nominal supply air temperature was increased to 21 °C, there was a slight reduction in the PD for case using the least air flow rate, i.e., case C6_SV had lower PD than case C1_SV. This is attributed to the narrower temperature difference between the supply air temperature and the room air temperature. The impact of the room air temperature on the overall thermal sensation, local thermal sensation and draught in stratum ventilated spaces has been reported [66]. From Table 5, lower PD values near the workstations which are represented by position P5 are obtained. The existence of lower PD at the workstations are a result of the even mixing of the supply air and room air after being in the room relatively longer since they are located
further away from the air inlets. The obtained result about the PD in all positions satisfy the requirements of the Category A in ISO 7730 on thermal environments [75].

The system was tested for capacity to provide heating in winter simulated climatic conditions. The climate chamber was set at -5 °C and nominal supply air temperature increased to 25.3 °C. There was a significant increase in the PD values for all the cases at all positions in comparison with the cases in cooling mode. The increase in the PD values is as a result of the higher supply air temperature. Like the conditions obtained under nominal supply air temperature of 17.6 °C and 21.0 °C, the PD decreased with increase in air flow rate at 25.3 °C. Case C11_SV recorded the highest and Case C15_SV the lowest PD values. Despite the relatively high values obtained at supply air temperature of 25.3 °C, all positions satisfy the PD requirements of category A in ISO 7730 on thermal environments [75].

5.2.3 Local air change index and air change efficiency

To evaluate the overall quality of ventilation at individual positions and the whole room, the local air change index and air change efficiency were used, respectively. The local air change index and the air change efficiency determined by the concentration decay tracer gas technique are shown in Table 6.

Under the nominal supply air temperature of 17.6 °C, all the positions show the local air change index slightly greater than 1. Relatively higher local air change indices are obtained for low air flow rates (from C1_SV to C3_SV) and so is the case with the air change efficiency. This trend illustrates that SV configuration performs better at lower flow rates. Similar results were reported by other researchers [28][32][24]. The air change efficiency for the mixing ventilation system is 50 , for displacement system lies between 50 and 100 , hence the system at this supply air temperature system had the features of MV system [70]. The proposed performance and design guidelines for the SV system stipulate that for better performance of the system, the supply air temperature should not below 21 °C for the room air temperature range of 25.5-27 °C [26]

When the supply air temperature was increased to 21.0 °C, similar values for the local air change index were obtained as those at 17.6 °C. However, lower values for the air change efficiency slightly below 50 were obtained. This gave an indication of some unwanted
ventilation occurrences like short-cut ventilation and possibly existence of stagnation points within the room. The relatively high supply temperature created a flow pattern where some portion of the supplied air did not pass through the lower parts of the room but flowed directly to the exhaust terminal. In this study, only the supply air temperature was increased without any corresponding increase in the room air temperature. The increase in the supply air temperature caused a smaller difference between the room air temperature, which caused a reduction in the heat transfer from the room air to the fresh incoming air and could be the reason for the poor performance of the system [52]. Thus, it can be contended that higher supply air temperature without a corresponding increase in the room air temperature leads to low ventilation effectiveness for the SV configuration [26]. This point of view is further strengthened by air change efficiency values obtained when the supply air temperature was increased to 25.3 °C.

Table 6 Local air change index and air change efficiency

<table>
<thead>
<tr>
<th>Case name</th>
<th>Nominal Supply Temp °C</th>
<th>Air flow rate (l/s)</th>
<th>Local air Change index</th>
<th>Air change efficiency [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>T1</td>
<td>T2</td>
</tr>
<tr>
<td>C1_SV</td>
<td>17.6</td>
<td>30</td>
<td>1.12</td>
<td>1.08</td>
</tr>
<tr>
<td>C2_SV</td>
<td>17.6</td>
<td>40</td>
<td>1.08</td>
<td>1.06</td>
</tr>
<tr>
<td>C3_SV</td>
<td>17.6</td>
<td>50</td>
<td>1.08</td>
<td>1.07</td>
</tr>
<tr>
<td>C4_SV</td>
<td>17.6</td>
<td>60</td>
<td>1.00</td>
<td>1.01</td>
</tr>
<tr>
<td>C5_SV</td>
<td>17.6</td>
<td>70</td>
<td>1.00</td>
<td>1.02</td>
</tr>
<tr>
<td>C6_SV</td>
<td>21.0</td>
<td>30</td>
<td>0.99</td>
<td>1.05</td>
</tr>
<tr>
<td>C7_SV</td>
<td>21.0</td>
<td>40</td>
<td>1.00</td>
<td>1.02</td>
</tr>
<tr>
<td>C8_SV</td>
<td>21.0</td>
<td>50</td>
<td>1.03</td>
<td>1.05</td>
</tr>
<tr>
<td>C9_SV</td>
<td>21.0</td>
<td>60</td>
<td>1.06</td>
<td>1.05</td>
</tr>
<tr>
<td>C10_SV</td>
<td>21.0</td>
<td>70</td>
<td>1.07</td>
<td>1.05</td>
</tr>
<tr>
<td>C11_SV</td>
<td>25.3</td>
<td>30</td>
<td>1.01</td>
<td>1.03</td>
</tr>
<tr>
<td>C12_SV</td>
<td>25.3</td>
<td>40</td>
<td>0.99</td>
<td>1.03</td>
</tr>
<tr>
<td>C13_SV</td>
<td>25.3</td>
<td>50</td>
<td>0.99</td>
<td>1.05</td>
</tr>
<tr>
<td>C14_SV</td>
<td>25.3</td>
<td>60</td>
<td>0.98</td>
<td>1.05</td>
</tr>
<tr>
<td>C15_SV</td>
<td>25.3</td>
<td>70</td>
<td>0.99</td>
<td>1.06</td>
</tr>
</tbody>
</table>

Under the supply air temperature of 25.3 °C, most positions in the room indicated lower local air change index less than unity. When the system was at 21.0 °C and 25.3 °C setpoints, the performance of the system was slightly below that of a MV system as evidenced by lower air change efficiency values reported in Table 6. Based on the results of the local air change index and air change efficiency, the ventilation system under study is incapable of satisfactorily provide heating in the office environment. In each case, the average exhaust air temperature
was about 24.3 °C, which further indicates that much of the heat added to the supply air, was lost through the exhaust. This result is in line with previous research results into SV system [26]. From previous studies by other researchers, the ventilation effectiveness of the SV system obtained was between 1.42 and 1.5 [25]. For design purposes the ventilation efficiency of 1.4 is recommended [26]. The highest local air change index and air change efficiency values obtained for the system were 1.15 and 53%, respectively. These results correlated with the first low air flow rates i.e., from 30 l/s to 50 l/s. SV system has been associated with better performance at low supply flow rates [81]. The results obtained closely relate to those obtained for the corner mixing ventilation system in previous laboratory study [72]. This can be attributed to the location of the air inlet devices which were in the corners and far from the occupants; the air reached the occupants after mixing with the room air. The location of the air inlet terminals relative to the occupants has been seen to have significant effect on the mean age of air and contaminant removal performance of the SV system [28]. For better performance of the SV system, short distance between the occupant and the air supply terminal must be maintained [82]. The relative distance between the air inlet terminal and occupants should enable fresh air inflow to be supplied directly to the occupants breathing zone level to achieve better performance of the system.

Sources of uncertainty in the obtained result pertain to the software used in the analysis of the results, improper mixing and leakage of tracer gas, incorrect positioning of the thermistors for velocity and temperature measurements in the room. Incorrect settings of the room air temperature could also have contributed to the errors.

Owing to the performance of the system as a MV system, the energy performance of the system is expected to be inferior to that of SV system.
6 Conclusions

6.1 Study results

The effect of the supply air temperature and supply air flow rates on the performance of the corner stratum ventilation system in an office was studied. The following main conclusions were made:

- Using the nominal supply air temperature of 17.6 °C caused the system to behave like a MV system despite the typical SV discharge height.

- Even though the supply air temperature was increased to 21.0 °C, the performance of the system deteriorated in performance to below the MV system as evidenced by low air change efficiency values. But the system retained the characteristics of the MV system in terms of the local air change indices. The poor performance was attributed to smaller temperature difference between room air temperature and supply air temperature since only the supply temperature was increased.

- Based on the results of the air change efficiency at supply air temperature of 25.3 °C, the system was found to be incapable of satisfactorily provide heating. This finding was also in line with other previous studies on the SV system.

- The draught rate levels decreased with increase in supply air temperature. The system satisfied the requirements of Category B of ISO7730 on thermal environments at all supply air temperature and air flow rate setpoints.

- In the occupied zone the room air velocity reduced with increase in height. The room air temperature increased with increase in height.

- More temperature stratification was observed for heights greater than 1.1 m and was much pronounced at lower air flow rates for all the nominal supply air temperature set points.

- The system satisfied the requirements of the Category A of ISO7730 on thermal environments in terms of PD at all supply air temperature and air flow rate setpoints.

- The performance of the system was better at lower supply air flow rates. Higher supply air flow rates led to poor performance of the system.
• The placement of the air inlet terminals had significant effect on the performance of the system.
• The SV system does not work efficiently when the air streams are heavily mixed before reaching the occupants. For better performance of the system, the fresh air inflow must be supplied directly to the occupants breathing zone level.

6.2 Outlook

SV system is a promising ventilation system that can provide better IAQ cost effectively. Further work in SV system is needed to determine the optimal location of air terminals in relation to occupants, the system’s performance in large spaces and its suitability for heating applications.

6.3 Perspectives

The study was aimed at examining the performance of the SV system with air inlet terminals placed in the corners of an office environment. The study sought to contribute to global efforts in research about better technology of providing enhanced air quality at low energy use in indoor environments. The development of ventilation systems with high ventilation effectiveness can lead to reduced energy use in buildings. This in turn reduces the primary energy demand, thereby decreasing the dependence on fossil fuel-related energy sources. Consequently, greenhouse gas emissions will be curtailed, and renewable energy sources promoted to meet the resulting energy demand. Furthermore, enhancing building energy efficiency through the optimal operation of HVAC systems can greatly contribute to the attainment of sustainable development goals associated with the clean and adequate energy supply. The strides made towards the establishment of reliable, affordable and sustainable energy systems for the global population can accelerate the achievement of other sustainable development goals.
References


[17] G. Cao et al., “A review of the performance of different ventilation and airflow distribution...


Appendices

Appendix A1: Measured air velocity at nominal supply temperature of 17.6°C
Appendix A2: Measured air velocity at nominal supply temperature of 21.0°C
Appendix A3: Measured air velocity at nominal supply temperature of 25.3°C
Appendix B1: Measured room air temperature at nominal supply temperature of 17.6°C
Appendix B2: Measured room air temperature at nominal supply temperature of 21.0°C
Appendix B3: Measured room air temperature at nominal supply temperature of 25.3°C
Appendix C1: Dimensionless air temperature at nominal supply temperature of 17.6 °C
Appendix C2: Dimensionless air temperature at nominal supply temperature of 21.0 °C
Appendix D1: Draught rate at nominal supply temperature of 17.6 °C
Appendix D2: Draught rate at nominal supply temperature of 21.0 °C
Appendix D3: Draught rate at nominal supply temperature of 25.3°C