CLIMA 2016 - proceedings of the 12th REHVA World Congress

volume 5

Heiselberg, Per Kvols

Publication date:
2016

Document Version
Publisher's PDF, also known as Version of record

Air Distribution and Indoor Climate in a Multipurpose Arena

Sami Lestinen[#1], Hannu Koskela[*2], Simo Kilpeläinen[#3], Risto Kosonen[#4]

[#]*Department of Mechanical Engineering, Aalto University*
Sähkömiehentie 4, 02150 Espoo, Finland

[1]sami.lestinen@aalto.fi

[3]simo.kilpelainen@aalto.fi

[4]risto.kosonen@aalto.fi

[*]*Turku University of Applied Sciences,*
Lemminkäisenkatu 14-18 B, FI-20520 Turku, Finland

[2]hannu.koskela@turkuamk.fi

Abstract

This study investigated a multipurpose arena located in Malmö, Sweden with a seating capacity of up to 13 000 individuals, and in which a combination of displacement and mixing ventilation was applied. The dimensions of the arena were 100 m (L) x 90 m (W) x 30 m (H) and the interior architecture was comprised of ice rink, seating area, cabinet region and building envelope with structures. The objective was to investigate thermal conditions, indoor air quality, thermal comfort, airflow patterns and ventilation performance. The measured operating conditions were ice hockey game, training session and maintenance situation. The research methods were CFD-simulation and experimental measurements. The measurements were conducted with air speed, temperature and humidity sensors, and by taking thermal images and visualizing with smoke. CFD-simulations were performed to investigate the arena flow field. It was found that the primary air movement is complicated and case-dependent. The temperature rise was only 2ºC during the game and the temperature stratification was small. Furthermore, the temperature level was 13-17ºC at the lower-seating area and 15-17ºC at the upper-seating area. The corresponding air speeds were below 0.25 m/s and 0.3 m/s, respectively. The relative humidity was 30-45% and the carbon dioxide content did rise by up to 300 ppm during the game, increasing the total to nearly 900 ppm. The supply air temperature, variable air flow rates and retractable stands had a notable effect on the principal air movement. The air distribution could possibly be improved by implementing individual ventilation and heating strategies.

Keywords - multipurpose arena; indoor climate; measurement; CFD

1. Introduction

Managing indoor climate plays an essential role in multipurpose arenas, because the climate is a continuously changing environment that requires controlling strategies for individuals. Consequently, the airflows as well as heat and mass transfer are controlled using HVAC-systems and a ventilation strategy that will enable sufficient amount of outdoor air and optimized conditions. Generally, reasonable air flow rates are sufficient, because the occupied zone is small in large enclosures [1]. The ASHRAE Standard 62.1-2013 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 [2]. Since energy efficiency and health have become increasingly important during the recent years [3], a more efficient air distribution is necessary that can reduce the air flow rates and energy consumption without compromising the indoor climate. The ventilation efficiency is one of the dimensioning parameters that are stated in the design standards.

This study investigated a multipurpose arena located in Malmö, Sweden with a seating capacity of up to 13 000 individuals. In the arena a combination of lower-part displacement and upper-part mixing ventilation strategy was applied to manage the indoor climate. Consequently, the objective was to investigate thermal conditions, indoor air quality, thermal comfort, airflow patterns and ventilation performance. The dimensions of the arena were 100 m (L) x 90 m (W) x 30 m (H) and the interior architecture was comprised of ice rink, seating area, cabinet region and building envelope with surrounding corridors with restaurants. The light structures and service catwalks were located at the height of 23 m above the ice floor.

The ventilation system mainly contained four air-handling units (301-304) and two air-recirculating units (305-306) that were operating during the events (Fig. 1). The recirculating units were designed to also distribute outdoor air when necessary. Displacement ventilation air was supplied from under the retractable stands beside the ice rink, and mixing ventilation air was supplied from the wall beyond the upper-seating area (Fig. 2). The exhaust air was taken from the ceiling zone from both ends of the arena.

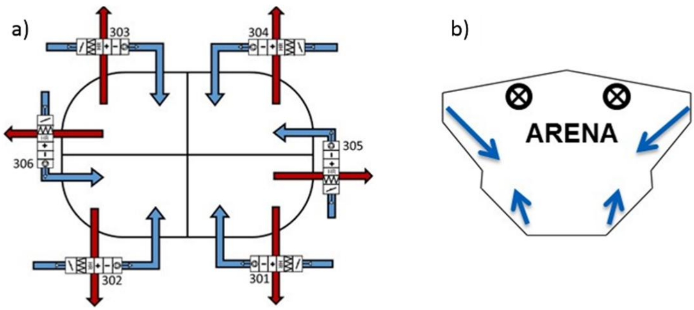

Fig. 1 A scheme of the ventilation system in the arena enclosure: a) Air-handling units (301-304) and recirculating units (305-306), b) Air distribution strategy.

The background of this study was that indoor climate and air distribution are not well-understood in large enclosures due to various conditions involving slow and rapid changes in time and space with a wide range of scales. These changes may grow up from the variation of pressure, temperature, humidity and air movement fluctuations due to advection, diffusion and heat flows occurring in this environment.

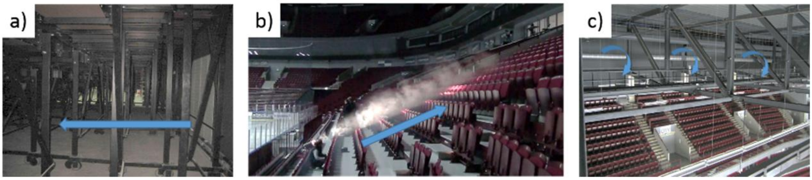

Fig. 2 Displacement and mixing ventilation: a) air distribution below the retractable stands, b) supply air flow from below the seatings at the lower-seating area, c) mixing ventilation above the upper-seating area.

Earlier research has frequently studied the smaller arena enclosures [5][6][10] whereas the present study has focused on an arena with a large air volume, a wide seating area and a high ceiling [7]. The previous studies have also concentrated on the energy optimization and heat transfer [8] as well as CFD-simulation [9]. One crucial topic has been the exposure to CO and NO2 concentrations [4] produced from combustion engines, which has contributed to the introduction of electric ice resurfacing machines.

2. Methods

The research methods were field measurements and CFD-simulations. The measured operating conditions were ice hockey game, training and maintenance period. The game was a middle season game and it was played between 19:00 and 21:30. The audience was around 4 000 individuals and additional heat gains were lights, screens and the scoreboard. The measurements were conducted with air speed, temperature and humidity sensors, and infrared thermography. The air distribution was visualized with smoke. The sensors were installed onto 3 m high measuring masts at the heights of 1.5 m, 2.5 m and 3.5 m over the seating-row (Fig. 3). The masts were fixed to the back of the seats and the variables were averaged over 3 min periods. Table 1 shows a summary of measurements.

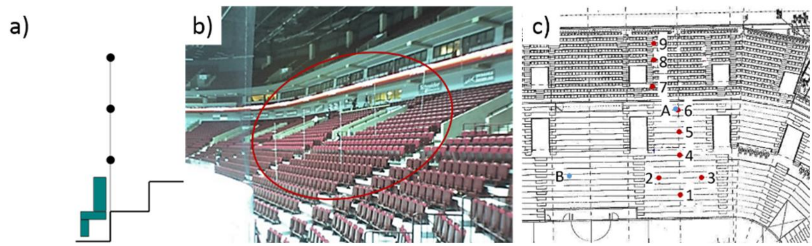

Fig. 3 The measuring masts: a) a measuring mast and the sensors, b) the masts fixed at the lower-seating sector, c) the location of masts (1-9) and termistor chains A and B.

Table 1. Indoor climate measurements.

Variable	Meter	Region
Temperature Air speed Turbulence intensity	Hot-sphere anemometers Dantec 54N10	Arena air volume Seating area
Temperature stratification	Chains of thermistor sensors	Above seating area Ice rink
Operative temperature	Black-globe thermometer	Seating area
Surface temperature	Infrared thermography	Envelope structures
Thermal radiation intensity	Brüel&Kjaer indoor climate analyzer 1213	Ice sheet
Surface temperature	Thermocouple-thermometer	Ice sheet
Temperature stratification	Radiation shielded Craftemp-thermistors	Ice rink
Relative humidity	Humidity indicators Tinytag loggers	Ice rink, Seating area
Carbon-dioxide	Sense Air sensors	Seating area, Service catwalks
Airflow patterns	Smoke, anemometers, videos	Air terminal units, Seating area

CFD-simulations were used to investigate the flow field in the arena enclosure. The CFD process was followed the guidelines by Nielsen et al. [11] and the validation was applied according to Chen and Srebric [12]. The predicted velocity magnitude and turbulence intensity values were modified to correspond to the omnidirectional measurements [13]. The CFD-geometry was created based on the architectural plans and the building information model (BIM). Both micro and macro models [12] were applied to describe the arena indoor climate and the box method [11] was implemented to describe the air terminal units. The RANS simulations were conducted with ANSYS CFX software [15], in which the Reynolds-averaged Navier-Stokes equations (RANS) were modelled using CFX finite-element-based control volume method and implicit pressure-based multigrid coupled solver. The high resolution discretization scheme was used with the SST-turbulence model [14] and the automatic wall treatment formulation. The thermal radiation was computed using the discrete transfer method.

The multi-model approach was developed to describe the supply air flow from the air terminal units (Fig. 4). In a multi-model approach, the solved flow profile is transferred from the small and detailed micro-model to the next-level model using a box-method principle. Consequently, the solved former-level flow profile (Fig. 4a) is implemented to the next-level model as a boundary condition (Fig. 4b) which describes the enclosed virtual domain that, in turn, corresponds to the given flow profile geometry. This step-wise process is continued until the flow profiles for the boundary conditions are generated to the arena macro-model. In this path, the range of different scales will remain reasonable if compared with the model that includes all the spatial scales from the tiny supply air nozzles to the large air-volume enclosure.

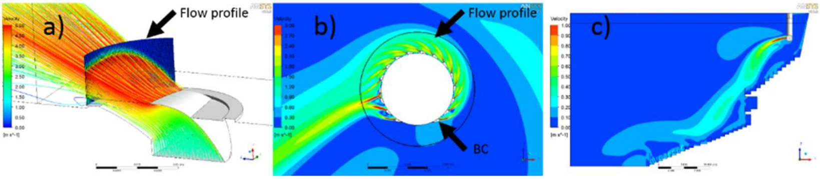

Fig. 4 CFD-simulation with a multi-model approach: a) a nozzle of the duct, b) a nozzle ring of the duct, c) seating sector model.

Furthermore, a wide range of geometry-scales may cause large changes in the computational grid resolution or alternatively very dense grid in the given flow field. In addition, the large changes in the grid can lead to higher instability and residual levels and therefore impair the iteration convergence. However, the disadvantage of this multi-model approach is generally a modeling error from the step-wise process, because the computational grid is non-similar and interpolation is necessary while transferring the data between the sequential models.

3. Results

The primary air movement was complicated and case-dependent and the major affecting parameters were the airflow rates, the retractable stand position and the temperature differences between the supply airflow and surroundings. Furthermore, the relationship of airflow rates in different sectors had a significant effect on the air movement. One factor was also a combination of forces caused by the pressure differences, heat sources, supplied airflow and cold surfaces. Generally, the displacement ventilation was suitable for the events where the retractable stands were in use. At this stage, the supplied air was discharged through the drilled holes from below the retractable stands to the occupied zone. The temperature gradient was below 1ºC/m, except near the ice rink. Mixing ventilation was introduced to the upper-seating area. The primary air movement was mainly upwards along the lower-seating area and downwards along the upper-seating area in the measured sectors as described in Fig. 1b. The flows confronted above the cabinet level and progressed towards the middle region. The measured airflow rate of the building management system (BMS) was near 30 m^3/s at the beginning of the game and it was increased up to 68 m^3/s based on air quality, i.e. CO2-sensors (Fig. 5).

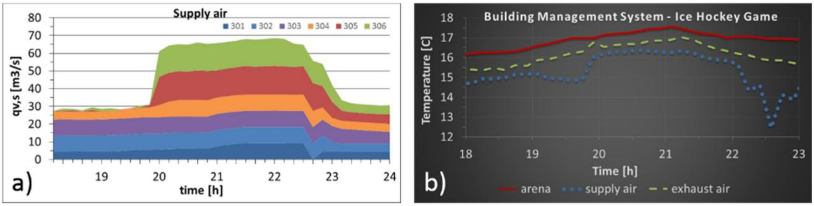

Fig. 5 Building management system (BMS) data during the game evening: a) supply air flow rate, b) the average temperature of the arena and the supply and exhaust air.

The arena sensors of the building management system were mainly located below the cabinet level, above the lower-seating area. The average arena temperature was 16-18°C, whereas the supply air temperature was 15-17°C. The exhaust air temperature was between those values. The average ice temperature was −4°C and thermal radiation intensity was 68 W/m^2. The polynomial curve fit for the air temperature above the ice (Fig. 6) with the squared correlation coefficient R^2=0.998 was expressed as

$$t = -2.917y^4 + 16.29y^3 - 31.89y^2 + 29.00y - 3.786 \;;\; 0 < y \leq 2, \qquad (1)$$

where t is the temperature and y is the height above ice. The air temperature was −4…10°C below the height of 2 m and the gradient dt/dy was increasing towards the ice surface. The temperature gradient was $t'(1) = 2.4$°C/m, $t'(0.5) = 7.9$°C/m and $t'(0.1) = 23$°C/m. In contrast, the linear temperature gradient on the seating sector of the occupied zone was mainly below 1°C/m. The air speed in the empty ice rink was low and the relative humidity was 45…60% while the outdoor temperature was −1…2ºC and the relative humidity 87% during the measurement period.

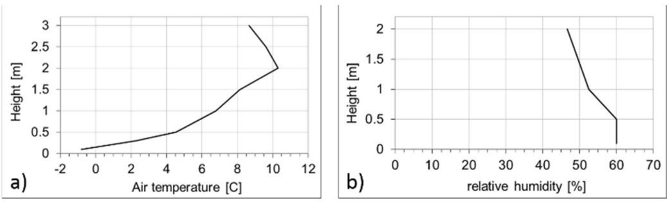

Fig. 6 The ice rink conditions: a) average temperature stratification, b) relative humidity.

The envelope wall temperatures increased slightly during the game and the audience warmed up the seats and the structures (Fig. 7). Cooler surfaces were observed not only near the ice rink but also near the entrances whereas warmer surfaces were found at the audience, cabinets and screens.

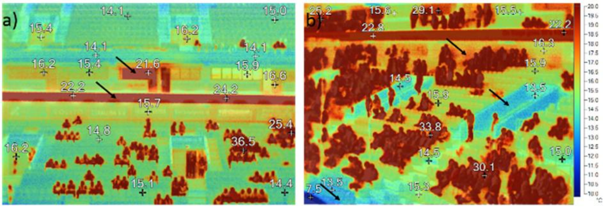

Fig. 7 The thermal images: a) before the game, b) third period.

The temperature rise during the game was only 2ºC when the displacement ventilation was introduced, and the temperature stratification was low. In the seating area, the relative humidity was 30-45% whereas the outdoor temperature was 2ºC and the relative humidity was 94-95% at the weather station. The carbon dioxide content had grown by up to 300 ppm increasing the total to nearly 900 ppm. The temperature was 13-17ºC at the lower-seating area and 15-17ºC at the upper-seating area. The corresponding air velocities were below 0.25 m/s and 0.3 m/s, respectively. The air temperature was lower on the rows near the ice rink (Fig. 8) and the temperature change was larger in the beginning of the game. The air speed level varied more in the beginning and smoothened later on. The fluctuation was higher near the ice rink at the lower-seating area (Fig. 9).

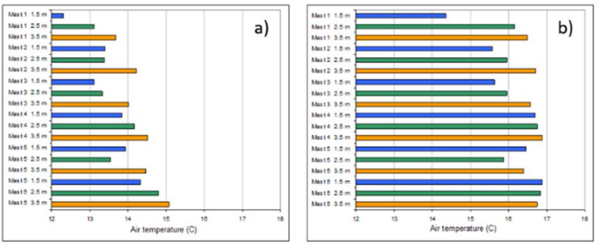

Fig. 8 The indoor air temperature at the masts 1-6 measured at the heights of 1.5 m, 2.5 m and 3.5 m: a) before the game, b) in the third period.

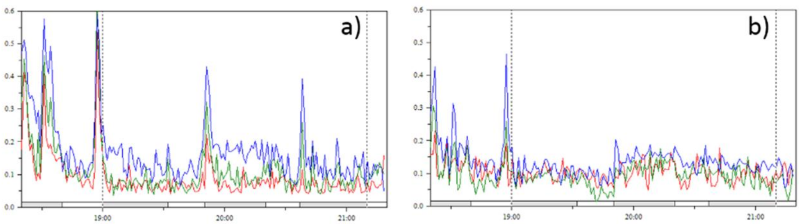

Fig. 9 Air speed at the lower-seating area: a) near the ice rink – mast 1, b) upper part of the retractable stand – mast 4.

Fig. 10 shows that temperature stratification was low in the arena enclosure. The predicted and the measured profiles were on the same level and the building management system (BMS) indicated higher temperatures, because the sensor location was different, below the cabinet level, where the temperature was generally higher.

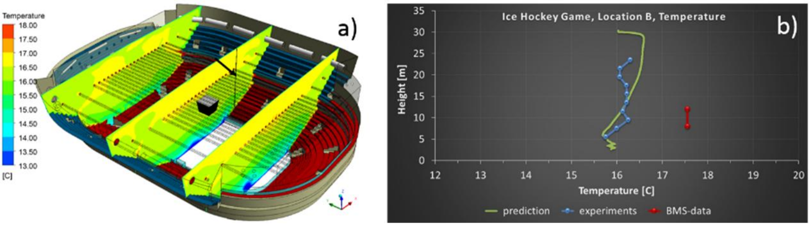

Fig. 10 A temperature distribution of the CFD-simulation result, 3rd period: a) color-scale plot of the predicted flow field, arrow shows location B. b) temperature distribution at location B.

The primary air movement was complicated. However, two major air-circulation regions were drawn for a flow idealization, above the lower-seating area and above the upper-seating area (Fig. 11). Those rotation flows can grow up from the convection flows due to supply air jets, heat sources, and cold surfaces as well as the openings to surrounding areas and outside. The outdoor air was distributed from the nozzle-ducts that were set to discharge supply air upwards from the upper-wall. The alternative choice would be to set the nozzle-ducts to the wall-confluent-jet adjustment [16]. In this path, the supply air is distributed gradually to the wall and downstream to the audience at the upper-seating area.

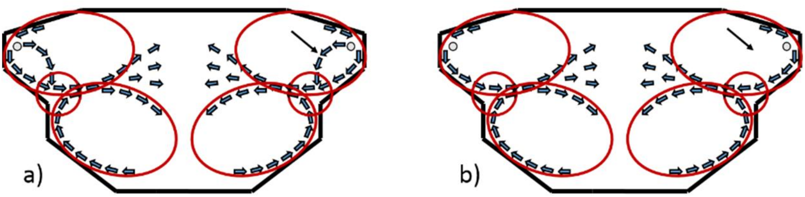

Fig. 11 An idealization of the indoor air flows in the arena enclosure: a) current ventilation strategy, b) wall-confluent-jet ventilation strategy

4. Discussion and Conclusion

The ventilation design of a multipurpose arena is challenging. The various entertainment events and their arrangements alternate with training sessions and maintenance periods which is why the flexible management systems of indoor climate are essential. The design challenges are highlighted during the fully occupied events and the key objectives are to distribute supply air evenly and to produce optimal indoor climate in an energy efficient manner, for assuring healthy environment and well-being.

In this arena, a combination of displacement and mixing ventilation was applied. The lower-part displacement ventilation distributed supply air evenly to the audience and it flowed upwards the lower-seating sector, when the retractable stands were in use. In the upper-seating area, the primary air movement was generally downwards and the supply air was distributed from the upper-walls.

Altogether, the indoor air quality was at a good level using the displacement ventilation, although the temperature stratification was low. The air speed was generally reasonable while comparing with the typical speeds in the large enclosures involving ventilation jets and heat sources. The supply air temperature, variable air flow rates and retractable stands had a notable effect on the large-scale air movement. The important topic is also the design of the supply openings below the seats. The primary air movement was produced from the combination of pressure differences, convection flows, heat sources and cold surfaces.

The control of relative humidity is important, not only for avoiding mold, but also for avoiding too dry conditions. CFD-simulation is a useful tool in the analysis. The CFD results can be used in developing the design solutions and in managing the indoor climate together with the building management system, because the advanced computing resources in the future will enable efficient URANS and LES simulations with measured sensor data. Overall, the air distribution strategy might be further improved by implementing the individual ventilation strategies and heating strategies, and by discharging the supply air from below the seats at the whole seating area.

Acknowledgment

This study has been done at Aalto University (2015-2016), Tekes RYM SHOK indoor environment programme (2011-2015), Euroacademy on Ventilation and Indoor Climate CLIMACADEMY (2008-2009) and Tekes Arena-IC project (2007-2010) in collaboration with the Finnish Institute of Occupational Health. The authors wish to thank Building Counsellor Mr. Reijo Hänninen for the research project and Emeritus Professor Dr. Timo Siikonen for guidance during these years. The study has been funded by Granlund Oy, Tekes, Aalto University, Marie Curie Actions, Oy Swegon Ab and partner enterprises.

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