Proceedings of the World Congress on Mechanical, Chemical, and Material Engineering (MCM 2015)
Barcelona, Spain – July 20 - 21, 2015
Paper No. 276

The Experimental Validation of Numerical Modeling of the Air Distribution in the Indoor Ice Rink Arena

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Abstract –One of the important applications of CFD technique is the modeling of the air distribution system in ventilated and air conditioned objects. This method is used for research facilities where there are complex flow phenomena, for example in ice rink arenas. The aim of this study was numerical modeling of humid airflow and heat in an actual ice rink arena, using ANSYS CFX code based on CFD technique. The simulation results were validated by comparison with the experimental results in an actual object.

Keywords: numerical modeling CFD, ventilation, air distribution, ice rink, thermal and humidity conditions, experimental validation

1. Introduction

One of the most important applications of the Computational Fluid Dynamics (CFD) technique, based on the basic equations of fluid flow and heat, is the modeling of the air distribution system in ventilated and conditioned objects. Therefore, this method can be used for researching the flow of air, heat, moisture and contaminants in such facilities, as well as the computer aided design of ventilation and air conditioning. It can aid us in obtaining results such as air velocity and temperature distribution, or the air humidity level. Moreover, it allows us to make an assessment of ventilation work effectiveness, already at the design stage. CFD modeling is furthermore a useful tool for the optimization of air distribution systems from the viewpoint of the fulfillment of requirements for users.

This method is used for facilities, where due to complicated geometry and complex flow phenomenon, it is difficult to predict the effects of a ventilation system by using traditional engineering methods. One such example is an indoor ice rink arena, where on the one hand we have to deal with the temperature of an ice surface below 0°C, while on the other hand we must consider the presence of people. Ventilation must meet the requirements of both parties.

Until now, air distribution issues in ventilated ice rink arenas are relatively poorly understood and underpinned by research, which causes the need for research on this subject. In literature only a few experimental and numerical studies about the ventilation systems in indoor ice rinks can be found- for example, in the ice rink arenas in Montreal (Canada) (Belache et al., 2005), in Greater Boston (USA) (Yang et al., 2001) and Gjøvik Olympic Cavern Hall (Norway) (Annex 26, 1998). The results of numerical calculations of these objects were subject to experimental validation. Air temperature and velocity distributions were compared in the ice rink arenas in Greater Boston and Gjøvik. Additionally, in Greater Boston, CFD technique was used to validate the spread of contaminants. In the arena in Montreal numerical results were compared with measured air velocity in the real structure as well as using its physical model.

This paper presents the results of airflow, heat and moisture numerical modeling in the mechanically ventilated real indoor ice rink arena, using ANSYS CFX code, based on CFD technique. The validation of numerical calculation results was carried out by using measurements data in such an object. The aim was to verify whether the developed numerical model enables the appropriate mapping of the occurring phenomena in a ventilated ice rink arena.

2. Ventilation of Ice Rink Arenas - Specification and Requirements

The ventilation system in an ice rink arena should fulfill the following functions:

- Maintaining adequate thermal and humidity conditions for users of the ice rink arena, which is the function of ventilation or air heating,
- Removing excess moisture above the ice surface, which is the function of dehumidification (Lipska et al. 2011).

Therefore in one room the appropriate conditions must be provided due to presence of people (both spectators and skaters) and the risk of exceeding the permissible moisture level in the indoor air. Excess moisture is an important problem because it leads to the condensation of water vapor contained in the air and the formation of fog above the ice surface. Fog above the ice surface is a very adverse event due to the reduced visibility in the arena and the safety of skaters. In addition, condensation of water vapor in the air leads to the growth of mold, decay in wooden structures, and metal corrosion.

For the accomplishment of functions fulfilled by the ventilation, there are two types of air distribution systems in the ice rink arena. The traditional solution is the integrated system, which performs both functions together. The modern solution is the separated system, in which the ventilation/air heating function and the dehumidification function are separated. A detailed description of the air distribution systems has been discussed by Stobiecka et al. (2013).

In the ice rink arena it is recommended to keep following air parameter values:

- the indoor air temperature: 10÷12°C,
- the relative air humidity: 40÷65%,
- the air speed above the ice: ≤ 0,25 m/s (Sormunen et al., 2007).

The air speed value just above the ice surface is very important from the viewpoint of the thermal comfort of users. Moreover, low air speed minimizes the load on the ice making system, whilst too high air speed could cause ice melting, adversely affecting its quality.

Air parameters in an ice rink arena should also be adjusted in order to ensure that the temperature of the ice surface is higher than the dew point temperature of the indoor air. The same applies to the surface wall temperature of the ice rink arena, and in particular to the inner surface of the roof. A cold ice surface, directly opposite the ceiling, absorbs heat by radiation. As a result, the inner surface of the ceiling is colder than the air below it (Desert Aire, 2007).

3. Investigated Object

The investigated object was the real indoor ice rink "Tafla" located in Gliwice (Poland). The external dimensions of this structure are: length 66 m, width 37 m, and a maximum height of 11 m. Inside the building there is an ice rink with the dimensions 30×60 m (Fig.1). The use of the hall as an ice rink is expected throughout the period October to May. The ice rink is used for different kinds of sports and recreational activities, including curling.

The internal partitions of the studied ice rink arena are: the south wall, which is adjacent to the café, the ice rink floor and partially the north wall, at which there are technical facilities. The remaining partitions are external. The west wall is inclined at an angle 75° to the horizontal and is partially sunk into the ground.

The internal heat sources are: the lighting system, spectators and skaters. However, the indoor sources of moisture are only people and the temporary work of the ice resurfacer. Moisture gains also occur from an ice rink surface when the temperature is above the dew point of the indoor air. Moreover, moisture is supplied into the ice rink arena with the ventilation air and by infiltration.

Inside the investigated ice rink arena thermal and humidity conditions are maintained by mechanical ventilation with an integrated air distribution system. It is a continuous air ventilation system with the possibility of reducing air volume flow rate during a non-use time period of the object.

Regulation of supplied air by treatment in the air heater is planned in the ice rink arena. The temperature of air supply is controlled by two temperature sensors, mounted in the hall, cooperating with the automatic control system. The designed indoor air temperature amounts to +5°C. Air dehumidification is not anticipated.

The air handling unit, located on the roof of the technology building, is designed for 2 fan speeds:

- higher (second II) (supply airflow rate = 17500 m^3/h) - provided for a transitional period of time, when outdoor air temperature is over +5ºC;
- lower (first I) (supply airflow rate = 10500 m^3/h) - provided for a winter period of time, when outdoor air temperature is below +5ºC.

The ventilation air distribution is carried out by side air supply and air exhaust under the roof.

It consists of 17 long-throw jet nozzles with diameters of 160 mm (above ice surface) and 8 long-throw jet nozzles with a diameters of 80 mm (above tribune). Larger nozzles are inclined at an angle of 75° relative to the vertical axis of the ventilation duct, while the smaller at an angle of 45°. On the exhaust duct there are 17 rectangular exhaust grilles with dimensions 800×150 mm. The supply air duct is located at a height of 6 m, and the exhaust duct at a height of 9 m. Both ventilation ducts have a diameter of 1000 mm.

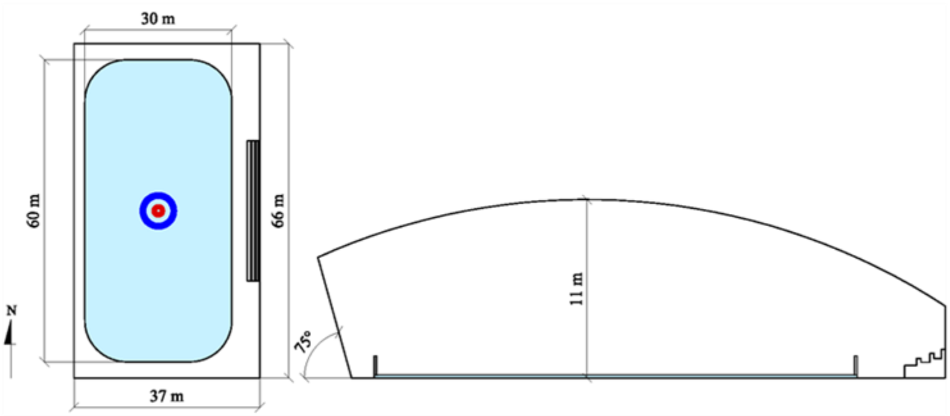

Fig. 1. Top view and cross-section of the ice rink arena "Tafla"

4. Measuring Process

In the investigated ice rink arena experimental research was carried out to obtain data for boundary conditions for numerical calculations and for validation of the results. Research included the following measurements:

- speed, temperature and relative air humidity above the ice surface,
- supply airflow rate in long-throw jet nozzles,
- temperature and relative humidity of the supply air and outdoor air,
- temperature of the ice surface.

The airflow visualization by smoke testing in the ice rink arena and thermovision measurements of the temperature distribution on the wall inner surfaces were also carried out.

Measurements were limited to one half of the ice rink arena, which is towards the northern partition. It was possible because of the approximate symmetry of the two halves of the hall. Due to the variable number of people using the ice rink it was not possible to carry out experimental measurements with the presence of people.

Eight–channel omnidirectional thermoanemometers, placed at 4 specific heights: 0.1 m, 0.6 m, 1.1 m, 1.7 m, in accordance with ISO 7726:2002, were used for measuring the speed and temperature of the indoor air. Measurements were made at 9 measuring points evenly distributed on the surface of the ice rink. An anemometer with a pitot tube was used for the measurement of air velocity (indirectly supply airflow rate) in long-throw jet nozzles. The air velocity was measured using a method based on dividing a circular cross section of the nozzle into rings, which number depends on the diameter of the nozzle. Temperature and relative air humidity loggers were used for measuring the temperature and relative humidity of supply air and indoor air above the ice surface. Outdoor air parameters were obtained from a metrological station located in the distance of about 500 m away from the object. An NTC probe for food was used to measure ice surface temperature. It was positioned in a drilled hole at a depth of 15mm. The measurement was made in the middle of the investigated half of the ice rink surface.

5. Numerical Model

The numerical model of the study object was worked out using the Ansys CFX code. It took into account the designed overall dimensions and geometry of the hall. The ice rink, lighting system and ventilation system were modeled (Fig.2). The XY symmetry plane was introduced because of the symmetry of this object. This enabled the numerical simulation to perform on an available server with more refinement of the discretization grid. The model did not take into account the presence of people, just like in the real object during the measurements.

The lighting system, consisting of 37 metal halide lamps, was modeled as rectangular heat sources located in the top part of the hall. Only those lamps, which were turned on during experimental measurements, were included as heat sources (marked in yellow color on Fig.2).

The long-throw jet nozzles were modeled as circular supply openings with a diameter of Ø160 and Ø80 mm, placed at a right angle on the ventilation supply duct, which was modeled as a circular duct. The exhausts were set as rectangular grilles with dimensions 800×150 mm.

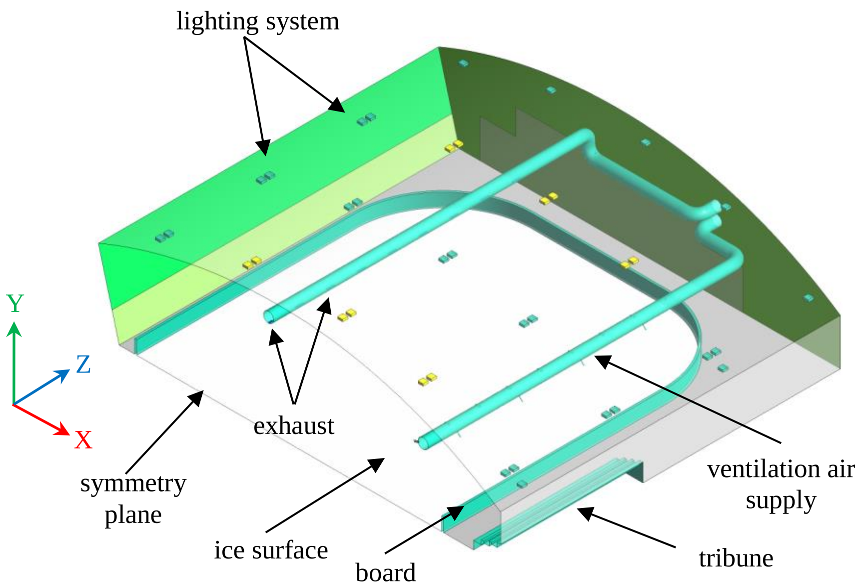

Fig. 2. The numerical model of half of the rink hall

Boundary conditions were prepared for the calculation case, which was followed by the installation work on the 2nd fan speed.

Heat transfer coefficients (obtained from technical documentation of the object) and outside temperature were set to internal and external walls. The exception was part of the north wall separating the ice rink arena from technical facilities. The mean temperature of its surface, obtained from thermovision measurements, was used.

The ice sheet was modeled as a surface of uniform temperature. Moisture flux has been expressed by the formula in accordance with the ASHRAE Handbook (2010):

$$\dot{W} = K \cdot (X_a - X_i) \cdot A_i \qquad (1)$$

$\dot{W}$ - moisture flux, kg/s

K - mass heat transfer coefficient, $K = 0{,}00023$ kg/(s·m^2)

x_a -specific humidity in air, kg/kg

x_i - specific humidity in saturated ice, kg/kg

A_i - ice rink area, m^2

The following averaged measured air parameters were set for supply openings: mass flow rate, temperature and specific humidity. Exhaust air mass flow rates were set for exhaust openings.

Table 1 shows the boundary conditions for the heat transfer between the walls and the environment. The remaining data required to perform the numerical calculations was presented in Table 2.

Table. 1. Boundary conditions for the heat transfer between the walls and the environment

Boundary condition	**Value**	**Boundary condition**	**Wartość**
Average outdoor air temperature	4.8°C	Heat transfer coefficient of the external walls (N, W, E)	0.187 W/m^{2}K
Mean temperature of the north wall surface separating the hall from technical facilities	3.5°C	Heat transfer coefficient of the wall in the ground (E)	0.140 W/m^{2}K
		Heat transfer coefficient of the floor	0.203 W/m^{2}K
Ground temperature	2.0°C	Heat transfer coefficient of the roof	0.301 W/m^{2}K

Table. 2. Boundary conditions for the indoor thermal and humidity conditions

Boundary condition	**Value**	**Boundary condition**	**Value**
Average ice surface temperature	-4.5°C	Average temperature of supply air	14.2°C
Specific humidity in saturated ice	0.0026 kg/kg	Average mass flow rate of supply air = Average mass flow rate of exhaust air	2.394 kg/s
Average specific humidity of supply air = Average specific humidity of outdoor air	0.0048 kg/kg	Lighting power	1206.15 W/m^2
		Ice surface emissivity	0,96

6. Calculation Process

The numerical calculations were carried out using ANSYS CFX code. The simulations have been performed for steady-state, three-dimensional and non-isothermal conditions. The CFD model used the Shear Stress Transport turbulence model. Thermal radiation between walls, ice surface and objects located inside the ice rink arena was completed using the Discrete Transfer Model.

Discretization of model equations was solved by the Finite Volume Method.

The applied unstructured discretization grid was composed of 12574808 cells, mostly of tetrahedral elements, and 3161134 nodes. It also included boundary layers and local refinement around the outlet, inlet openings and just above the ice rink surface. The basic mesh element size was 35 cm, with a refinement over the ice surface up to 7 cm.

The calculations were carried out using the iteration method as long as the convergent solutions were obtained.

7. Results

Fig. 3 shows the distribution of the particular air parameters in the ice rink arena: speed, temperature and relative humidity, in the vertical plane XY (Z = 15 m) and the horizontal plane ZX (Y = 0,6 m).

In Fig. 3a, showing the air speed distribution, there are very low values of this parameter: just above the ice surface and locally in the upper part of the ice rink arena. In these regions are so-called "stillness zones”- areas where the air speed is extremely low, or non-existent, additionally near the tribune also.

Air speed did not exceed the permissible value of 0.25 m/s at no point in the area near the ice sheet. Air speed reached a maximum value of 0.34 m/s at a height of 0.6 m.

On the basic of the temperature air distribution (Fig. 3b), it can be concluded that the indoor temperature oscillated around 4.0°C across almost the entire area of the ice rink arena. Lower values of the air temperature occurred in the area over the ice surface, up to the height of the board. Air temperature oscillated around 3.0°C at a height of 0.6 m above the ice surface.

Analyzing the distribution of relative air humidity (Fig. 3c) it can be concluded unequivocally that there are high values of this parameter in the region above the ice surface, up to a maximum value of 100%. This means the possibility of the formation of fog. In addition, it indicates the condensation of water vapour, combined with the freezing of the ice surface. In the rest of the ice rink arena the relative air humidity was in the range of 80÷90%. Lower values of this parameter were outside the ice surface, i.e. between the walls and the board.

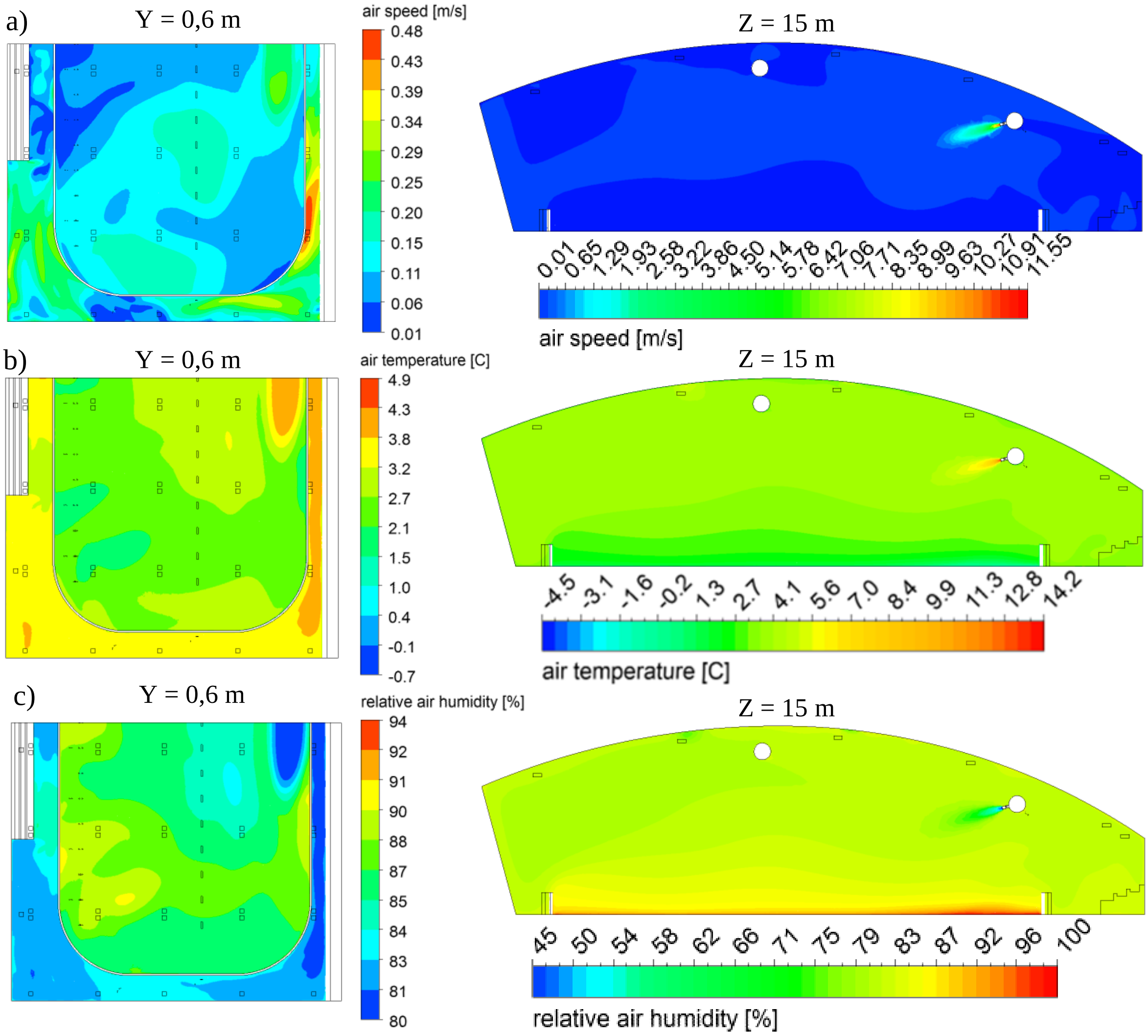

Fig. 3. The distribution of a) air speed b) air temperature c) relative air humidity in the ice rink arena

8. Validation

Validation of numerical results was made using the results of previously carried out measurements to verify the correctness of occurring phenomena simulation in the ice rink arena by numerical model.

Image of airflow (air velocity of 0.7 m/s) was first compared with visualization by smoke test, carried out during measurements (Fig. 4). An explicit deflection of the supply stream at a height of about 3 m was observed both in the CFD model and in reality. In both cases supply air did not spread into the ice surface, maintaining in this way the recommended speed value in such area. Supply air was directed toward the exhaust openings on the ventilation duct.

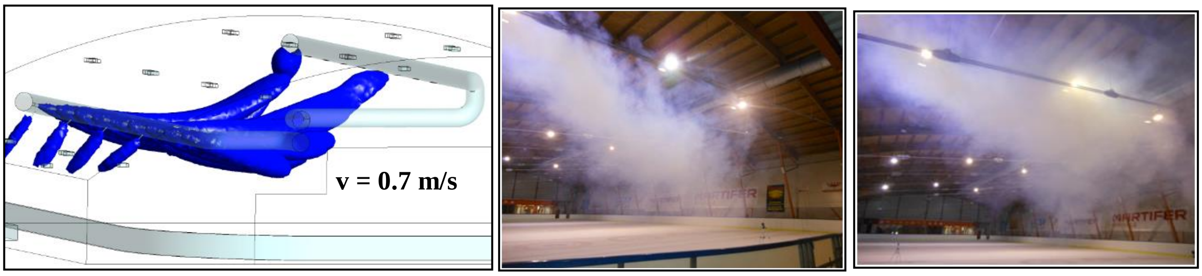

Fig. 4. The comparison of airflow CFD image (left) and visualization by smoke test (middle, right)

Fig. 5 shows the comparison of numerical and experimental results for the following parameters of air: speed, temperature, relative and specific humidity.

The results were compared to mean values of 9 measuring points on the investigated half of the ice rink arena, at the following heights: 0.1 m, 0.6 m, 1.1 m, 1.7 m.

Increment of air speed with height can be clearly seen from the results (Fig. 5a). The difference between the measured and obtained from calculations value did not exceed 0.05 m/s. The measurement device error was 0.02 m/s, therefore such differences are slight. However, the maximum calculated relative error was 50%, which is related to very low air speed values.

Considering the air temperature distribution in the ice rink arena at various heights (Fig. 5b) we can observe very similar values of measurements and numerical calculation results, beyond the measuring point at a height of 0.10 m. This error could be caused by a too small grid refinement in this region, where there is a high vertical gradient of air temperature. A similar problem was encountered in (Annex 26, 1998), where it was found that too rare grid discretization could cause weaker thermal stratification. At the other heights, i.e. 0.6 m, 1.1 m, 1.7 m, differences were almost in the range of measurement device error, which was 0.2°C. Except the measuring point at a height of 0.10 m, the relative error for the air temperature did not exceed 7%. Moreover, there was a similar curve of air temperature distribution from the numerical calculations in relation to the measurements. The results for the relative air humidity in the area above the ice surface were also similar to each other (Fig. 5c). The relative error for this parameter did not exceed 11%.

The specific humidity was also compared to a more appropriate assessment of air moisture level (Fig. 5d). Specific humidity above the ice surface, determined indirectly on the basis of the measurement results, was very close to the values obtained from the numerical calculations. As before, an exception was the point at a height of 0.10 m above the ice sheet. On the other heights, i.e. 0.6 m, 1.1 m, 1.7 m, differences were in the range of the intermediate maximum device error for measured temperature and relative humidity, which was approximately 0.0006 kg(H_2O)/ kg(dry air).

Excess of permissible level of specific air humidity was also noticed at the distribution of this parameter. The permitted value of specific air humidity is 0.0026 kg(H_2O)/ kg(dry air) (indicated in Fig. 6d) for a mean ice temperature of -4.5°C. It is a limit value for which the indoor air dew point temperature is equal to the ice temperature. Above this value, there is a risk of fog formation above the ice surface. This means there was moisture condensation on the ice sheet, which was confirmed by preparing the moisture balance in the object.

9. Conclusions

1. Used numerical modeling method allowed us to take into account most of phenomena associated with the flow of air, heat and moisture in the ventilated ice rink arena.
2. The results of the experimental measurements were necessary to prepare the CFD model of the investigated ice rink arena: to obtain the boundary conditions and to validate the numerical calculations.
3. The results of the numerical calculations enabled us to see how thermal and humidity conditions are forming inside the studied ice rink arena. Air parameters such as temperature and relative humidity were different from recommended. The air speed over the ice surface did not exceed the permissible value.
4. Based on the validation of numerical calculations carried out for the various parameters of the air, using the results of measurements and smoke testing, it was found that the CFD model of the ice rink arena was able to map the real conditions in the object. The exception

was the only area directly above the ice surface, where the differences were larger. Improvement could be achieved by further refinement of the discretization grid.

5. The numerical model will be useful in the future to help solving the problems related to the effectiveness of the ventilation system operation under adequate thermal and humidity conditions for users, and maintaining a good technical condition of the object.

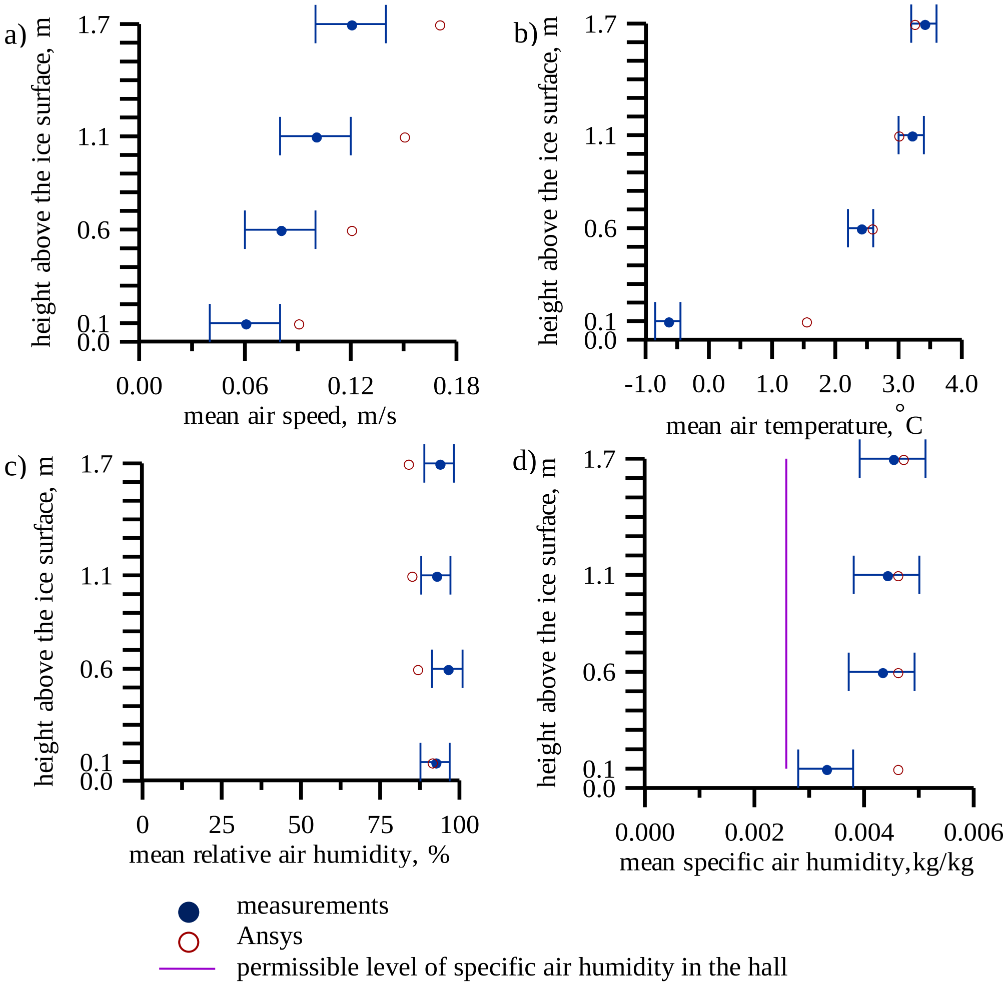

Fig. 5. The comparison of experimental and numerical calculation results for a) mean air speed, b) mean air temperature, c) mean relative air humidity d) mean specific air humidity at different heights above the ice surface

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