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The use of displacement and zoning ventilation in a multipurpose arena

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Abstract

This study investigated the performance of ventilation and indoor climate in a multipurpose arena located in Malmö, Sweden with a seating capacity of 13 000 individuals, and in which a combination of displacement and zoning ventilation was applied. The main objective was to explore thermal conditions, indoor air quality, airflow patterns and air distribution. The measured operating conditions were ice hockey game and training situation. The measurements were conducted to observe the indoor climate and CFD-simulations were performed to get a generic view of the air distribution and the flow field over the whole arena indoor environment. The results show that air movement was highly case-dependent. However, the experiments indicated an upward flow along the lower-seating area and a downward flow along the upper-seating area. During the game, the average rise of temperature was around 2°C in arena with low stratification while using displacement ventilation. The temperature range was 12-17°C at the lower-seating area and 15-17°C at the upper-seating area. The corresponding air speed level was 0.06-0.36 m/s. The relative humidity was about 30-40% where a part of the humidity was transferring into the ice sheet. The carbon dioxide concentration increased locally to near 900 ppm indicating reasonable operation of ventilation. The CFD-simulations predicted well-mixed conditions in arena, thus supporting the measured low temperature stratification. Overall, the air movement was significantly affected by supply air temperature, variable air flow rates and retractable stand position (on-off) in the arena enclosure. The results support the use of displacement and zoning ventilation in multipurpose arenas.

Key words: indoor climate, air distribution, displacement ventilation, zoning ventilation, measurement, CFD

1. Introduction

In a multipurpose arena, a wide variety of activities can be organized for the individuals, e.g. ice hockey games, concerts, exhibitions and other public events. The multipurpose arena is classified as a large enclosure, where the specific feature is typically a small occupied zone compared to the total air volume, enabling local control of indoor conditions (Heiselberg et al., 1998; Liddament, 1998; Müller et al., 2013). The large enclosure can contain higher temperature stratification and a larger variety of air jets and sources compared to smaller enclosures. However, the main difference is a wider range of scales in flow patterns, thermal plumes and turbulent structures than in the smaller enclosures. This can yield a complex air movement and flow interaction scenario which is affected by convection, diffusion, pressure and buoyancy forces.

Generally, the air movement can be closely associated with thermal comfort, diluting of airborne particles and outdoor air distribution into the breathing level. Consequently, the air distribution and principal air movement are important in large enclosures (Liddament, 1998). The affecting parameters describing the flow are typically the cooling or heating mode, supply air flow rate and supply air temperature, as well as the temperature difference between the exhaust and supply air, and the location and direction of supply air openings (Nielsen, 2011). Furthermore, the airflow rate and temperature should be varied to satisfy the thermal comfort requirements in different zones (Zhai et al., 2002). In addition, the individuals can be in a continuous or an occasional motion that may have an effect on the flow patterns.

The air distribution is a flow process that can be characterized by different elements such as supply air jets, thermal plumes and boundary layer flows. The other substantial elements are the exhaust flows as well as the infiltration and gravity flows (Heiselberg and Nielsen, 1996). The airflow conditions may be characterized using dimensionless numbers, e.g. the Reynolds number and the Archimedes number. The Reynolds number is the ratio of inertial and viscous forces which can be expressed as

$$Re = \frac{ud}{v} \tag{1}$$

where u is the velocity, d is the characterized length and v is the kinematic viscosity, hence the fluid velocity and the fluid viscosity are closely related to turbulent flows. The Archimedes number, in turn, describes a ratio between the gravity and the momentum forces and can be expressed as

$$Ar = \frac{\beta g H (T - T_0)}{{u_0}^2}$$
(2)

where β is the volume expansion coefficient, g is the gravitational acceleration, H is the characteristic length, T is the temperature e.g. the exhaust air temperature, T_0 is the supply air temperature and u_0 is the average supply air velocity.

In large enclosures, a wide variety of Reynolds numbers and Archimedes numbers can occur depending on the flow condition and characterized length scale referring a certain flow phenomenon. This means that turbulent flows may dominate in the high Reynolds number flows and free convection may dominate in the high Archimedes number flows. However, the indoor airflows are usually turbulent (Müller et al., 2013), and the turbulence intensity is typically higher in the low velocity zones than in the high velocity zones. Furthermore, the significant mixing layers can occur near the supply air jets and thermal plumes. In any case, the gravity and free convection forces can have a remarkable effect in the large enclosures (Nielsen, 1993), and these issues have to be taken account in ventilation design.

The air distribution strategies can be classified based on the flow conditions in an enclosure. Hagström et al. (2000) proposed the classification for indoor air conditioning strategies. The first alternative is a piston ventilation strategy that can be performed using a low-momentum and unidirectional air distribution. Consequently, the piston ventilation produces a unidirectional airflow field and may provide the highest ventilation effectiveness. The second alternative is a stratification ventilation strategy that can be carried out using a lowmomentum air distribution method, i.e. displacement ventilation method. The stratification ventilation strategy produces mainly a buoyancy-forced flow field and thus density differences cause the flow patterns. The third alternative is a zoning ventilation

strategy that can be performed by controlling only the selected zones in an enclosure, e.g. the occupied zones. The zoning ventilation strategy produces both the momentum and buoyancy forced flow field, and the performance is highly dependent on the methods and operating conditions. The fourth choice is a mixing ventilation strategy that can provide nearly uniform flow conditions by using the highmomentum air distribution method. The authors concluded that several strategies should be available for different operating conditions. This is especially necessary in large enclosures because the geometry and occupied zones may vary and thus using only one strategy can be challenging. Therefore the combination of mixing ventilation and displacement ventilation have been common.

The multipurpose arenas are intermittently fully occupied, and may contain various heat and contaminant sources such as audience, lights and screens. Therefore, managing indoor climate plays an essential role in multipurpose arenas (IIHF, 2011). The ASHRAE Standard 62.1-2013 (ANSI/ASHRAE, 2013) recommends a minimum ventilation rate of 3.8 l/s per person in the seating area and 10 l/s per person in the play area.

The seating area is large and inclined which has to be taken account in the ventilation design. This means that free and forced convection as well as gravitational forces may cause an air movement along the inclined seating sectors that can have an effect on the ventilation performance and indoor climate. Consequently, the suitable air conditioning strategies have been usually the mixing ventilation, zoning ventilation or displacement ventilation methods. However, the inclined occupied zones and ice rink restrict the selection of the air distribution systems. Hence, a common strategy is to supply outdoor air directly into the occupied zones.

A combination of zoning and displacement ventilation was performed with high-momentum jets and with low-momentum jets, and by letting the supply-air to flow naturally along the occupied zones. Displacement ventilation can be a good choice particularly for the fully occupied conditions, e.g. when the thermal flows control the air movement (Nielsen, 2011). In multipurpose arena environments, displacement ventilation can offer both reasonable thermal conditions and a capable heating alternative. This is because warm air rises up high to the ceiling zone regardless of the air distribution method. Furthermore, ventilation strategy can be improved by using hybrid ventilation, e.g. a combination of mixing and displacement ventilation (Cao et al., 2014).

Many studies have been conducted with smaller ice arena enclosures (Yang et al., 2001; Bellache et al., 2005; Sunyé et al., 2007; Omri and Galanis, 2010). However, air movement can be different in large arenas due to a wider variety of length and time scales in flow patterns. Also the greater ratio of seating area to ice rink area, as well as the higher ceiling altitude, may have an effect on the flow field.

CFD simulation has been demonstrated to be capable of predicting fluid mechanics in an indoor environment (Chen and Srebric, 2002; Nielsen et al., 2007; Chen et al., 2010; Li and Nielsen, 2011; Nielsen, 2015). In ice arena environments, Yang et al. (2001) validated the CFD-model predicting the air temperature, air velocity and contaminant concentration distributions. Subsequently, Bellache et al. (2005; 2005) and Sunyé et al. (2007) developed indoor ice rink modelling methods. They presented methods for calculating the heat fluxes into the ice pad and recognizing the effect of ceiling emissivity on the thermal radiation. In a later study, Omri and Galanis (2010) investigated mixed convection and reported remarkable air velocities above the spectator stands and near the ceiling zone. Recently, Omri et al. (2015) showed that CFD-simulation can be a powerful tool in providing a detailed description of flow field and heat flow into the ice. In addition, Ferrantelli et al. (2012) and Ferrantelli and Viljanen (2015) investigated energy optimization and heat transfer in ice hockey halls.

Another common research topic has been the exposure to CO and NO₂ pollution produced from combustion engines in ice arenas which has contributed to introduction of electric ice resurfacing machines (Pennanen, 2005). In addition, Rundell (2003) concluded that high levels of airborne particles provides poor air quality and can be related to airway dysfunction in the ice rink athletes. Rosenlund et al. (2004) proposed that NO_2 exposure may be associated with airway symptoms even several years after the exposure and Kahan et al. (2007) reported significant symptoms after only a single exposure among the ice hockey players. Furthermore, Salonen et al. (2008) stated that insufficiently ventilated ice arenas may produce a risk of breathing poor quality air and therefore a risk of acute health effects can be substantial. The authors recommended worldwide governmental regulations, electric resurfacers and reasonable air exchange rate of 0.25-0.5 1/h.

The main motivation in the present study is that the indoor climate, air distribution and indoor air movement are not well-understood in large multipurpose arenas due to varying conditions involving slow and rapid changes in time and space with a wide range of scales. The main objective was investigate ventilation performance using to experimental methods and to solve how CFDsimulation can improve the understanding of the comprehensive arena indoor climate, where heat and mass transfer occur due to conduction, advection, diffusion and pressure gradients as well as due to thermal radiation and condensation or deposition. This complex flow phenomenon also involve free and forced convection with turbulent motion including velocity fluctuations, vortical structures and pressure differences. The flow field can then be characterized not just by air movement regions with air jets but also by flow interaction regions with convection flows. The novelty of this study comes from the measurements in challenging indoor environment and from the CFD-simulations that deepen the knowledge of probable trends under prevailing boundary conditions. Generally, this means limited but improved knowledge of the flow patterns and thermal conditions using a combination of displacement and zoning ventilation strategies that may complement the earlier studies.

2. Methods

2.1 Multipurpose arena

The multipurpose arena was located in Malmö, Sweden with a seating capacity of up to 13 000 individuals. The arena dimensions were approximately 100 m (L) x 90 m (W) x 30 m (H) and the interior architecture was comprised of the ice rink, seating area, cabinet region and building envelope with the side spaces, to name a few (Fig. 1a). The light structures and service catwalks were located at the height of 23 m above the ice (Fig. 1bc).

Displacement ventilation was employed for the lower-seating area and zoning ventilation was employed for the upper-seating area. The supply air flow of displacement ventilation was discharged through drilled holes from below the seats into the lower-seating occupied zone. The overall ventilation system consisted of four air-handling units (301-304) and two air-recirculating units (305-306) that were operating during the events (Fig. 2). The varying air flow was designed to operate at rates up to 70 m³/s,

thus fulfilling the standard regulations. The air flow rates were controlled based on air temperature and CO_2 concentration in an enclosure. This means that the recirculation units 305-306 were adjusted to outdoor air handling mode when the temperature or CO_2 content was increased above a certain threshold value. The target ranges for arena temperature and relative humidity were 14-21°C and 20-65% during the events, respectively. The target maximum level for the CO_2 concentration was 900-1200 ppm.

The supply air of displacement ventilation was distributed from under retractable stands beside the ice rink (Fig. 3a-b). The zoning ventilation supply air was distributed from the wall beyond the upperseating area (Fig. 3c). The extraction air was taken from the ceiling zone from both ends of the arena (Fig. 2b). This air was partly returned to the arena enclosure. Zoning ventilation was used because the spatial arrangement did not allow the use of displacement ventilation. values were 16°C and 35% on average. The average ice-sheet temperature was -7.2°C in the building management system (BMS). According to design data, the 4000 individuals generated 85 W of sensible heat, 20 l/h of carbon dioxide and 40 g/h of water vapour per person. The corresponding values for the 50 players were 200 W, 350 l/h and 175 g/h per person, respectively. Furthermore, the lighting load was 280 kW and the scoreboard produced a heat load of 30 kW whereas the infiltration rate was 0.05 l/h. The initial values are summarized in Table 1.

2.2 Measurements

The measurements were carried out for the thermal conditions, ventilation performance, air flow patterns and indoor air quality. The measurements were conducted with air velocity, turbulence intensity, temperature, CO_2 and humidity sensors as well as with infrared thermography. In addition, the air flow patterns were visualized with smoke. The measured regions were ice rink, seating area and middle of the

Sources	Heat gain		Carbon-dioxide		Water vapor	
Gain sources	kW	%	1/s	%	g/s	%
Audience	340	51.5	22.2	82.1	44.4	94.8
Players	10	1.5	4.9	17.9	2.4	5.2
Lights	280	42.4	-	-	-	-
Scoreboard	30	4.5	-	-	-	-
Total gain sources	660	100	27.1	100	46.9	100
Infiltration	-46.1	97.6	-1.3	100	0.5	-
Ice	-1.1	2.4	-	-	-0.4	-
Total sink sources	-47.2	100	-1.3	100	0.03	-
Total gain to cover with air-conditioning	612.8	-	25.7	-	46.9	-

Table 1. The heat, carbon-dioxide and humidity sources.

The ice hockey game was a midseason game that was played on 2.12.2009 between 19:00 and 21:30. During the game, the outdoor temperature was 2°C and the relative humidity was 95% at the Malmö weather station, whereas the corresponding indoor

air volume at different altitudes. The sensors were installed onto measuring masts at the heights of 1.5 m, 2.5 m and 3.5 m over the seating-row and the variables were recorded over a 3 min average. The masts were fixed to the back of the seats (Fig. 4a-b).

The measured sector and the measuring-masts (locations 1-9) as well as the temperature-probewires (locations A-B) are shown in Fig. 4c. The lowest locations 1-3 were on the retractable stands and the measuring-masts 4-6 were on the fixed area, whereas the measuring-masts 7-9 were on the upperseating area.

The flow patterns were investigated using measurements and smoke visualization experiments. The smoke was released into the ventilation ducts and also directly into the occupied zones and air jets. High intensity lights and a video camera were used to observe the flow patterns. The air speed, temperature and turbulence intensity were measured with hotsphere anemometers. The middle of the air volume was measured gradually from the service catwalks down to the occupied zones using a hot-sphere anemometer and a long cable wire. The arena temperature stratification was measured with chains of thermistor sensors. The operative temperature was observed with black-globe thermometers and the envelope temperature levels were investigated using infrared thermography. In the ice rink, the ice-surface temperature and the air temperature stratification were explored in an empty arena. The measuring equipment was moved with a carriage between the measured locations. Table 2 shows a summary of the measurements.

2.3 CFD-simulations

CFD-simulations were used to investigate the flow field in the arena enclosure. The CFD process was guided by Nielsen et al. (2007) and the validation process was applied according to Chen and Srebric (2002). The Reynolds-averaged Navier-Stokes equations (RANS) were conducted with ANSYS CFX software (ANSYS Inc., 2009). The CFD geometry and computational grid was created with ANSYS ICEM CFD software (ANSYS Inc., 2009) based on architectural plans and a building information model (BIM). The predicted velocity magnitude and the turbulence intensity were modified to correspond to the omnidirectional measurements (Koskela et al., 2001). Micro and macro CFD models (Chen and Srebric, 2002) were applied to describe the arena indoor climate and the box-method was implemented to describe the air terminal units (Nielsen, 1991; Srebric and Chen, 2002; Nielsen (ed.) et al., 2007).

The different air terminal units were modelled individually and the step-wise produced flow profiles were used as a boundary condition in the whole arena CFD simulation model. The CFD was performed using the CFX finite-element-based control volume method and the implicit pressure-based multigrid coupled solver. The high resolution discretization scheme was conducted with the SST-turbulence model (Menter, 1994) and the automatic wall treatment formulation. Thermal radiation was computed using the discrete transfer method.

The CFD simulations were based on the prevailing conditions at 21:00 when the 3rd period was ongoing, that is near the end of the game. The seating facility and the arena envelope boundary conditions were based on the average temperatures from the infrared thermography. Audience sources were applied at the seating-area surfaces to generate heat flux, carbondioxide and humidity. The players were described as a volumetric source above the ice sheet. The boundaries of the air terminal units were based on the measurements, design data and laboratory experiments in which the temperature and velocity or pressure were estimated together with turbulence quantities. The air model was a multicomponent ideal gas model with water-vapor and CO₂ components.

An unstructured tetrahedron grid was generated with 3-10 prismatic element layers at the wall surfaces with certain expansion factor. The grid sizes were between 0.1-0.6 m in the whole arena model and around 0.5 mm in the nozzle model (d_{nozzle} ~50mm). In the arena model, the total number of grid nodes was 19.68x10⁶ with 102.78x10⁶ elements describing the ice hockey game conditions in discrete points. The larger grid elements were located in the middle of the air volume, because the gradients were smaller than near the occupied zones.

3. Results

In the ice hockey game, the building management system (BMS) displayed a total airflow rate of near 30 m^3 /s at the beginning of the game that was later increased to 68 m^3 /s when the recirculating units 305-306 were introduced. (Fig. 5). Furthermore, the recirculation units were adjusted to supply outdoor air flow if necessary. In the BMS-system, the air temperature inside the arena was on average 16.6-17.6°C and the supply air temperature was 14.8-16.4°C. The exhaust air temperature was lower than the arena temperature, i.e. 15.9-17.0°C, because the

arena sensors were located below the cabinet region under slightly warmer conditions. The temporary rise of temperature was on average by only 1.8°C at the

3.1 The ice rink conditions

The ice rink conditions were measured at 11 locations

Variable	Meter-type	Region	Model		
Temperature Air speed Turbulence intensity	Hot-sphere anemometer and analyzer	Arena air volume Seating area	Dantec hot-sphere anemometer Dantec 54N10 analyzer		
Temperature stratification	Chains of thermistor sensors	Air volume above the seating area and above the ice rink	Term Elfa 10K sensor Agilent 34970A data logger		
Operative temperature	Black-globe thermometer	Seating area	Tinytag Ultra 2 datalogger inside a black sphere		
Surface temperature	Infrared thermography	Envelope structures	Fluke Ti55 IR FlexCam thermal imager		
Thermal radiation intensity	intensity Indoor climate analyzer		Brüel & kjær type 1213		
Surface temperature	Thermocouple- thermometer	Ice sheet	Delta Ohm HD 2328.0		
Temperature stratification	Radiation shielded sensor	Ice rink	Craftemp-thermistor Agilent 34970A data logger		
Relative humidity	Humidity indicator	Seating area Ice rink	Tinytag Ultra datalogger Vaisala HMI 11 Vaisala HMP35E		
Carbon-dioxide	CO2-meter	Seating area Service catwalks	Sense Air CO2-meter Sense Air aSense-D		
Airflow patterns	Smoke visualization Hot-sphere anemometer Videorecording	Seating area Air terminal units	Smoke generator Martin Magnum 1500 Dantec hot-sphere anemometer Videocamera Canon DC-40		
Sound pressure	Sound level meter	Seating area	Cesva SC-15c		
Illuminance	Photo radiometer	Ice rink	Delta Ohm HD 9221		

Table 2. Indoor climate measurements.

location A and 1.7°C at the location B during the game at 19-21:30 (Fig. 6), and also the temperature stratification was low because the temperature difference was below 1°C compared to upper regions. The temperatures of the envelope surfaces were slightly increased during the game, and the audience warmed up the seats and structures (Fig. 7). Cooler surfaces were observed with the IR camera not only near the ice rink but also near the entrances whereas warmer surfaces were found at the audience, cabinets and screens when an emissivity of 0.95 and a background temperature of 16°C were assumed.

in the empty arena. The average ice surface temperature was -4.0° C, the measured thermal radiation intensity was 68 W/m² and the illuminance level was 260 lx. A polynomial curve fit was generated for the air temperature profile above the ice (Fig. 8) in which the squared correlation coefficient was R2=0.998. The standard deviation in the measured temperature profile was within 0.33°C. The curve fit produces a function from which the temperature and the temperature gradient can be obtained at different heights. The curve fit was expressed as

$$T = -2.917y^4 + 16.29y^3 - 31.89y^2 + 29.00y - 3.786$$
(3)
$$0 < y \le 2$$

where T [°C]is the temperature and y [m] is the height above the ice surface. The air temperature was -4...10°C below the height of 2 m and the gradient was increasing towards the ice surface, i.e. dT/dy(1)= 2.4°C/m, dT/dy(0.5) = 7.9°C/m and dT/dy (0.1) = 23°C/m. In contrast, the linear temperature gradient on the seating sector was mainly below 1°C/m. The air speed was low in an empty ice rink and the relative humidity was 45-60% while the outdoor temperature was -1...2°C and the outdoor relative humidity was 87% at the weather station. The standard deviation in the measured relative humidity profile was within 1.66%.

3.2 Air movement

In the upper-seating area, the smoke tests (Fig. 9) indicated a downward flow along the occupied zone from where the air progressed into the middle of the air volume. In the lower-seating area, the smoke tests indicated an upward flow along the occupied zone (Fig. 10). It seems that the primary air movement was generally from the ice rink to the measured seating sector. However, when the airflow rate was decreased, the upward air movement decayed and turned towards the end-side of the arena, although the supply air temperature was slightly increased. Hence, the primary air movement was strongly influenced by the ventilation control strategy and the prevailing conditions.

Two major air-circulation regions were drawn for a flow idealization (Fig. 11), above the lower-seating area and above the upper-seating area based on observations and predictions. Those rotation flows may grow up from the convection flows due to supply air jets, heat sources and cold surfaces. The airflow in the occupied zone was turbulent and the flow eddies can be mainly from the order of the airflow size down to the Kolmogorov microscales (Pope, 2000). However, much larger flow patterns may exist in the middle and in the air circulating regions of the arena enclosure during the events.

3.3 Seating area

The air temperature, air speed, turbulence intensity, carbon-dioxide and relative humidity were observed using measuring masts. In the beginning of the game, CO_2 content varied between 484-701 ppm at the measuring locations (Fig. 12). During the game, the

 CO_2 content changed between 155-246 ppm resulting in a maximum of 889 ppm at the location 6 on the upper part of the lower-seating area. The minimum level was 600 ppm at the location 3 on the retractable stands near the supply air inlets. Hence, it seems that CO_2 concentration increased towards the upperseating area. The outdoor concentration was around 400 ppm and the supply-air concentration was 514-582 ppm, including the recirculated air. The concentration at the service-catwalks varied between 582-745 ppm. The observed relative humidity was 31-41% and the change varied between 1.7-3.9% at the measuring locations (Fig. 12).

The air temperature range was increased averagely from $12.3-15.1^{\circ}$ C to $14.4-16.9^{\circ}$ C in the lowerseating area at the locations of 1-6 during the game (Fig. 13). In the upper-seating area, the temperature change was from $15.1-15.9^{\circ}$ C to $16.5-17.1^{\circ}$ C at the locations of 7-9. Hence, the average growth of air temperature was 2.4° C in the lower-seating area and 1.3° C in the upper-seating area. The corresponding air speed change was from 0.07-0.36 m/s to 0.06-0.19 m/s in the lower-seating area (Fig. 14) and from 0.07-0.18 m/s to 0.14-0.25 m/s in the upper-seating area and increased -0.07 m/s in the lower-seating area and increased 0.08 m/s in the upper-seating area during the game.

In CFD-simulations, a multi-model approach was developed to describe the enclosed virtual domains for the air terminal units. In the multi-model approach, the air terminal unit is modelled stepwise from the micro-nozzle-model (Fig. 15a) to the macrodiffuser-model (Fig. 15d) by transferring the solved flow profile between the models for generating the boundary condition for the whole arena CFD-model. In the beginning, the solved micro flow profile (Fig. 15a) is transferred to the next-level model (Fig. 15b) as a boundary condition that corresponds to the given flow profile. Then, the flow profile is solved in this 2nd level model (Fig. 15b) and the given flow profile is transferred to the 3rd level model as a boundary condition (Fig. 15c). This process is continued stepwise unless the boundary condition for the whole arena CFD-model is created (Fig. 15d). The multimodel approach will keep the range of scales more reasonable in space and time, but produces an error source because the computational grid is usually different in different models, thus interpolation is necessary between those sequential models.

The CFD-simulations for the prevailing conditions at 21:00 indicated well-mixed indoor climate in the

arena (Fig. 16). The results show a small temperature stratification above the retractable stand, and the predictions follow quite closely the corresponding measured profile. Also, the air speed was low in the middle of the air volume. The measured temperature was $15.8-16.3^{\circ}$ C, but the BMS-system indicated a higher average temperature of 17.6° C because the BMS-sensors were placed below the cabinet area at warmer conditions. The measured turbulence intensity was $30-50^{\circ}$. The highest intensity level was found above the retractable stands near the ice rink. Generally, the turbulence intensity was increased towards the smaller air speed levels (Lestinen et al., 2012).

When analysing the CFD-simulation results, the near-field investigation was conducted using 1 m offset values from the actual measuring locations in every coordinate direction for recognizing the changes in the surroundings. The near-field values offer the trend in which the predicted values occur close to the measured location, since the CFD model is an approximation under the prevailing boundary conditions. The near-field investigation revealed both higher temperature gradients and higher velocity gradients near the heat sources, e.g. the occupancy loads. Furthermore, the change in the flow field was generally getting smaller towards the upper arena regions. This reflects a common behaviour in this flow field; higher changes near the air jets and sources and smoother changes beyond those areas.

The measured results showed a temperature level of $15.9-16.5^{\circ}$ C at location 5 (Fig. 17), but the seat sensor indicated a lower temperature of 14.8° C. The range of difference between the predicted and the measured temperature was 0.50° C. Consequently, both the measurements and the predictions implied a rather small vertical gradient, except near the ice rink. The measured air speed level was 0.13-0.18 m/s at location 5. In predictions, the upward airflow was at the same level along the seating sector than in the measurements.

4. Discussion

Designing ventilation is challenging in a multipurpose arena. Various occupied events and their arrangements alternate with non-occupied training and maintenance periods, which is why a flexible ventilation strategy with variable air flow rates is necessary. The indoor climate can be stable during the maintenance and training period, but throughout the events, rapid and slow changes may occur when individuals are either in sudden or continuous motion in the arena environment. Consequently, the design challenges are certainly highlighted in the fully occupied events in which the key objectives are to distribute supply air evenly to the occupied zones and to produce an optimal indoor climate for assuring comfortable environment and an individual's well-being in an energy efficient manner.

In the arena, the measurement and modelling results showed that primary air movement was complicated, case-dependent and affected by many parameters. The air movement was mainly upwards along the lower-seating area and downwards along the upperseating area in the measured sectors. The flows collated above the cabinet level and progressed into the middle region. The major affecting parameters were the airflow rates, the on-off position of the retractable stands and the temperature difference between the supply airflow and surroundings. Furthermore, the relationship of the airflow rates in different sectors had an effect on the indoor air movement as well as the combination of forces caused by the pressure differences, heat sources, supplied airflow rates and cold surfaces.

In this arena, a combination of displacement ventilation and zoning ventilation was applied. Generally, the displacement ventilation strategy was suitable for the events where the retractable stands were in use. An alternative could be a wall-confluentjet adjustment (Cho et al., 2008) for the upper nozzleducts. This might produce more equal and gradual supply air flow into the occupied zones. Altogether, the indoor air quality was at a good level and the temperature stratification was low due to well-mixed conditions. Obviously, temperature stratification was not able to build up because the inclined seating area and flow movement were preventing it due to recirculation flow and effective mixing, and also the ice rink was cooling down the indoor climate. The air speed was at a reasonable level for a large enclosure containing ventilation jets and heat sources with a relatively high temperature range that may locally accelerate the airflows. The air distribution might be further improved by distributing supply air from below the seats at the whole seating area.

5. Conclusion

This study investigated the performance of ventilation and the indoor climate in a multipurpose arena located in Malmö, Sweden in which a

combination of displacement and zoning ventilation was applied. The main objective was to explore thermal conditions, indoor air quality, airflow patterns and air distribution by using a combined air distribution strategy. Furthermore, the ventilation performance was investigated with experimental methods and CFD-simulation, which improved the comprehensive understanding of the indoor climate of the arena. The measured operating conditions were an ice hockey game and a training event. Measurements were conducted for observing the indoor climate and CFD-simulations were performed to get a generic view of air distribution and the flow field over the whole arena enclosure. The results show that air movement was highly case-dependent and affected by many parameters such as supply air temperature, variable air flow rates and retractable stand position in the arena enclosure. The combined air conditioning strategy was well-operated and brought outdoor air into the occupied zones. The displacement ventilation strategy was particularly suitable for the events where the retractable stands were in use. During the ice hockey game, the average rise of temperature was about 2°C with low stratification, and the temperature level was 12-17°C at the lower-seating area and 15-17°C at the upperseating area. The corresponding air speed level was 0.06-0.36 m/s. The results support the use of displacement and zoning ventilation in multipurpose arenas.

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Figure 1. The multipurpose arena indoor environment: a) Ice rink (white), lower-seating area (red), upper-seating area (blue), cabinet area between the seating areas. b) a view from a service catwalks, retractable stands in off-position. c) light structures, field and scoreboard.



Figure 2. A scheme of the ventilation system in the arena enclosure: a) Air-handling units (301-304) and recirculating units (305-306), b) Air distribution strategy: Lower-part displacement ventilation and upper-part zoning ventilation, exhaust air from the ceiling zone.



Figure 3. Displacement and zoning ventilation: a) air distribution below the retractable stand, b) supply air flow from below the seats at the lower-seating area, c) zoning ventilation above the upper-seating area.



Figure 4. The measuring masts at the seating sector: a) a measuring mast and the sensors, b) the measuring masts fixed at the lower-seating sector, c) the location of the measuring masts (1-9) and the thermistor chains (A-B) above the seating area.



Figure 5. Air flow rates and temperature level during the game evening: a) supply air flow rate, b) the average temperature of the arena and the supply and exhaust air.



Figure 6. The temporal rise of air temperature in different altitudes, locations A and B.



Figure 7. Infrared thermography image at the beginning and at the end period of the game.



Figure 8. The ice rink conditions: a) average temperature stratification, b) relative humidity with standard deviation.



Figure 9. Air movement along the upper-seating sector and from there to the middle of the air volume.



Figure 10. Air movement along the lower-seating sector.



Figure 11. An idealization of the indoor air movement in the arena enclosure.



Figure 12. The carbon dioxide concentration and relative humidity on the seating area.



Figure 13. The measured indoor air temperature at the heights of 1.5 m, 2.5 m and 3.5 m (loc. 1-6): a) before the game, b) in the third period.



Figure 14. The measured indoor air speed at the heights of 1.5 m, 2.5 m and 3.5 m (loc. 1-6): a) before the game, b) in the third period.



Figure 15. CFD-simulation with a multi-model approach – duct diffuser model: a) nozzle of the duct, b) nozzle ring of the duct, c) duct-diffuser model including combined rings, d) seating-sector model.



Figure 16. The temperature at the location B above the retractable stand. The colored area for the prediction on the right side diagram describes the range of actual location and near-field values (1 m offset).



Figure 17. The indoor air temperature and the air-speed at the location 5.