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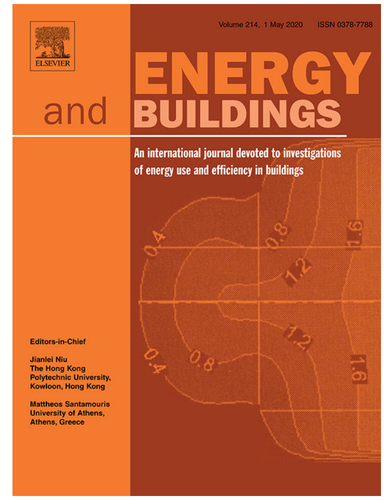

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Developing energy calculation methodology and calculation tool validations: application in air-heated ice rink arenas

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Abstract

The ice-pad refrigeration and space heating energy demand in air-heated ice-rink arenas dominate the total energy demand. Accordingly, the overall energy usage is a weak function of ambient temperature. This makes it possible to analyze the energy demand in steady-state conditions. This study aims to develop a simplified calculation tool to enable non-expert users to accomplish the following: First, they can roughly calculate the energy requirements/costs of ice rinks using a simplified calculation method. Second, they can calculate energy performance value, which is mandatory for obtaining energy performance certificates. Third, they can clarify discrepancies between real and official energy performance values. The study verifies the applicability of the tool, which is analyzed by applying theoretical and physical principles in steady-state conditions. The tool is intended to be applied in the early design phase to assist authorities in developing clear vision about energy consumption/costs and making right choices. The accuracy of the tool is validated against dynamic energy simulation. The tool results deviate upto 10% with IDA-ICE results in refrigeration and space heating demands but approximately 30% in cooling/dehumidification demands, where steady-state calculation causes considerable faults. However, the tool assesses overall yearly energy costs with 8% inconsistency compared with simulation results.

Keywords:

process-dominated buildings, energy performance, E-value, energy certificate, steady-state analysis, simplified calculation tool

Nomenclature

P_{in}	Saturation pressure entering boundary layer	Q	Heat transfer
P_{ice}	Saturation pressure at ice surface	$\dot{Q}$	Heat rate
RH_h	Relative humidity at height h	Q_{cond}	Heat transfer due to condensation
dp	Water vapor pressure difference (Pa)	Q_{conv}	Heat transfer due to convection
dp_{atm}	Converted to atmospheric pressure (atm)	φ_{air}	Enthalpy of the air
T_{air}	Air temperature	φ_{wv}	Enthalpy of water vapor
T_{in}	Air temperature at 0.1m above ice	$\Delta\varphi$	Enthalpy difference
$T_{S\text{-}ice}$	Ice surface temperature	C_{air}	Specific heat of the air
$T_{S\text{-}ceiling}$	Ceiling surface temperature	X_{air}	Specific humidity
$T_{1.5}$	Air temperature at 1.5m above ice	C_{wv}	Specific heat of water vapor
ε_i	Emissivity of the ice surface	c_{ext}	CO2 level of extract air
ε_c	Emissivity of the ceiling	c_{sup}	CO2 level of supply air
F_{ci}	View factor between ceiling & ice-pad	c_{fresh}	CO2 level of fresh air
R_i	Surface resistance to radiation	q	Volumetric flow rate
h_{conv}	Convection heat transfer coefficient	U	Thermal transmittance
h_{cond}	Conduction heat transfer coefficient	v	Indoor air velocity
h_{condn}	Condensation heat transfer coefficient	$q_{lighting}$	Total radiation due to lighting
$h_{rad.}$	Radiation heat transfer coefficient	$P_{lighting}$	Lighting power
$C_{freez.}$	Specific latent heat of water freezing	U_t	Lighting utilization factor
$U_{ins.}$	U-value of insulation layer under ice pad	σ	Stefan-Boltzmann constant
α	Outdoor air fraction of the supply air	$N_{Players}$	Number of players
h_{ci}	Convection heat transfer coefficient between ceiling and indoor air	$N_{Spectators}$	Number of spectators
		ρ_{air}	Air density

1. Introduction

The energy consumption of ice rinks is among the highest in the building sector. Whereas less efficient arenas consume up to 3,500 MWh/year [1], an energy-efficient ice rink arena may consume approximately a quarter of that [2].

Energy usage in ice-rink premises has a strong correlation with the presence of a large indoor ice surface. In their earlier publications, authors widely studied interactions between the ice surface and indoor air conditions, representing the major cause for refrigeration and heating loads [3,4]. In fact, the energy use of the refrigeration process plays a dominant role in the overall energy consumption of ice rink arenas. Simultaneously, the refrigeration plant generates a large quantity of condensation heat, which may simply be utilized for space-heating purposes. Figure 1 schematically illustrates the contribution of each section to an ice rink arena's energy consumption, whereas the width of the arrows approximately represents the energy usage portion of each sector.

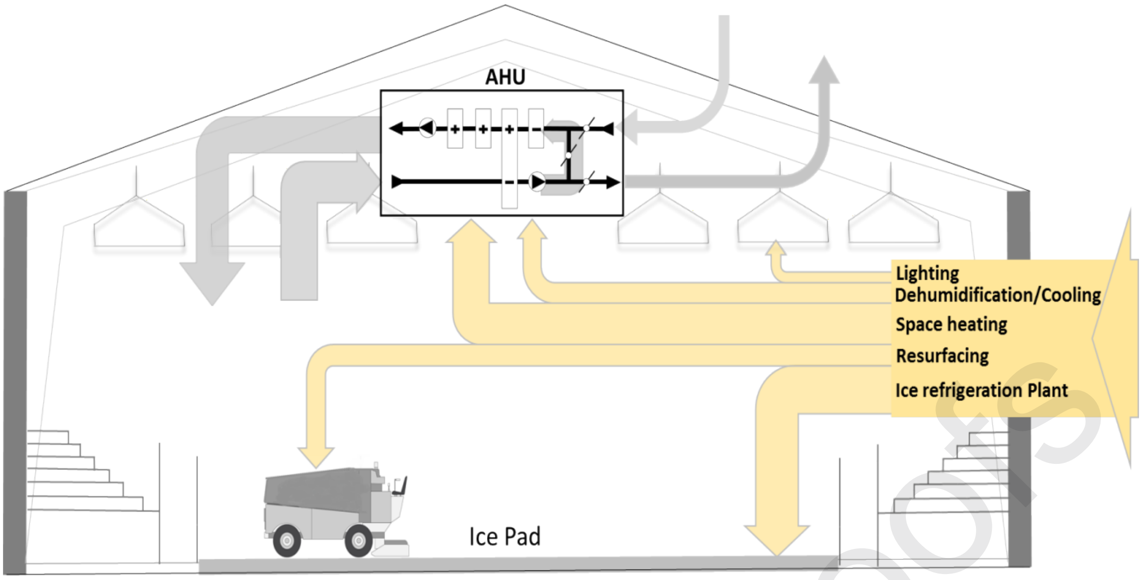

Fig. 1. Energy use scheme of ice rinks.

Seghuani et al.[5] studied the effects of ceiling emissivity on the refrigeration load and showed that an approximately 20% reduction of the refrigeration load can be achieved by reducing the ceiling emissivity from 0.9 to 0.05. Similarly, Daoud et al. [1] studied the effects of the ceiling emissivity and temperature set point in the stands. They conducted a parametric study to calculate the heat transfer toward the ice pad and showed that a low ceiling emissivity can significantly reduce the refrigeration load, and consequently, reduce the yearly energy consumption of ice rink arenas.

Caliskan et al. [6] analyzed energy and exergy in ice rinks; they investigated the effect of varying reference state temperatures on the system exergy efficiency and found in which the state temperatures exergy efficiency would be minimized and maximized. Erol et al. [7] conducted an exergy analysis on an ice rink refrigeration system using both conventional and advanced exergo-economic analyses. They discussed on possible solutions to reduce inefficiencies and finally determined that about 47.15% of the total exergy destruction of the system was avoidable.

Seghuani et al. [5,8]worked on the prediction of the yearly energy consumption of indoor ice rinks, considering the effects of various climates. They studied the energy requirements of four indoor ice rinks in North American cities with extremely different climates. They showed that the annual refrigeration load among those four locations did not vary significantly (approximately 7%) despite their significant climatic differences [8]. This outcome supports the reasoning that is expressed and discussed in this paper.

The demand for reliable predictive building energy models is rapidly increasing as the building sector and its energy use become a prominent concern for society. Lü et al. [9] proposed a new method for modeling and forecasting energy consumption in heterogeneous buildings using a physical–statistical approach. Khalid et al. [10] also developed a predictive model to calculate, compare, and analyze cooling capacity and heat loads of ice rinks.

Since ice refrigeration and its maintaining process are the main factors in the overall energy consumption of ice rinks, we can consequently argue that the energy requirements in ice rinks can be analyzed and estimated in steady-state conditions [1]. Tuominen et al. [11] developed a calculation tool for larger scale building stocks to assess the effects of various energy measures on their energy consumption in Finland. Later in this section, we argue for the development and application of a calculation tool based on a simplified steady-state approach.

Matsuo et al. [12] analyzed the measured data and energy-saving measures using building performance simulation. They proposed ways to improve the management and reduce the energy consumption of buildings.

Sun et al. [13] studied energy saving strategies and conducted a field investigation on various data measurement schemes. They concluded that the running costs of refrigeration units represent a significant share of the overall running cost. They finally proposed an energy-saving operation strategy to reduce energy consumption.

The need for the tools outlined above is generally expressed by investors and authorities. They want to make decisions about new projects since they have no clear vision about their future energy need and costs. However, due to the relatively high energy need and costs of indoor ice rinks, it is more requested to roughly assess their future energy costs in advance to enable investors and authorities to make the right choices. Implementing dynamic energy simulation or applying Computational Fluid Dynamic (CFD) models is time consuming; moreover, these methods are only applied by experts later in the design development phase of a project. Therefore, a simplified tool that can be applied even by non-expert users in the early stage is required to estimate future energy use/costs.

Energy analysis is becoming more significant due to increasing requirements of energy efficiency regulations in the European Union, as well as in Finland. The regulations and guidelines regarding the energy efficiency of buildings in Finland consist of two parts: First, they involve calculating the annual primary energy (E-value), which must comply with the general principles of the Regulations and guidelines [14]. Second, they necessitate determining an energy performance certificate according to the reference classification list in the guideline [14].

To calculate the E-value, the annual delivered energy is multiplied by primary energy factors per heated area (kWh/m^2 a). The calculation of energy performance includes all energy delivered to the building and the equipment to maintain a comfortable indoor climate, while the process energy use is excluded. The “E-value” is developed for ordinary buildings. It is not intended for process-dominated buildings, such as ice rinks, swimming pools, grocery stores, servers, or kitchens. In cases like that of indoor ice rinks, the refrigeration process is not included in the E-value calculations. Therefore, the E-value does not reflect a realistic energy efficiency scheme of such buildings.

Finnish regulations regarding the energy performance indicates that primary energy, the “Energy Performance” E-value, is required for all “ordinary” buildings, whereas ice rinks belong to the ninth category of premises, “Other Buildings,” implicating that no limited E-value is required. However, the regulations obligate calculation of the E-value which is required for the energy performance certificate to be issued [14]. It is mandatory according to building regulations to obtain the energy performance certificate to determine energy rating of the buildings. The energy performance of buildings is classified on a scale from A as the finest class to G as the poorest class.

To obtain the energy performance certificate, the energy efficiency (E-value) for the ice rinks must be calculated using the input data taken from design documentation for new and existing buildings, if available. If design values are not available, then measured or estimated data for internal loads, indoor temperatures, ventilation air flow rates, and running times should be applied. The classification for sport halls is determined based on the E-value ranges according to Table 1.

Table 1. Class Ranges for Energy Performance Certificate

Energy efficiency classification	E-value [$kWh_E/(m^2\ a)$]
A	E-value ≤ 90
B	91 ≤ E-value ≤ 130
C	131 ≤ E-value ≤ 170
D	171 ≤ E-value ≤ 190
E	191 ≤ E-value ≤ 240
F	241 ≤ E-value ≤ 280
G	281 ≤ E-value

In this paper, we propose a new method of assessing the energy consumption/costs of ice rinks. Our focus is on developing a simplified calculation tool based on steady-state analysis of physical principles. The method is implemented in the calculation tool using Finnish climatic data, but it is not principally limited to this climate. It is made to be compact and simple to use. The proposed method applies basic physical and heat transfer principles to derive an efficient, explicit procedure for calculating refrigeration, thermal, dehumidification/cooling, and electricity demands. We provide a detailed description of the novel calculation tool and validate the model using both field measurements of two ice rinks in Finland and simulation results using IDA Indoor Climate and Energy (IDA-ICE).

This paper is structured so that the method is presented in section 2 by introducing the inputs of the calculation tool, energy simulation tool, and case study building. Then, the theoretical principles are presented in section 3, the results of the tool are described in section 4, and calculation examples are given in section 5. Parametric analysis is applied in section 6, and finally, the discussion and conclusion are given in sections 7 and 8, respectively.

2. Method

As stated above, the major portion of energy consumed in air-heated ice rink arenas directly or indirectly relates to the ice pad. This includes ice creation, ice resurfacing, and maintaining indoor air, all of which are weak functions of outdoor weather conditions. Therefore, energy demand can be analyzed and calculated using a simplified tool with reasonable accuracy. However, developing the calculator needs some general assumptions to simplify the calculation; this will make the system mathematically less complicated and the calculations easier.

General concepts and assumptions: To implement the calculator, we need several assumptions for the parameters that may vary depending on the time or place. First, the heat losses through external partitions are calculated based on average monthly outdoor temperatures. Second, the indoor temperatures at a position of 1.5 m above the ice and under the ceiling are assumed to remain constant. Third, the ice surface temperature, as well as ceiling surface temperature, are assumed to be kept approximately constant. Fourth, the vertical room temperature is assumed to increase linearly with a constant slope called a *temperature gradient* in both the calculation tool and simulation model. Fifth, the coefficient of performance (COP) of the refrigeration machines changes depending on the times and refrigeration loads, particularly during the resurfacing periods. However, to simplify the calculation, we assume that the COP remains constant throughout the operation. Therefore, a realistic average of the COP, which is mainly obtained from the manufacturer, is applied as input to the tool.

2.1. Input data

In the first page of the calculator, the user must provide the basic input data required for the calculation process. Each category of the required input data is described in the following paragraphs. They are categorized according to building dimensioning, its specifications, and the precise indoor conditions. The list of input data is shown in Fig. 2.

Ice rink energy calculation—Aalto University—2019	
Ice rink energy calculator inputs—Enter the data into the green cells	
Ice rink and the whole building	**Input data**
Ice rink length [m]	65.0
Ice rink width [m]	35.0
Ice rink surface area [m²]	**2,275**
Total building area [m²]	3,000.0
Facility building area	**725.0**
Ice rink height [m]	6
Ice rink volume [m³]	**13,650**
Overall building volume [m³]	15,825
facility building volume [m³]	**2,175**
Ice layer length [m]	58
Ice layer width [m]	31
Ice layer surface area [m²]	**1,798**
Spectators	
Number of spectators [kpl]	500
Indoor condition	
Temperature 1.5 m above the ice [°C]	5
Relative humidity 1.5 m above the ice [%]	60
Temperature under the ceiling [°C]	12
Temperature gradient [°C/m]	**1.56**
Ice resurfacing	
Resurfacing on weekdays (Mon-Fri) [kpl]	8
Resurfacing on weekends (Sat-Sun) [kpl]	12
Daily resurfacing on average	9.14
Resurfacing water temperature [°C]	40
Resurfacing water volume [lit]	500
Ice layer condition	
Ice thickness [mm]	40
Ice surface temperature [°C]	−3.5
Lighting	
Lighting power [W/m²]	10
Lighting total power [W]	**22,750**
Lighting utilization rate [-]	0.25
Emissivity of the coating [k/e]	K
Thermal insulation of external surfaces	
External walls average U-value [W/ (m² °K)]	0.26
Roof average U-value [W/ (m² °K)]	0.26
Refrigeration plant	
Coefficient of Performance (COP) of the Refrigeration machines > COP	2.5
Percentage of condensation heat recovered [%]	30
Snow melting	
Snow defrosting (k/e)	K
Thermal insulation underneath the ice rink (100 mm/200 mm)	100
Building location / zone (Zone I / Zone III / Zone IV)	Zone I
Air handling unit	
Total airflow including recirculated air flow [(dm³/S)/m²]	2
Outdoor air contribution (including leakage air) [-]	0.05
Heat recovery efficiency [-]	0.7
Specific fan power (SFP)	0.8
Cooling coil position (e/j)	E
Energy prices	
Electricity prices, daytime [€/kWh]	0.08
Electricity prices, night [€/kWh]	0.06
District heating prices [€/kWh]	0.05
Selling prices of extra heating energy [€/kWh]	0.01

Fig. 2. Overview of the first page of the calculator where the user enters input data.

Dimensioning of the building: In the first step, the length, width, and height of the arena hall; the length and width of the ice pad; and dimensioning of supplementary building facilities are asked for. The tool then calculates the hall and the ice pad area, additional building area, and volume of the arena.

All the calculated values in this page are aligned to the right. The maximum number of spectators that the arena is designed for is requested in the next step.

Indoor air: Indoor air temperatures at the heights of 1.5 m above the ice pad and at the ceiling level (under the ceiling), as well as relative humidity at a height of 1.5 m, are given as input data to the tool. The tool then calculates the temperature gradient based on the temperatures and heights. The temperature gradient is an important measure to demonstrate the heat flux between the ice surface and the indoor space. This has been extensively discussed in the authors' earlier articles [3,4]. The linearized temperature gradient curves are shown in Fig. 3.

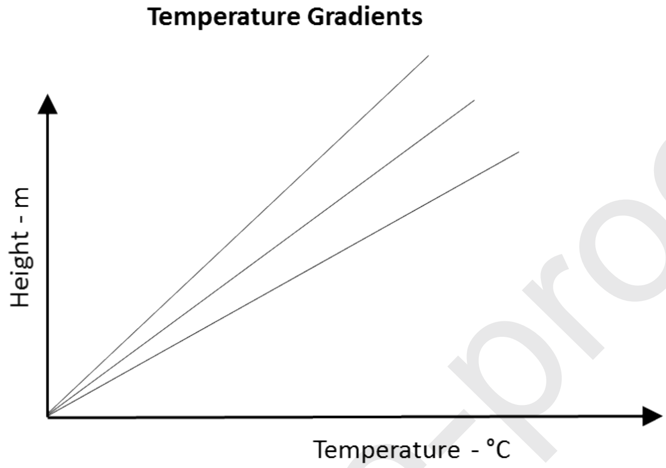

Fig. 3. Linearized temperature gradient curves.

Ice specifications: The ice surface temperature and ice thickness are asked for in this step. Then, the ice surface maintaining information, such as the ice resurfacing periods, and the resurfacing water temperature and volume are asked for.

Lighting: In this stage, the required lighting power per square meter area, the lighting utilization factor, and the coating emissivity factor (either 0.2 or 0.8) of the internal surfaces are specified to the calculator. The tool calculates the total power required for lighting with the given data.

Building envelope (U-Value): The U-values of walls and ceiling are asked for in this stage.

Refrigeration plant and snow melting: The COP of the refrigeration machines and percentage of recovered condensation heat are asked for in this part. The tool also asks whether snow melting is implemented, either indoors or outdoors. In addition, the thickness of the insulation layer under the ice pad is asked for here.

Location: According to its National Building Code, Finland is divided into four climatic zones for calculating the energy consumption of buildings; these are labelled Zone I to Zone IV. These climatic zones are mainly defined based on 30 years of air temperature data from representative meteorological stations for each climatic zone. Zone I includes all the southern areas of Finland, with relatively high outdoor design temperature in winter. Zones II and III are located in the middle, and Zone IV includes all the northern parts of Finland, with the severest winter conditions. The outdoor winter design temperatures of Zone I up to Zone IV are −26°C, −29°C, −32°C, and −38°C. These winter design temperatures are close to the average of the lowest hourly temperature of the year for 30 years. The tool is able to make the calculations based on three of the climatic zones—Zone I, Zone III, and Zone IV. For Zones I, III, and IV, weather data from Vantaa, Jyväskylä, and Sodankylä, respectively, are used. As the average monthly temperature of Zone II has a minor difference from that of Zone I, similar data on Zone I are used for Zone II. Consequently, all the calculations are implemented based on the yearly outdoor temperature and humidity of the selected zones.

Ventilation machines and air handling units (AHUs): In this section, the tool asks for the total air flow, including recirculating air; outdoor share of supply air; heat recovery temperature ratio; and specific fan power (SFP). The user must also provide information about the cooling coil location with the two options, located either before or after the heat recovery.

Outdoor air contribution of supply air: The user must provide the outdoor share of the supply air. The typical range is instructed to be within 0.1 up to 0.3.

Cooling coil location: The user must specify whether the cooling coil position is before or after the heat recovery.

Condensation heat recovery: There is a large quantity of condensation heat that is produced in the refrigeration process; this is depleted, but it can be utilized in the heating system. The percentage of condensation heat utilized in the heating system must be specified by the user as input to the tool. This is not normally more than 50% of the entire condensation heat.

Energy prices: Prices of various energy forms need to be added to the calculation tool; these include electricity prices for day and night, district heating prices for both purchase and selling.

Supply air heating and total heating demands: The supply air heating is similarly calculated by summing up external partition heat losses, heat loss due to cooling effects of the ice pad, and fresh air heating demand. The total heating energy consumption is calculated, summing up the total ventilation heating, heating energy required for domestic hot water and resurfacing, heating energy required for freezing protection of the ground under the ice pad and heating energy for snow melting (if applicable).

Refrigeration demand: The refrigeration energy demand of the ice pad is calculated by adding up the total heat load of the ice surface through radiation, convection, and conduction and the load due to resurfacing. In addition, the total cooling energy demand is calculated by summing up the cooling energy required for refrigeration of the ice pad, cooling energy demand for dehumidification and cooling energy to compensate for the summer heat load through external partition. The total generated condensation heat is also calculate summing up electricity consumption of refrigeration machinery with generated cooling energy. The percentage or share of recovered condensation heat given as input is used to calculate the quantity of utilized condensation heat simply by multiplication.

Electricity consumption [MWh]: The electricity power of the refrigeration plant is calculated by dividing the total refrigeration load by the COP, and a similar calculation is used for the electricity power of dehumidification/cooling. The electricity consumption of the refrigeration plant is calculated by multiplying the refrigeration machines' power by total yearly times in hours. The same applies for the electricity consumption of dehumidification. The electricity consumption of lighting is calculated using the lighting total power by lighting utilization factor multiplied by total yearly times in hours.

The supply fan's full power is assumed to be functioning at 100% of the total yearly times and the exhaust fan at 20%. The SFP is assumed to be 2 kW/(m^3/s) divided between the fans by giving 1.3 to the supply fan and 0.7 to the exhaust fan. Finally, the electricity consumption of AHU fans is calculated by the multiplying total ventilation flow rate by "SFP" by 1.2 by total yearly time in hours.

Annual consumed or sold energy [MWh]: The overall electricity consumption is calculated by summing up electricity consumption of the refrigeration plant and cooling/dehumidification, lighting, and AHU fans. The overall electricity consumption is considered as purchased electricity, as there is no on-site electricity generation.

Since the total required heating energy is calculated as in section 4.1, and the total generated and recovered condensation heat is calculated as in section 4.3, the overall purchased or sold district heating energy can be computed using a simple if-then logic. This is implemented so that, if the recovered condensation heat is greater than the overall required heating energy, then it subtracts them, and the resulting additional heat will be shown as sold heating energy. If the recovered condensation heat is less than the overall heating energy demand, then it is subtracted, and the result will signify the purchased district heating.

Annual energy costs: The primary energy factors (PEFs) in Finland are required to calculate the E-value of the ice rink. These factors are shown in Table 2 for various energy forms according to the regulations in Finland.

Table 2. Primary Energy Factors in Finland

Electricity	1.2
District heating	0.5
District cooling	0.28
Renewable fuels	0.5

The annual energy costs and overall E-value are calculated using the sum of delivered and exported energy multiplied by energy prices [€/kWh] and the subsequent PEFs.

2.2. Energy Simulation (IDA-ICE)

An energy simulation tool named IDA-ICE (IDA "Indoor Climate and Energy"), v. 4.7.1 with the Ice Rinks and Pools 0.912 add-on, was used to model the ice rink arenas in this study. IDA-ICE models the building and simulates its performance against internal and external loads with typical meteorological conditions for Helsinki, Finland, considering the operation of the energy system and its controllers while maintaining desired indoor conditions. It also accounts for both the ice-rink process and the air handling unit (AHU) with variable air volume (VAV) strategy operating simultaneously in the space.

IDA-ICE is a well-validated software program; validation was conducted using experimental data, with the IEA BESTEST [15] cases included in the ASHRAE [16] and CIBSE [17] procedures, as discussed in [18]. In this study, the measurements are implemented for only a few weeks and include ice rinks with their specific characteristics. Then, the IDA-ICE model of the ice rink is set to the same characteristics as each of the measured ice rinks and simulated for the same time periods of the year, as measurements have been conducted using typical meteorological conditions for similar locations. After model calibration, IDA-ICE can simulate and obtain the results for any proposed ice rink with any planned and set characteristic for any time period of up to a year.

IDA-ICE is used for validation, first because we need annual results and taking yearly measurements is a time-consuming and extremely slow process. Second, and more important, in parametric analysis, we need to change parameters and obtain the energy consumption results. It is not only problematic but also costly to change some characteristics of the system or the building for the sake of measurement in such large buildings with huge machineries. Normally, we simulate for a 1-year period with typical meteorological conditions for Helsinki, Finland, except for those simulations carried out for comparison with the measurements that were performed for several days only. This is how the IDA-ICE model is validated.

We apply IDA-ICE to validate the calculation tool and its results. It is used in parametric analysis to compare its results with the results of the calculation tool to validate the tool's results and accuracy.

Theoretical differences of IDA-ICE and the calculation tool: The calculation tool is developed based on steady-state conditions using average monthly outdoor temperatures. Therefore, radiation and convection heat transfers are calculated using a simplified condition, as explained in detailed in section 3 of this paper. The main differences between IDA-ICE and the calculation tool are described below.

First, IDA-ICE mainly uses a similar physical concept and thermodynamic equations, but it does so dynamically. This means that all the calculations are applied once in every time step, which is 0.1

hours, considering the instant indoor and outdoor climatic conditions, accurate occupants' presence and their loads for different times/days, and so on. Consequently, the demand response control strategy responds to the demands based on temperature, relative humidity, and CO_2 measurements from the hall space according to occupants' instant presence and activities to keep humidity and CO_2 within an acceptable level.

Second, the radiation heat transfer is the largest contributor among heat transfer forms in ice rinks [1]. Both calculation tool and IDA-ICE apply similar radiation heat transfer concepts and equations. The difference is that IDA-ICE takes into account the view factor between all surfaces, but the calculation tool accounts the view factor only between the ice surface and the ceiling, which is obviously the largest.

Third, a similar difference exists between IDA-ICE and the calculation tool in calculating conduction and convection heat transfer. Thus, the calculation tool simplifies the calculation by ignoring smaller factors, but IDA-ICE calculates in more detail.

Fourth, IDA-ICE applies accurate instantaneous COP for the refrigeration machineries depending on the refrigeration loads at different times, for example, resurfacing time. However, the calculation tool applies an average constant COP for the full year, which is clearly inaccurate.

2.3. Case study building

The energy consumptions in two ice rink arenas in Finland, Tapiola and Klaukkala, were measured. The temperature gradients and AHU configurations in these arenas are different. The measurements were not implemented for the whole year but only for several days. Therefore, we need to break down the yearly results of the calculation tool to obtain the results for the same measured periods. However, we can use the measured values to validate the IDA-ICE dynamic simulation model results and the calculation tool.

The energy consumption in an illustrative hypothetical ice rink arena example is calculated using the measured results of similar ice rinks, the simulation results, and the calculation tool. The example ice rink arena has dimensions of 65 x 35 x 6 m, ice pad size of 31 x 58 m, and occupants counted as 20 players and 100–500 spectators. The indoor conditions involve average temperature of +8°C, maximum relative humidity of 60%, and minimum ventilation rate of 2 $dm^3/s.m^2$–4.5 m^3/s. Finally, the outdoor air fraction is 5% of the ventilation rate, with a minimum lighting of 10 W/m^2 and usage factor of 25%. The domestic hot water consumption is 500 dm^3/m^2 a.

Fig. 4. Temperature measurements at different elevations in Tapiola Metro Arena, taken to obtain the temperature gradients.

We carried out an extensive range of measurements at two ice rink arenas in Finland, Tapiola and Klaukkala. The results have mainly been presented in earlier articles published by the authors [3,4]. In addition, the measurement and devices have been described in more detail previously [4]. The measurement results presented in the Table 3 are originally from the previous measurements, but they are energy consumption results calculated based on those measured values.

The measured data that have been used here consist of two parts: First, temperature measurement was carried out for measuring temperature stratification or the temperature gradient. This was performed by installing the sensors at different heights, as shown in Fig. 4. The measurement device used for this measurement was the "Thermadata HTD," which measures temperature and relative humidity simultaneously. The accuracy levels of the sensors were as follows:

Temperature: ±0.5°C (0…+45°C), ±1°C (−20…+70°C), ±1.5°C (+70…+85°C),

Relative Humidity: ±2%RH at +25°C (10…90%RH).

However, the humidity measurements of these sensors were not used in this paper.

The second part of the measurements used for this study are those used for computing the energy consumptions, as presented in Table 3. They were measured mainly at different sections of the AHU. The sensors were built in by the AHU manufacturer so that the data can be simply extracted from the Building Automation System by maintenance personnel. The data that were obtained from the machine and used in this study for computing the energy consumptions comprise the air flow rate, relative humidity, and temperature at different sections of the AHU. The CO_2 also has been measured, but it is not used in this paper. The accuracy and types of the sensors used in different AHU sections are as follows:

Temperature:	IMMERSION STCZ-12	± 0.3 K (typ. at 21 °C),
Air flow:	STBZ-22, STCZ-35	±1% of reading,
Relative Humidity:	EE33	0...100% RH, ±1.3% RH.

Table 3. Measurement Results Versus the Results of the Calculation Tool and Simulation

Location	**Klaukkala**			**Tapiola**		
Temp. gradient	**1.6°C/m**			**2°C/m**		
Measurement date & time	2016-05-03T10:50–2016-05-12T9:00			2016-06-15T14:50–2016-06-21T10:20		
Air handling units	AHU1		Ice pad Refrigeration power (kWh)	AHU2		Ice pad Refrigeration power (kWh)
	Heating (kWh)	Cooling (kWh)		Heating (kWh)	Cooling (kWh)	
Measurement result	8,592.8	2,907.6	13,090.3	10,695.5	4,939.8	
Simulation result	8,954	2,861	12,936	9,682.24	5,108	8,753
Calculation tool	9,538	2,918.7	13,158.4	9,610.7	5,173.5	9,226.45
	Calculated based on a 6-m height			Calculated based on a 6-m height		

As shown in Table 3, the results of different arenas are not equivalent since they have been taken in different periods of the year and with different time extents. In addition, various AHU configurations have been used in each of the arenas (AHU1 and AHU2). Therefore, the results of Tapiola with Klaukkala are not logically comparable with each other. However, the measurement results of each arena can be compared with the simulation and the tool results of the same arena to validate them.

The correlation factors are applied in the calculation tool for conditions where the steady state approach of the tool cannot accurately consider all the details taking place in the arena. The correlation factors are set so that the tool results also comply with the previously published papers by the authors [3,4].

3. Theoretical principles of the calculation

To calculate the heating and cooling demands, we need to analyze various possible thermal interactions, including the following:

a) The heat losses between the indoor and outdoor air through the building envelopes
b) The heat loads between the ice pad and surrounding environment either upward to the indoor environment or downward as heat transfer from the ground underneath to the ice
c) Internal loads
d) Loads due to ice resurfacing process

The theoretical principles for the implementation of each of the above-mentioned items are discussed in the following subsections.

3.1. Ice surface modeling

To calculate the heat exchanged between the ice surface and indoor air, we need to concentrate on the transient layer above the ice. To do so, it is initially required to determine the heat transfer coefficients of the air layer above the ice surface (as a boundary layer). Theoretical challenges are how accurately the model calculates the convection (h_{Conv}) and condensation heat transfer (h_{Cond})

coefficients. To calculate the heat transfer coefficients, it is required to first calculate the saturation pressures on the ice surface and on a height 0.1m above the ice surface, as in the following:

$$P_{in} = 10^5 \exp\left(17.391 - \frac{6142.83}{273.15 + T_{in}}\right) \quad [19], \tag{1}$$

$$P_{ice} = 10^5 \exp\left(17.391 - \frac{6142.83}{273.15 + T_s}\right) \quad [19]. \tag{2}$$

In their measurement results, Ferrantelli et al. [19] showed that relative humidity does not vary significantly within 0.005–0.1 m above the ice. A control volume over the ice surface with the height of 0.1 m is assumed to calculate temperature and relative humidity of the air inside the control volume. The same specification can be applied to calculate condensation and convection heat transfer. Therefore, relative humidity at a height of h = 0.1 m above the ice surface is calculated as

$$RH_h = \left(\frac{h}{1{,}5}\right) \times (90 - RH_{1,5}), \tag{3}$$

$$dp = \left(\frac{RH_h}{100}\right) \times (p_h - p_{ice}), \tag{4}$$

$$dp_{atm} = \left(\frac{dp_{pa}}{101325}\right). \tag{5}$$

The heat transfer coefficient for condensation is also calculated as:

$$h_{Cond} = 1750 \times h_{conv} \times \Delta P/\Delta T(RH_h/100), \quad [19] \tag{6}$$

$$q_{cond} = h_{Cond} \times (T_{in} - T_{ice}). \tag{7}$$

3.2. Airflow balance equations

The calculated and measured airflow rates, along with measured temperature and relative humidity changes over the components were used to calculate the components' theoretical energy output over the measurement periods. The heating coils' and heat exchangers' heating powers are calculated as

$$Q_{heat} = q_{air} \times \rho_{air} \times c_{air} \times \Delta T_{air}, \tag{8}$$

and the required cooling power for the cooling coil is calculated as

$$Q = q_{air} \times \rho_{air} \times \Delta h_{air}. \tag{9}$$

Here, the enthalpy of air can be expressed as

$$\varphi_{air} = c_{air} T_{air} + x_{air}(c_w T_{air} + h_{we}). \tag{10}$$

The fresh air intake of the AHU was calculated based on CO_2-level differences between the extract, supply, and fresh airs. Any decrease in CO_2 level from extract to supply air means a portion of the supply air is fresh air, since it is reasonable to assume that there are no other CO_2 sources in the unit. Fresh air intake can be calculated as

$$q_{fresh\ air} = q_{sup}\left(\frac{C_{ext} - C_{sup}}{C_{ext} - C_{fresh}}\right). \tag{11}$$

The resulting flow rate for fresh air intake serves as more of an approximation than an exact value, but its accuracy is enough to determine whether the unit operates in full mode or partial recirculation mode.

3.3. Heat loss calculations through external envelopes of the building

The thermal loss of the building is calculated using the conventional heat loss calculation equations, as follows:

$$Q_{Heat\ loss} = \{\sum H_{Building} \times \sum(\Delta T \times \Delta t)_{Cold\ months}\}, \tag{12}$$

$$\sum H_{Building} = \{\sum(U \times A)_{External\ wall} + \sum(U \times A)_{ceiling} + \sum(U \times A)_{Floor} + \sum(U \times A)_{doors\&windows}\}. \tag{13}$$

The specific thermal transmittance of the building components is used when calculating the heat loss. Thermal loss of leakage air is also calculated as

$$Q_{Leakage\ air} = [\rho_{air} \times C_P \times q_v]_{Leakage\ air} \times \sum(\Delta T \times \Delta t)_{months}. \tag{14}$$

The heat loss is calculated using weather data from three zones—Zones I, III, and IV. The calculator requests the users to specify the climatic zone in which the ice rink is located. The heat loss is then calculated based on the outdoor weather data of the selected zone. For each zone, average monthly temperatures and time extent of each month in hours are applied. Finally, those months where their average temperatures are lower than the ice hall indoor temperature are considered as cold months for the heat loss calculation. The average indoor temperature of the ice rink arena is assumed to be +10˚C:

$$q_{heat\ loss} = \{\sum(U \times A)_{External\ walls} + \sum(U \times A)_{ceiling} + \sum(U \times A)_{Floor} + \sum(U \times A)_{doors\&windows}\} \times [(\Delta T_1 \times \Delta t_1) + (\Delta T_2 \times \Delta t_2) + \ldots]_{cold\ months}. \tag{15}$$

Here, ΔT is the temperature difference between the average outdoor temperature of a cold month and the average indoor temperature, which is assumed to be +10˚C, while Δt is the total time of the cold month in hours.

The number of cold months is chosen based on the average monthly temperatures compared with the average indoor temperature of the arena (~+10˚C). If the average temperature of the month is colder than +10, then the tool assumes it is a cold month. Accordingly, Zone IV has two more cold months than other zones do. The total heating power is delivered through the heating coil of the AHU to the air supplied into the space.

3.4. Heating energy demand due to domestic hot water consumption

There are basically three main consumers for domestic hot water, which are as follows: players showering in the locker room, spectators in the washroom, and resurfacing operations (a considerably large consumer). The number of players for each match is similar in every arena, but the active times of the arenas may differ from each other. Thus, the hot water consumption for players is approximately the same unless an arena is specified to have a higher or lower activity period than usual. The number of spectators depends on the stand capacity of the arena, and thus, the user of the tool must specify the number of spectators. The spectators' consumption, which is limited to washroom usage, can be calculated based on the number of spectators. Finally, the quantity of domestic hot water used for resurfacing operations, which is the largest domestic hot water consumption, is defined based on the information provided by the user about the hot water quantity and its temperature, in addition to the sequence of weekdays and weekend resurfacing operations:

$$Q_{hot\ water} = Q_{resurf.water} + Q_{DHW}. \tag{16}$$

3.5. Heat load of the ice pad

There are various forms of heat transfer occurring between the ice pad and the surrounding environment, both upward to the room and downward to the ground. Each form of heat transfer is analyzed individually, and their theoretical concepts applied in the calculation are described in the following paragraphs.

Convection heat transfer: One of the main thermal interactions occur between the ice pad and indoor air is the convection heat transfer. The convection heat transfer is calculated using the air temperature at the height 0.1 m above the ice, assuming a linear relationship between the ice surface temperature and indoor air temperature above the ice at the height of 1.5 m as

$$T_{h\,=\,0.1} = T_s + (\frac{h = 0.1}{1.5}) \times (T_{1,5m} - T_s), \quad (17)$$

where T_s and $T_{1,5m}$ are asked for on the first page of the tool, and these values are provided as input data by the user. Then convection heat transfer coefficient at the height of 0.1 m above the ice using the indoor air velocity at the same height 0.1 m is calculated according to ASHRAE (2010) as follows [20]:

$$h_{Conv.} = 3.41 + (3.55 \times V), \quad (18)$$

where V is the indoor air velocity at a height of 0.1 m above the ice, and h_{Conv} [W/(m^2 °K)] is the heat transfer coefficient at the same height above the ice, calculated with V assumed to be 0.15 m/s. Then [19,20],

$$\dot{Q}_{Conv.} = h_{Conv} \times (T_{in} - T_S), \quad (19)$$

where q_{conv} [W/m^2] is the convection heat transfer between the ice surface and indoor air, T_S is the ice surface temperature [°K], and T_{in} is the indoor air temperature at height 0.1 m. Furthermore,

$$T_{in} = T_{h\,=\,0.1m} \text{ [°K].}$$

Radiation: To calculate the radiation heat transfer, we first need to assume that the inner surfaces, particularly the ice pad and ceiling, are opaque, diffuse, and gray surfaces. The problem is solved assuming that the ceiling and ice pad form a two-surface enclosure consisting of two opaque surfaces, at the specified surface temperatures $T_{S\,-\,Ceiling}$ and $T_{S\,-\,ice}$, with emissivity values of $\varepsilon_{Ceiling}$ and ε_{ice} and surface areas of $A_{Ceiling}$ and A_{ice}, as shown in Fig. 5.

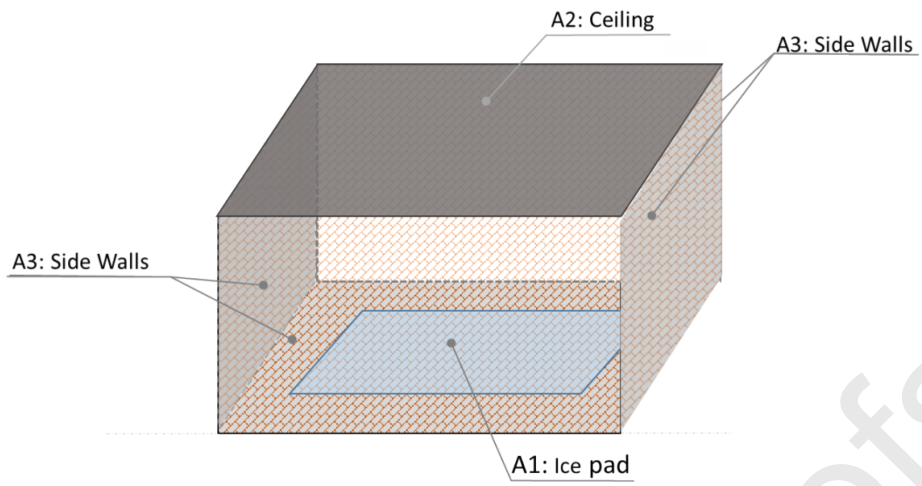

Fig. 5. Ice rink model for calculating the radiation heat transfer.

Then, the network method, which is based on the electrical network analogy, is used to calculate the net radiation heat transfer to each of the surfaces [21]. In this method, which is based on Ohm's law, the radiation equation can be written as

$$\dot{Q}_i = \frac{E_{bi} - J_i}{R_i} \qquad \& \qquad R_i = \frac{1-\varepsilon_i}{A_i \varepsilon_i},$$

where R_i is the surface resistance to radiation. The radiation heat transfer for a two-surface enclosure, its electrical network model, and the resistances are shown in Fig. 6.

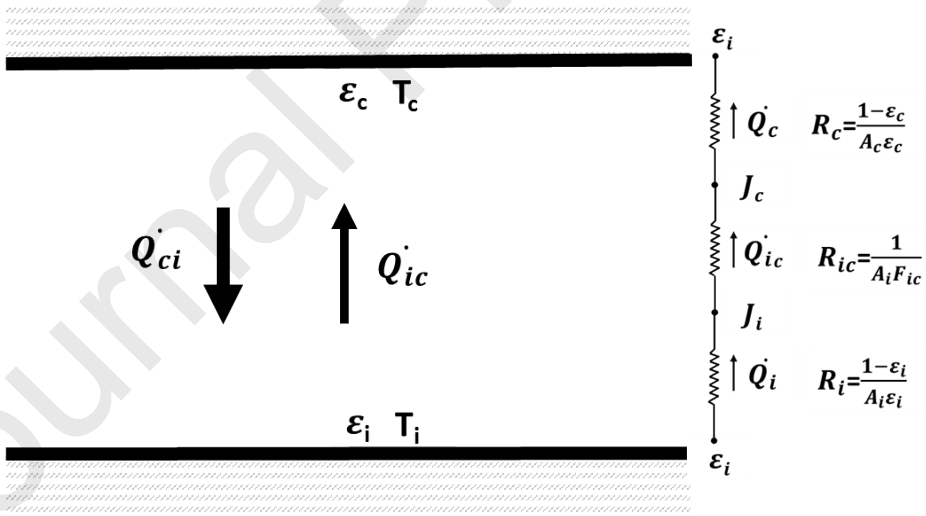

Fig. 6. Two-surface enclosure and its electrical network model.

The radiation heat transfer is calculated using radiation heat transfer coefficients. To calculate the coefficients, the emission factor between the ice and the ceiling must be calculated. The equivalent emission factor for the radiation from the ceiling to the ice surface is calculated according to ASHRAE (1994) with the following equation [19-21]:

$$\varepsilon_{12} = \varepsilon_{ci} = \left[\frac{1}{F_{ci}} + \left(\frac{1}{\varepsilon_c} - 1\right) + \frac{A_c}{A_i} \times \left(\frac{1}{\varepsilon_i} - 1\right)\right]^{-1}, \tag{20}$$

where A_C is the surface area of the ceiling (m^2), A_i is the surface area of the ice (m^2), and ε_i is the emissivity of the ice, which is equal to 0.98. In addition, $\mathcal{E}_{ci}$ is the emission factor between the ice and the ceiling, and F_{ci} is the view factor from the ceiling to the ice surface. The view factor is 0.68 to calculate the radiation heat load from the ceiling to the ice pad. However, to calculate the radiation from the ice surface to the ceiling, the view factor is assumed to be 1.

The emissivity of the ceiling, ε_C, depends on the surface coating emissivity of the ceiling. ε_C is given as the input data to the tool by the user on the first page with two options, either 0.2 or 0.8. It is restricted to these two options because they are the most common coatings, representing either a new low-emissivity coating or a conventional high-emissivity one.

The heat transfer coefficient is calculated through the following equation [20]:

$$h_{rad} = \varepsilon_{12} \times \sigma \times (T^2_{S-Ceiling} + T^2_{S-Ice}) \times (T_{S-Ceiling} + T_S), \qquad (21)$$

where h_{rad} is the radiation heat transfer coefficient (W/(m^2 °K)); σ is the Stefan–Boltzmann constant, which is equal to 5.67037E-08; $T_{S\text{-}ice}$ is the ice surface temperature (°K); and $T_{S\text{-}Ceiling}$ is the surface temperature of the ceiling (°K).

The ceiling temperature is assumed to be the same as the air temperature at the height of h = 6 m, which is available as input to the tool. Finally, the radiation heat transfer is calculated using the following equation [21]:

$$Q_{rad.} = h_{rad.} \times (T_{S-ceiling} - T_{S-ice}), \qquad (22)$$

where q_{rad} is the radiation heat flow rate in W/m^2.

The effects of ceiling emissivity make the calculation of ceiling surface temperature challenging [1]. Therefore, the ceiling surface temperature is calculated using radiation heat transfer to the ceiling as input energy and the output energy are convection heat transfer to the indoor environment and conduction and convection to the outdoors. The heat transfer analysis and the electrical resistance analogy model of that are shown in Fig. 7.

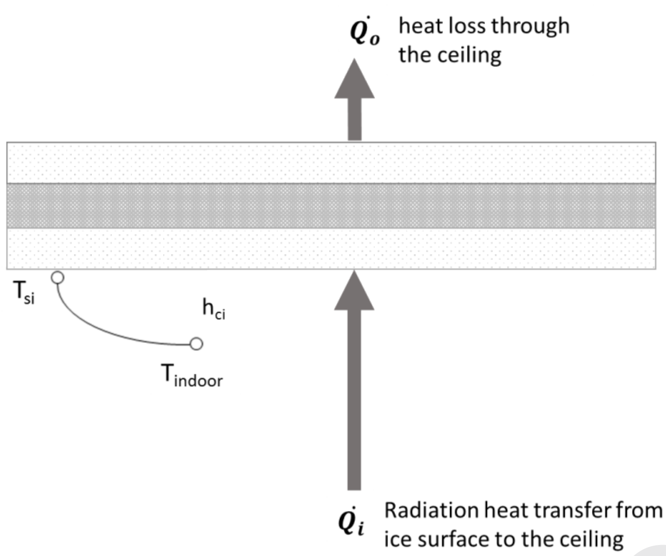

Fig. 7. Thermal analysis for ceiling surface temperature calculation.

The conduction heat transfer through the ceiling is calculated as

$$\dot{Q}_i - \dot{Q}_o = \frac{T_{si} - T_{indoor}}{R_{ci}} = \frac{T_{si} - T_{indoor}}{\frac{1}{Ah_{ci}}}. \quad [21] \tag{23}$$

The heat loss to the outdoors, $\dot{Q}_o$, is calculated using an average outdoor temperature. As the energy balance of the ceiling justifies, the radiation energy, which cools down the ceiling, finally releases its net received energy to the indoor air through convection. Here, the ceiling acts as a ceiling radiant cooling panel for the indoor air. Therefore, the convective heat transfer coefficient in such cases is written as follows [23]:

$$h_c = 2{,}13(T_{indoor} - T_{si})^{0{,}31}. \tag{24}$$

The convective heat transfer coefficient, h_c, for pure free convection in a cooled surface facing downwards or ceiling radiant cooling was found to be in the range of 5.9–6.5 W/m^{2}K [24]. Finally, the surface temperature is calculated by iterating among equations 22 and 23 [21].

As for the calculation tool, we ignored view factors, and therefore, radiation among other surfaces except the ice surface and the ceiling. Using a similar method, IDA-ICE calculates all the view factors and then the radiation values between all the surfaces.

Thermal radiation due to lighting: From the total heating emitted to the space through the lighting radiation, the radiation is absorbed by the ice surface to warm up the ice. This is subsequently calculated as a refrigeration load to be compensated for by the refrigeration machines as follows:

$$q_{lighting} = P_{lighting} \times U \times 0.62, \tag{25}$$

where $q_{lighting}$ is the total radiation heating power generated by lighting (W), $P_{lighting}$ is total lighting power (W), and U is the utilization percentage of the lighting (%).

Heat load due to condensation on the ice surface: First, we need to calculate the relative humidity at the height of 0.1 m above the ice ($RH_{h\,=\,0.1}$). The relative humidity at height h = 0.1 m is calculated by assuming a linear relationship between the relative humidity on the ice surface, which is assumed to be 100%, and the relative humidity at 1.5 m above the ice, which has to be no more than 65% according to the guidelines. This is calculated using a linear equation as follows:

$$RH_{h=0.1} = 100 - [\left(\frac{h_{Conv}}{1.5}\right) \times (100 - RH_{h=1.5})], \qquad (26)$$

where h_{conv} is the convection heat transfer coefficient at the height of 0.1 m, and $RH_{h=1.5}$ is the relative humidity at the height of 1.5 m above the ice, which is provided by the user. To calculate the condensation heat transfer coefficient (h_{cond}), we need to calculate the pressure difference between the ice surface and the air entering to the boundry layer above the ice, using the saturation pressures which are calculated as follows:

$$P_{in} = 10^5 \times \exp\left(17.391 - \frac{6142.83}{273.15 + T_{in}}\right), \qquad (27)$$

$$P_S = 10^5 \times \exp\left(17.391 - \frac{6142.83}{273.15 + T_S}\right), \qquad (28)$$

$$\Delta P = \left(\left(\frac{RH_h}{100}\right) \times p_{in}\right) - p_S \qquad [\Delta P\text{: Pa}], \qquad (29)$$

$$\Delta P(atm) = \frac{\Delta P(Pa)}{101325}, \qquad (30)$$

$$h_{condn} = 1750 \times h_{conv} \times \frac{\Delta p}{\Delta T}, \qquad (31)$$

$$h_{conv}\text{: } 0.1m\ ,\ \Delta T = T_S - T_{0.1\text{m}}, \qquad (32)$$

$$Q_{cond.} = h_{condn} \times (T_{\text{h}=0.1} - T_S). \qquad (33)$$

Heat load due to resurfacing operation: The resurfacing operation is implemented after spreading hot water on the ice surface. This includes three stages. The first is cooling down the hot water to zero degrees centigrade; the energy need for this stage is calculated as

$$Q_{water\ cooling} = m \times C_{water} \times (T_{hot\ water} - 0). \qquad (34)$$

Then, the refrigeration machine has to cool down the water to be frozen to ice. The required freezing energy is calculated as

$$Q_{water\ freezing} = m \times C_{freezing}. \qquad (35)$$

Finally, the ice must be cooled down to an extent equal to the surface temperature of the ice. The cooling energy required is then calculated as

$$Q_{ice\ Cooling} = m \times C_{ice} \times (0 - T_{S.ice}). \qquad (36)$$

After these three steps, the overall heat load due to resurfacing is calculated as

$$Q_{resurf.} = Q_{water\ cooling} + Q_{water\ freezing} + Q_{ice\ cooling}. \qquad (37)$$

Heat transfer from the ground: There is heat transfer between the ice pad and the ground beneath the ice surface due to the temperature difference. This may cause the ground to be frozen. Refrigeration machinery must compensate for cooling energy, which is transferred to the ground. At the same time, the heating system must also warm up the ground to protect it against freezing. Therefore, the controllers of the ground heating will maintain the ground temperature at a minimum temperature of +1°C to protect against ground freezing risks. Then, the heat transfer due to conduction into the ground under the ice surface per surface area is calculated as

$$Q_{Conduc.} = U_{ins.} \times (1 - T_{ice\ sur.}), \tag{38}$$

where $U_{ins.}$ is the U-value of the insulation layer under the ice pad, and $T_{S.ice}$ is the ice surface temperature.

3.6. Cooling demand of the AHU's cooling coil

The indoor environment of ice rinks is normally cold due to the cooling effects of the ice pad. However, the cooling coil is mainly applied for dehumidification purposes. There are two main sources of humidity, which make dehumidification necessary. First, there is supplied outdoor air during the warm months, when the average outdoor temperature and humidity are high. Second, there is the human presence indoors and their activity, which includes players and spectators.

Cooling demand due to dehumidification: Humidity is generally added to the indoor air via either internal sources like the human bodies of spectators or supplied outdoor air. The dehumidification of the outdoor contribution of supply air is calculated as follows:

$$Q_{AHU.\,Cooling} = \sum_{1}^{warm\ months} [(\alpha \times \dot{m}_{air} \times (\Delta\varphi) \times t_{month})], \tag{39}$$

where α is the outdoor air fraction of the supply air (assumed to be 20%), $\dot{m}_{air}$ is the mass flow of the supply air, Δh is the enthalpy difference between the average outdoor air enthalpy of the month and the enthalpy of required indoor air, and t_{month} is the total time of the month in hours. The enthalpy of the outdoor air is calculated using the weather data corresponding to the selected zone where the ice rink arena is located. This is done by calculating the average enthalpy of outdoor air and comparing it with the enthalpy of the indoor air that is to be maintained. The calculator chooses the months with higher average enthalpy than indoor enthalpy as the months in which cooling is required. The differentiated enthalpies (outdoor from indoor) for all the warm months of the selected zone are added. The external partition load is also the load due to internal and external temperature difference:

$$Q_{AHU.\,Cooling} = (Q_{External\ partition\ load} + Q_{fresh\ air\ cooling}), \tag{40}$$

$$Q_{AHU.\,Cooling} = \{(Q_{External\ partition\ load}) + (Q_{Fresh\ air} \times 1.2 \times \sum(\Delta h \times \Delta t)_{warm\ months})\} \times \frac{\text{Imperical coefficient for cooling coil position}}{1000}. \tag{41}$$

The empirical coefficient reflecting the effects of cooling coil position is extracted from the simulation presented in the authors' previous publication [3]. According to the results of the published paper, this coefficient is 2.5 when the cooling coil is located prior to the heat recovery and 1 when the cooling coil is located after the heat recovery.

Dehumidification due to the human body: The required cooling load due to human body consists of players and spectators. It is important to consider that players are engaged in high sport activity, while the spectators are mainly inactive in the sitting state. Therefore, the added humidity is calculated based on the number of occupants multiply by the Metabiolic Rate – (Met), heat or power production of human body, related to their activity. Added humidity due to the presence of human bodies is calculated as

$$\Delta X = \frac{(N_{Players} \times 400\frac{g.\,water}{h})}{weekly\ hours\ present} + \frac{(N_{Spectators} \times 80\frac{g.\,water}{h})}{weekly\ hours\ present}. \tag{42}$$

Finally, the required cooling capacity for removing the additional humidity from the indoor space is calculated by the enthalpy difference between the increased indoor air enthalpy and the acceptable indoor air enthalpy.

$$\varphi = 1{,}006T + x(2501 + 1{,}85T), \quad (43)$$

$$Q_{AHU.\,Cooling} = \alpha \times \dot{m}_{air} \times \Delta\varphi \times \Delta t \quad (44)$$

3.7. Overall refrigeration power demand

The overall required refrigeration power to be generated by refrigeration machines is a sum of the following previously calculated portions:

a) The cooling power for the ice pad is the summation of different load portions calculated in sections 3.5.1 to 3.5.6:

$$Q_{ice\ pad\ cooling\ power} = Q_{Conv.} + Q_{rad.} + Q_{lighting} + Q_{Condens.} + Q_{resurf.} + Q_{conduc.}. \quad (45)$$

b) The cooling power required in the cooling coil of the AHU $Q_{AHU\ cooling}$, calculated in section 3.6:

$$Q_{total\ refrig.\ power} = Q_{Ice\ pad\ cooling.} + Q_{AHU\ cooling}. \quad (46)$$

3.8. AHU heating demand

The major part of heating energy demand in ice rinks is due to the cooling effect of the ice surface to indoor air, and it is calculated using the summation of the previously calculated values as

$Q_{Conv.}$	Convection heat transfer,
$Q_{rad.}$	Radiation heat losses of the ice pad,
$Q_{Condens.}$	Condensation heat load,

$$Q_{cooling\ effects\ of\ ice\ pad} = Q_{Conv.} + Q_{rad.} + Q_{Condens.}. \quad (47)$$

The overall heating energy demand is the summation of the external partition heat loss, the fresh air heating demand, and the heat loss due to cooling effects of the ice pad, as expressed the following equation:

$$Q_{AHU\ heating} = Q_{External\ partition} + Q_{cooling\ effects\ of\ ice\ pad} + Q_{Fresh\ air\ heating}, \quad (48)$$

where $Q_{External\ partition}$ is the heat loss through the external envelope of the building with the outdoor air.

$$Q_{AHU\ heating} = Q_{AHU\ heating} \times (emperical\ coefficient\ for\ location\ of\ cooling\ coil). \quad (49)$$

The applied empirical coefficient is consistent with the authors' published results, depending on the AHU type; as described in the previous paper, this is either AHU1 or AHU2 [3].

3.9. Overall heating demand

The thermal radiation of lighting ($Q_{lighting}$) and resurfacing ($Q_{resurf.}$) loads are excluded because they are basically additional heating to the space, and their heating effects are to be compensated for by the refrigeration machines. The total heating power required to be delivered to the building is the summation of

$$Q_{total\ Heating} = Q_{AHU\ heating} + Q_{hot\ water} + Q_{ground\ heating}, \quad (50)$$

where the $Q_{ground\ heating}$ is the thermal conduction from the ice pad into the ground under the ice pad, which is also equal to the heating energy required to protect the ground against freezing.

3.10. COP calculation

The COP for the entire cooling system is a key factor for calculating electricity consumption. The COP of the cooling system is a function of several variables, such as pumping power, brine pumps, the chiller, the compressor, and condensers. Ferrantelli et al. [22] showed that the pumping power corresponds to approximately 7% of the total electricity consumption. Therefore, the COP of the system is not very relevant and sensitive to the volumetric flow. Instead, the brine pumps, chiller, compressor, and condensers consume the remaining 93% of the electricity. It was also shown that choosing an optimal secondary refrigerant is instead determinant to achieve a higher COP, as the COP_{sys} is particularly sensitive to the specific heat of the refrigerant. It was found that ammonia gives a higher COP than ethylene glycol does, although ethylene glycol at concentrations between 20% and 34% is preferred for other reasons. Design possibilities regarding pipe size and depth inside the concrete slab confirmed that these attributes neither enhance nor hinder the process significantly. However, increasing the number of pipes provides a more uniform temperature profile at the ice surface.

In this study and for the calculation tool, we apply an average yearly COP, which is normally obtained from the manufacturer of the refrigeration machineries at time of purchase. The COP must be provided as input data to the first page of the tool by the users. However, the IDA-ICE takes an accurate instant COP into account and applies it in the calculation.

3.11. Total electricity demand of the refrigeration plant

The overall electricity demand of the refrigeration plant is calculated using the total cooling or refrigeration load, as described earlier in detail, divided by the COP of the refrigeration machines:

Total ice rink heat load kW = total heat load of the ice per surface area x surface area /1000

$$Required\ electricity\ power\ (kW) = \left\{\frac{\text{Heat load of ice pad (kW)}}{\text{COP}} + \frac{\text{cooling power of AHU (kW)}}{COP}\right\}. \quad (51)$$

The yearly electricity consumption of the cooling and refrigeration machineries is calculated as

$$Yearly\ electricity\ consumption\ of\ refr.\ (MWh) = \frac{Total\ cooling\ power \times 24 \times 365}{1000}\ . \quad (52)$$

4. Results

In this section, we first introduce the output results of the calculation tool. We then apply the tool by calculating the annual energy consumption and costs of an example ice rink building, once excluding and then including the refrigeration process of the ice rink. The idea is to show the difference between the real and the official E-values and demonstrate how unrealistic the official E-value would seem in such process-dominated buildings. It would cause an unrealistically high classification in the energy performance certificate.

4.1. Outputs of the tool

The output data are shown on a separate page, as depicted in Fig. 8. This includes additional pie curves presenting the contribution of each part to the heating energy, electricity, and entire energy consumptions, as well as delivered and exported energy.

Energy calculation results

Heating energy consumption [MWh]	
Heat loss due to external envelopes [MWh]	**66.3**
Heat loss of the ice rink [MWh]	**145.0**
Domestic hot water consumption [MWh]	**34.9**
Hot water consumption due to ice resurfacing [MWh]	**67.8**
Ice melting [MWh]	**72.4**
Heating consumption for freezing protection of the ground underneath the ice pad [MWh]	**21.0**
Ventilation heating [MWh]	**393.7**
Total [MWh]	**589.8**
Condensation heat [MWh]	
Recovered condensation heat	280.7
Refrigeration plant's power and consumption	
Heat loss power of the ice rink [kW]	62.6
Electrical power of refrigeration machinery [kW]	30.5
Cooling energy consumption in cooling coil of the air handling unit (AHU) [MWh]	120.3
Total generated cooling energy [MWh]	668.3
Produced condensation heat [MWh]	935.6
Electricity consumption	
Electricity consumption of refrigeration machinery [MWh]	267.3
Electricity consumption of lighting [MWh]	49.8
Electricity consumption of ventilation machines [MWh]	38.3
Total [MWh]	**355.4**
Total energy consumption (purchased & sold energy)	
Electricity (purchased energy) [MWh]	355.4
District heating (purchased) [MWh]	309.1
Heating sold [MWh]	0.0
Total (electricity + district heating) [MWh]	**664.5**
Annual energy costs	
Electricity costs [€/a]	25,945.0
District heating costs [€/a]	15,453.8
Sold heating (income) [€/a]	0.0
Total [€/a]	**41,398.8**
Total energy consumption e-value [kWh/m2 a]	**193.7**

Fig. 8. Overview of the output page of the tool.

The more essential parts of the output results are heating energy consumption, utilized condensation heat, cooling energy consumption, the refrigeration plant's electricity consumption, total electricity consumption, and yearly energy costs. Each part is color coded on the output page of the tool.

4.2. Calculation example

The E-value of the previously described example is calculated with two alternatives, which are as follows:

1. As the base case scenario, the energy consumption of the refrigeration process and subsequent heating energy required to compensate for the cooling effects of the ice pad are excluded
2. The energy consumption of the refrigeration process and the heating energy required to compensate for the cooling effects of the ice pad are included in the calculation procedures

The aim is to demonstrate the differences between the realistic and official E-values for such spaces as ice rink arenas where the process dominates the entire energy consumption. The E-value results of using the calculation tool, excluding the energy consumption of the refrigeration process and heating energy required to compensate for the cooling effects of the ice pad, are shown in Table 4.

Table 4. E-Value Results, Excluding the Energy Consumption of the Refrigeration Process

Ventilation heating:	

External partitions heat loss	66,287 kWh/a, ~ 29.14 kWh/m² .a
Fresh air heating	13,140 kWh/a, 5.78 kWh/m² .a
Domestic hot water	34,900 kW/a, 15.3 kWh/m².a
Electricity:	
Lighting:	49,800 kW/a, 21.9 kWh/m² a
Ventilation fans	38,300 kWh/a, 16.84 kWh/m² a

The total heating energy and electricity consumptions are calculated by summing up the values in the table 4, as follows:

Total heating energy consumption = 29.14 + 5.78 + 15.3 = 50.22 (kWh/m².a),
Total electricity consumption = 21.9 + 16.84 = 38.74 (kWh/m².a),

Finally, E-value is calculated by substituting the total heating and total electricity consumptions in the following equation:

$$\text{E} - \text{Value: } [1.2 \text{ x (total Electricity)}] + [0.5 \text{ x (total Heating)}],$$

$$\text{E-Value: } [1.2 \text{ x } 38.74] + [0.5 \text{ x } 50.22] = 71.6 \text{ kWh/m}^2 \text{ a},$$

$$\text{E-value: } 71.6 \text{ kWh/m}^2\text{.a } = \text{“A” certification, the finest class in energy performance.}$$

The results of E-value calculation using the calculation tool and including the energy consumption of the refrigeration process, as well as the required heating energy to compensate for the cooling effects of the ice pad, are presented in Table 5.

Table 5. E-Value Results, Including the Energy Consumption of the Refrigeration Process

Refrigeration demand:	
Cooling energy demand to maintain the ice pad	548,020 kWh/a, ~ 240.9 kWh/m² a
Heating energy demand:	
External partitions heat loss	66,287 kWh/a, ~ 29.14 kWh/m² a
Ventilation heating	393,700 kWh/a, ~ 173.05 kWh/m² a
Domestic hot water	34,900 kWh/a, ~ 15.34
Resurfacing hot water	67,800 kWh/a, ~ 29.8 kWh/m²a
Electricity consumption:	
For the refrigeration plant	267,300 kWh/a, ~ 117.49 kWh/m² a
Lighting	49,800 kW/a, ~ 21.9 kW/m² a
Ventilation fans	38,300 kwh/a, ~ 16.84 kWh/m² a
Total heating energy consumption	**247.33 kWh/m² a**
Total electricity consumption	**156.23 kWh/m² a**

The E-value is then calculated by substituting the total heating and electricity consumption in the following equation:

E-Value: [1.2 x (total Electricity cons.)] + [0.5 x (total Heating cons.)],

E-Value: [1.2 x (156.23)] + [0.5 x (247.33)] = 311.14 kWh/m² a,

E-value: 311.14 kWh/m².a = “G” certification, the poorest class in energy performance.

The E-value is equal to 311.14 kWh/m² a, which means that is it higher by a factor of 4.35 times compared with the previous result, when the refrigeration process is not included in the calculation.

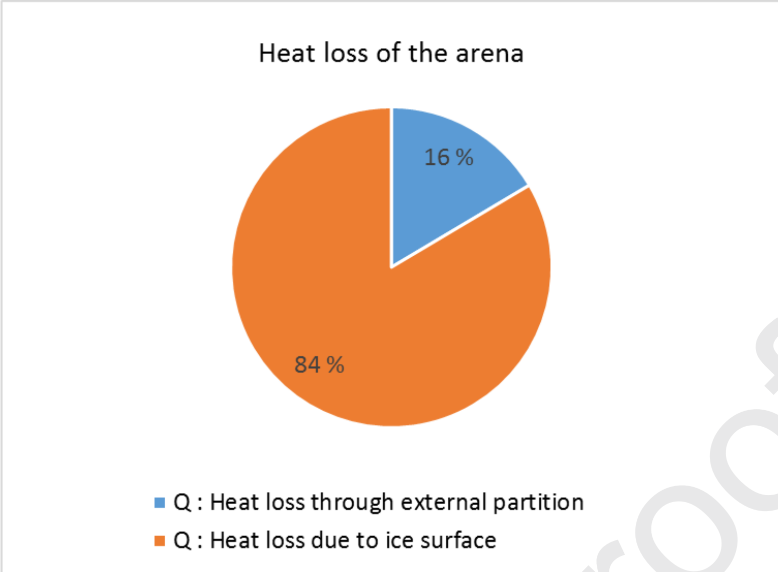

Fig. 9. Contribution of external partitions and internal loads to overall heat loss.

The official E-value considers only 16% of the required heating in the calculation, which is the heat loss through the building envelope, fresh air heating, and all other heating demands of the building except the cooling effect of the ice pad.

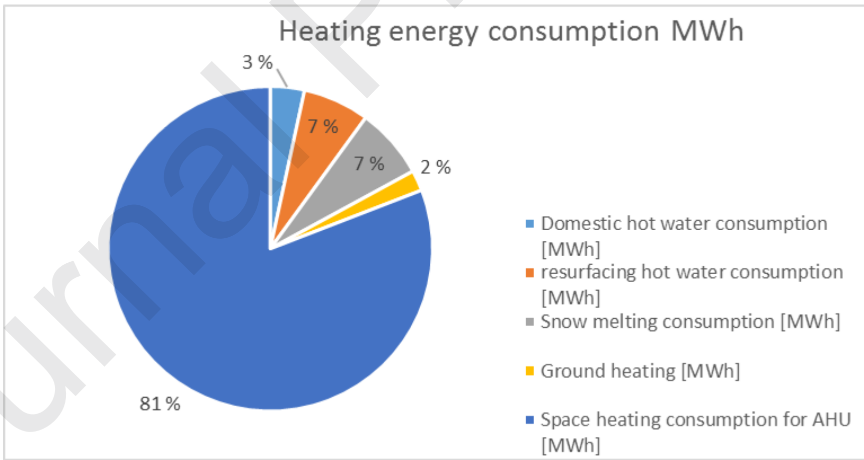

Fig. 10. Contribution of each sector to heating energy consumption.

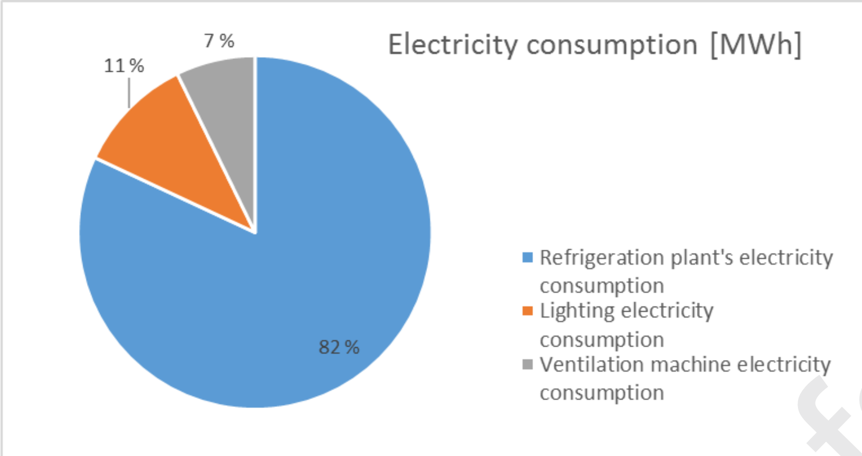

Fig. 11. Contribution of each sector to electricity consumption.

The refrigeration plant of the ice pad consumes 82% of the entire electricity consumption, as shown in Fig. 11. However, this is not accounted for when the official E-value is calculated. The official E-value represents only approximately 18% of the primary energy use. Subsequently, the major heat loss in the ice arenas is caused by the ice pad, as shown in Figs. 9 and 10. Therefore, the heat supply to compensate for the effects of the ice pad dominates the entire heating energy demand of the building. In contrast, the refrigeration plant of the ice pad is also the major consumer of electricity in the ice rinks (Fig. 11). Therefore, the energy consumed to generate cooling power for refrigeration of the ice pad and the heating energy supplied into the building due to the effect of the ice pad both play a major role in defining the real E-value.

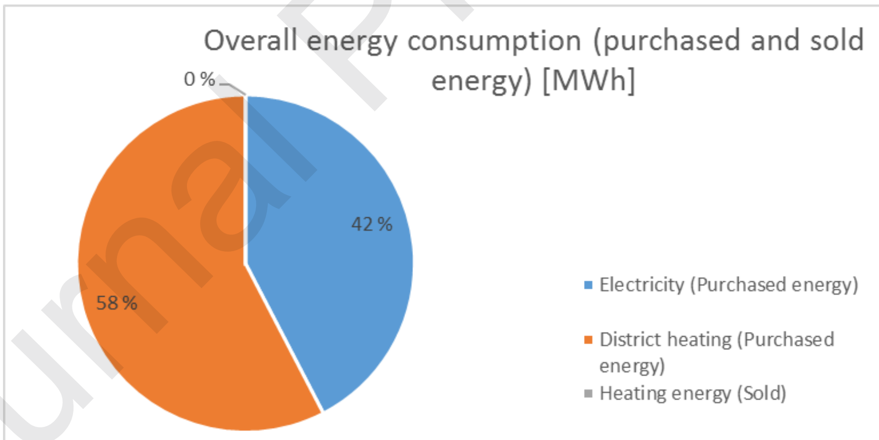

Fig. 12. Contribution of each energy form to the overall energy consumption (delivered/exported energy).

The measurement results of three ice rink arenas in Finland with various vertical temperature gradients has been presented in a previous article by the authors [3]. The total heating and cooling energy demand in two ice rink arenas with various AHU configurations were measured. The heat exchanger, cooling coil energy demands, and indoor temperature gradients were independently studied to highlight the significance of the AHU configurations, as well as the effects of temperature gradients on heating and cooling energy demands. The measurement results of the previous study are applied as required measurement data in this study. However, the simulation models can also be applied to verify the calculation tool results when changing temperature gradients, heat recovery positions, and other parameters.

4.3. Parametric analysis

At this stage, the IDA-ICE simulation results of the refrigeration, cooling, and heating energy demands are compared with the results of the calculation tool while changing one parameter at a time in both the simulation model and the calculation tool. However, altering parameters, such as indoor air temperatures at a specific height, in the IDA-ICE model and keeping them constant during the whole simulation period is not as easy as it is in the calculation tool.

Indoor temperature and temperature gradient variations: We start by altering indoor temperatures by assuming a constant temperature gradient. It is easy to assume that the indoor air temperature is constant at certain heights in the calculation tool. This parameter can be easily set as constant in the tool, either at a height 1.5 m above the ice surface or under the ceiling. However, in the IDA-ICE simulation model, it does not remain constant. In fact, IDA-ICE behaves more like reality, while the constant temperature assumption in the calculation tool is unrealistic, and this is the main source of inaccuracy.

A detailed image from the IDA-ICE simulation results indicates how indoor temperatures at every specific height fluctuate over time. Figure 13 shows indoor air temperatures at the heights above the ice surface and under the ceiling.

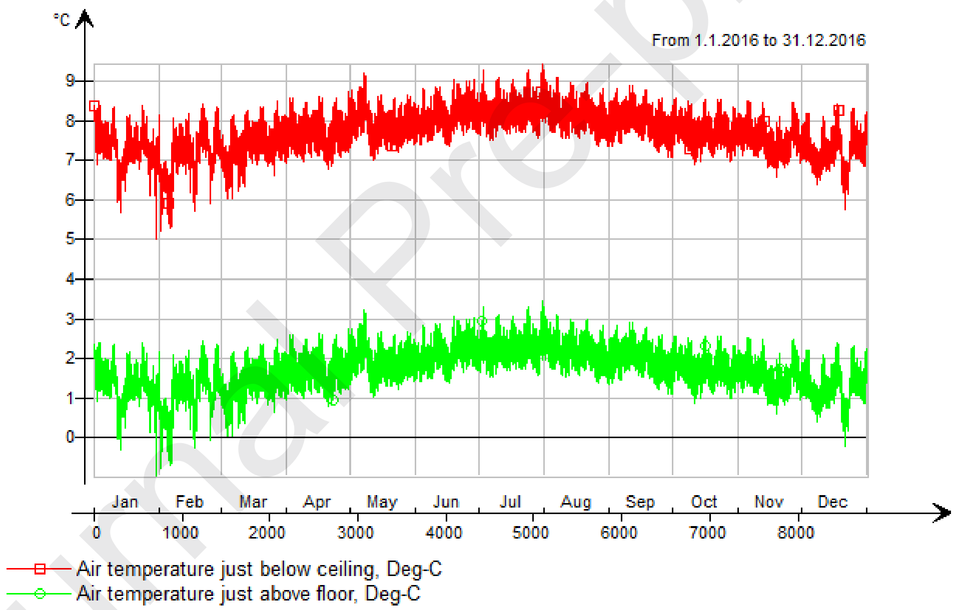

Fig. 13. Indoor air temperature variations.

As the first step in the parametric analysis, we set the temperature gradient to 2°C/m and alter the indoor temperatures to 5°C and 14°C, 4°C and 13°C, and 3°C and 12°C, where the first value is the indoor air temperature at a 1.5-m height above the ice and the second value is the indoor air temperature at 6 m height above the ice (under the ceiling). The second value is in fact the correspondent temperature to the temperature gradient. In the next step, we change the temperature gradient to 1.5°C/m and repeat the calculations by altering the indoor temperatures. Finally, temperature gradient of 1.0°C/m are applied with similar indoor temperature alternatives. The results are presented in Figs. 14, 15, and 16, showing the AHU heating, AHU cooling, and refrigeration demand results of IDA-ICE compared with the calculation tool.

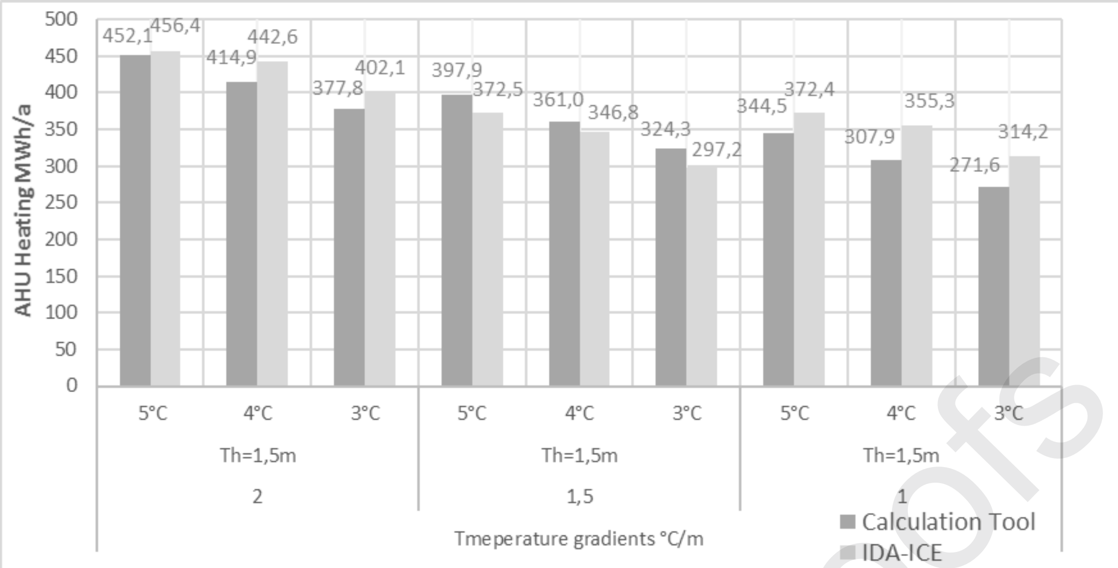

Fig. 14. AHU heating demand results of IDA-ICE compared with the calculation tool results (MWh/a).

As the results show, the heating demands are mostly less than 10% different than the IDA-ICE results. In a few cases, the deviations are higher, at up to 15% compared with the IDA-ICE. However, in most cases, the results of the calculation tool seem more reasonable and stable. The reason is that, there are complicated control settings in IDA-ICE to make the temperature levels and surface temperatures similar to those of the calculation tool. In most cases, this affects other parameters, such as the supply air flow rate or fresh air fraction, changing them as well. The same occurs when we change the temperature gradient. To make the results comparable to the calculation tool, we had to continue adjusting the controller repeatedly when the condition in IDA-ICE became closer so that the comparison would be logically possible. Therefore, a lot of trial and error was required to make the indoor condition in IDA-ICE close enough to the calculation tool. It was generally more difficult to maintain lower temperatures with either of the temperature gradients in IDA-ICE, such as 3°C, at 1.5 m above the ice surface, making the controllers unstable and the IDA-ICE results less trustworthy.

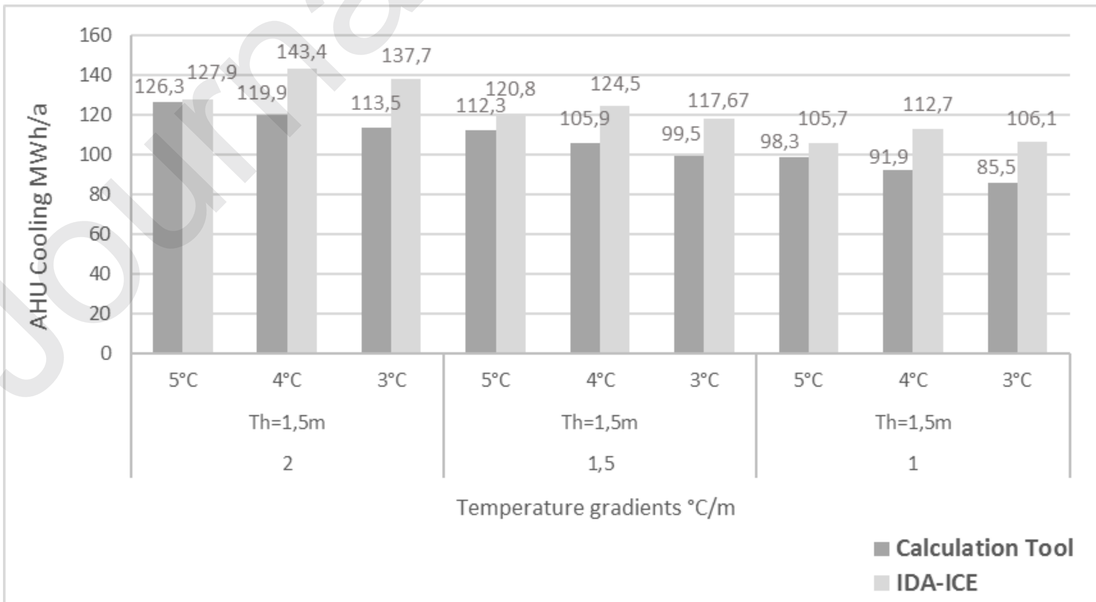

Fig. 15 AHU cooling demand results of IDA-ICE compared with the calculation tool results (MWh/a).

Figure 15 represents the AHU cooling demand results of IDA-ICE compared with those of the calculation tool. There are generally larger deviations in the AHU cooling results compared with the AHU heating. The deviations in AHU cooling results are up to 25% compared with the results of the calculation tool. Figure 16 represents the refrigeration demand results of IDA-ICE compared with the calculation tool. As expected, the deviation in the refrigeration loads were generally lower, ranging from few percentage points up to approximately 10%. As stated above, the reasons for the deviations mostly involved difficulties setting similar conditions in IDA-ICE, which complicated the situation and created more instabilities.

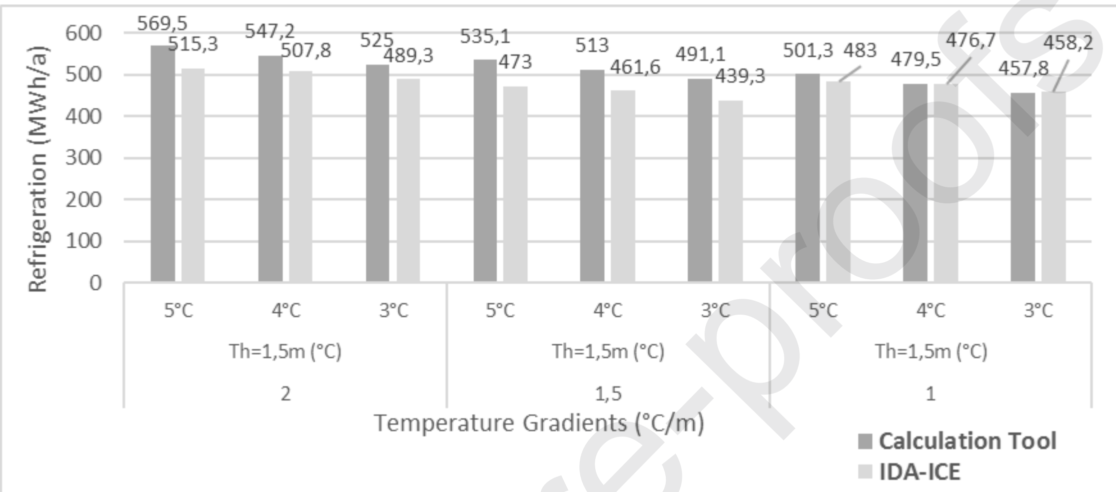

Fig. 16. Comparison of refrigeration demand of IDA-ICE with the calculation tool (MWh/a).

Figure 17 illustrates the final results of the total costs calculated by IDA-ICE compared with the calculation tool. The discrepancies in total costs calculated by the tool compared with the IDA-ICE cost results are smooth. Most importantly, the total cost results remain within a few percentage points up to maximum discrepancy of around 10%. This can be an acceptable deviation range, particularly if we note that this is a cost estimation result provided in an early stage using primary information of the project.

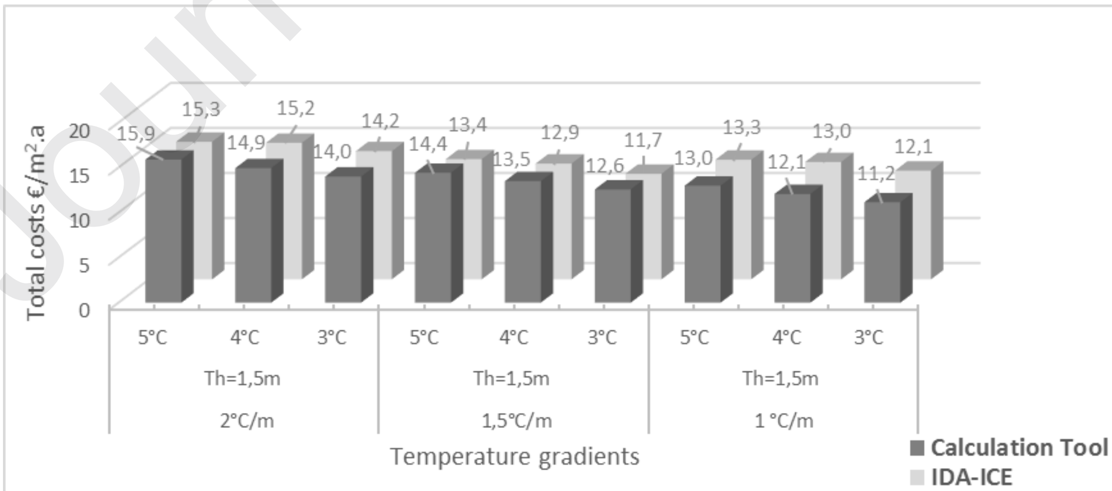

Fig. 17. Total costs calculated by the tool compared with IDA-ICE (€/m².a).

Changing outdoor air fraction: The minimum fresh air fraction in the IDA-ICE model was first set to zero, and the simulation was run. The output results showed that the fresh air fraction was forced to become 6.7% (on average) by the demand response control system. This means that the minimum fresh air fraction in IDA-ICE cannot be lower than 6.7% due to the demand control ventilation based on CO_2 levels. In other words, the controllers, which control CO_2 according to the number of occupants and their requirements, do not allow the fresh air fraction to be lower than 6.7%. Therefore, instead of starting parametric analysis from zero as the minimum fresh air fraction, we started from 6.7% for both the calculator and IDA-ICE. In the next steps, we set the fresh air contribution to 10%, 15%, 20%, 25%, and 30%. Figure 18 shows the AHU heating, AHU cooling, and refrigeration results with each of the fresh air fraction's alternatives. The results show that the heating, cooling, and refrigeration results of the calculation tool are within 0 up to 7.5% deviation from the IDA-ICE results. The total costs calculated by the tool compared with the costs calculated by IDA-ICE are shown in Fig. 19. The cost results of the calculation tool are approximately within 3 to 4.5% deviation from the IDA-ICE results, which are desirable results for the calculation tool.

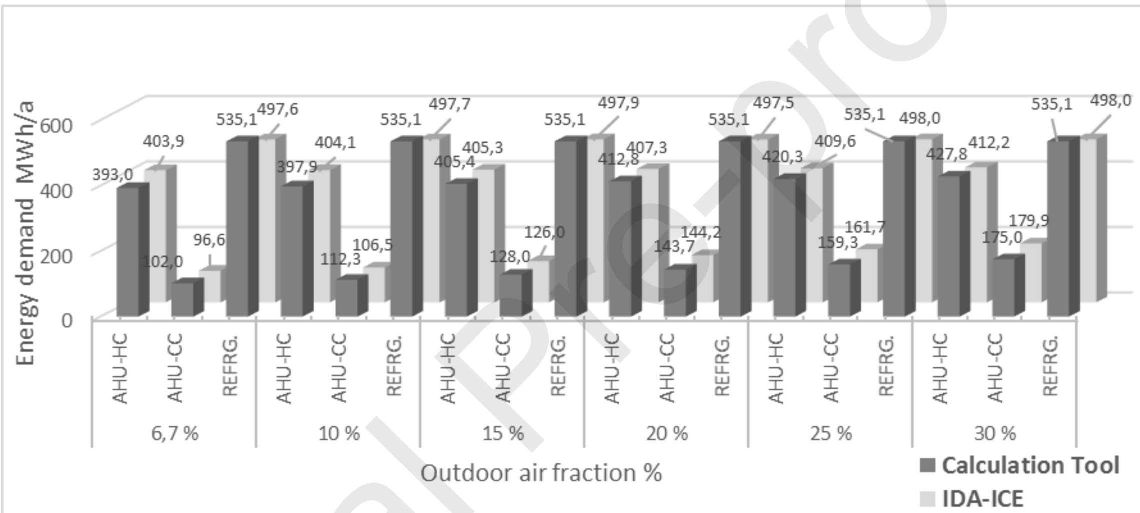

Fig. 18. Outdoor air fraction effects on energy consumption calculated by the tool compared to IDA-ICE (MWh/a).

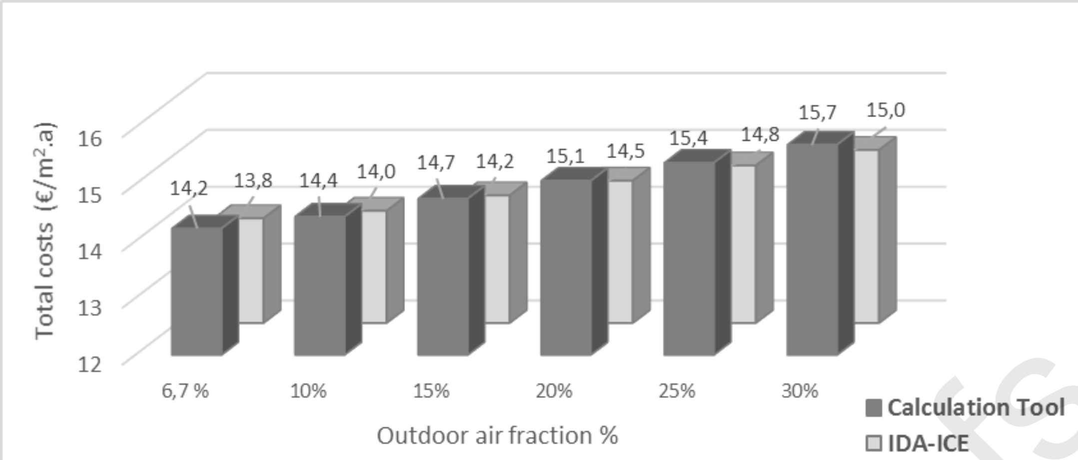

Fig. 19. Outdoor air fraction effects on total costs calculated by the tool compared to IDA-ICE (€/m2.a).

Cooling coil location: In this stage, we compare the AHU heating, AHU cooling, and refrigeration results using different AHU configurations—AHU1 or AHU2 [3,4]. AHU1 and AHU2 are similar except at the cooling coil location. AHU1 is the AHU in which the cooling coil is located before heat recovery, while in AHU2, the cooling coil located after heat recovery. We employed these two AHU configurations while altering temperature gradients as 2°C/m, 1.5°C/m, and 1°C/m. The total cost results of all the alternatives are shown in Fig. 20. The results show that the discrepancies are within about 0% up to approximately 8% compared to the results of IDA-ICE.

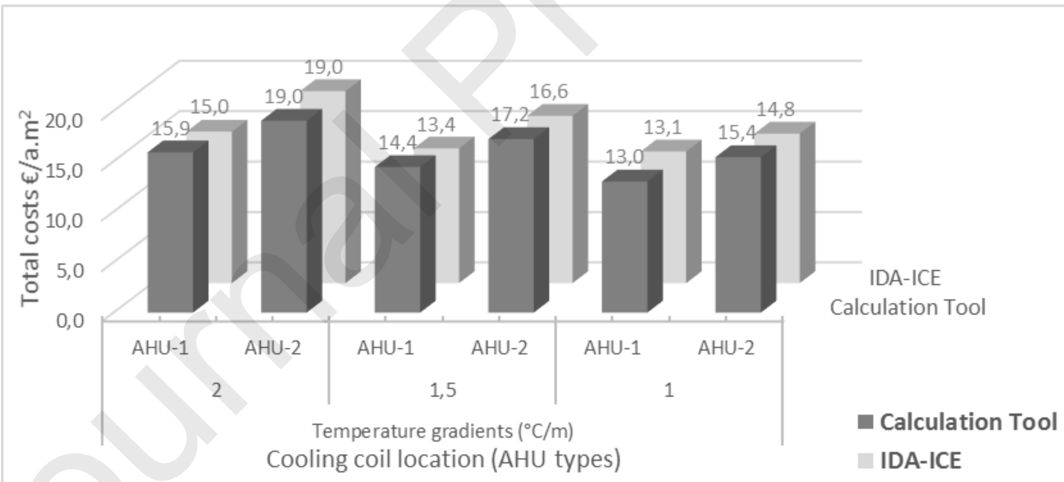

Fig. 20. Cooling coil location effects on total costs calculated by the tool compared with IDA-ICE (€/m².a).

Ceiling emissivity: At this stage, we alter the ceiling emissivity via two options, low (e = 0.2) and high (0.8). The reason for having only two options is because these are the available options in the calculation tool. The results show that the deviation between the costs by the calculation tool compared with IDA-ICE is within approximately 4–8%.

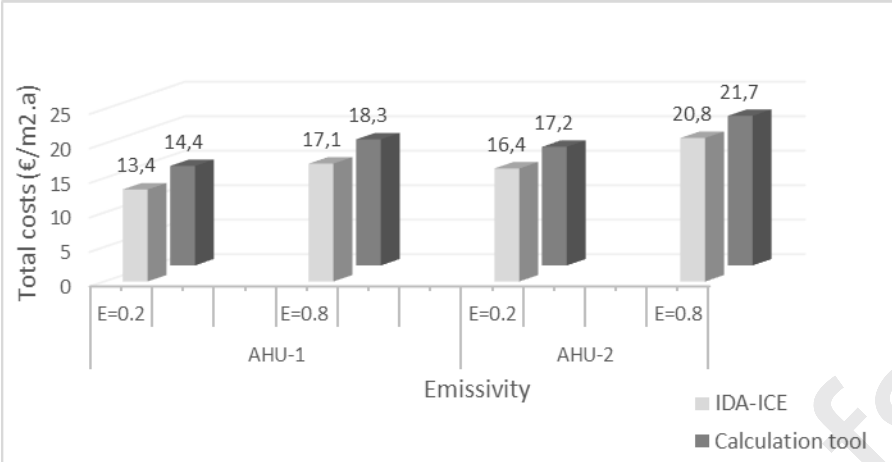

Fig. 21. Emissivity effects on total costs calculated by the tool compared with IDA-ICE (€/m².a).

In the IDA-ICE model, the controller connected to the ice surface. Therefore, the Proportional Integral (PI) controller controls the ice surface temperature according to a set value. However, IDA-ICE outputs show that the surface temperature does not remain constant. Rather, it slightly fluctuates above the set temperature.

5. Discussion

To correctly develop the calculation tool and its algorithm, it is important to have the right understanding of thermal interactions occurring in the zone. One of the main challenges that the authors faced was how to determine real surface temperatures while calculating radiation heat transfer between the ice and the ceiling surfaces. It was noted that the ice surface temperature does not vary significantly, as it may rise from half a degree (in the case of low ceiling emissivity) to a maximum of 2 degrees (in the case of high ceiling emissivity). However, the ceiling surface temperature may have several degree variations (mainly lower than adjacent indoor air temperature under the ceiling), particularly when the ceiling emissivity is high. This makes the ceiling temperature a very important parameter for calculating the refrigeration load due to radiation with the ice pad. Therefore, any wrong estimation of ceiling surface temperature may cause considerable faults in the results.

The IDA-ICE results of surface heat flux show that heat exchange interactions between the ceiling, floor and walls with the indoor air represent a bi-directional, time-varying process. This is because of the indoor air and surface temperature fluctuations, which interact in two directions. Ultimately, this means that the indoor air sometimes heats up and sometimes cools down the adjacent surfaces, depending on instant surface temperatures and the difference with adjacent indoor air. The surface temperature variations also cause inaccuracies, and therefore, discrepancies in the radiation results of the tool compared with the IDA-ICE results, as the tool assumes constant indoor air and surface temperatures throughout the year. Therefore, the results of IDA-ICE should be more reliable compared with the calculation tool.

When the temperature gradient changes, the fresh air fraction and total air flow are changed in IDA-ICE. Therefore, to have similar conditions for the comparisons, the control settings in IDA-ICE needed to be adjusted several times. However, despite trying many times to create similar conditions as in the calculation tool by adjusting the controllers, they never became the same; for example, the temperatures at different elevations, fresh air fractions, or surface temperatures were different between the two models.

It is important to note that the calculation tool cannot realistically consider all the AHU cooling/dehumidification demands; this is due to instant variations in the indoor air temperature, humidity and CO_2 level over the year. This is mainly the reason for the larger deviations in the AHU cooling demand results of calculation tool compared with IDA-ICE. The variations mostly cause extra cooling demand for dehumidification to control the humidity levels. They may also demand extra heating or cooling for the fresh air to control the CO_2 level. Such extra heating and cooling of supply air due to indoor air temperature fluctuations, sometimes even contradicting each other's effects, occur in reality and in IDA-ICE. However, this is not principally considered in the steady state approach of the calculation tool. Therefore, a correlation factor must be applied mainly for the AHU cooling to roughly cover all such loads that are unconsidered by the calculation tool. This is still an important source of inaccuracy for the AHU cooling load calculation by the tool. Therefore, a coefficient was implemented to correlate all the neglected instant loads due to demand response ventilation control based on CO_2 and relative humidity.

6. Conclusions

As the refrigeration process and space heating are the largest energy consumers in ice rinks, their contribution to energy usage dominate the total energy consumption; this was shown in different parts of this paper. In other words, the overall energy usage is a weak function of the ambient air temperature. Therefore, a simplified calculation tool can analyze ice rinks in the steady state to calculate the energy consumption and its costs. The results of this study verified that the simplified tool can estimate energy demands and costs with a reasonable range of accuracy. The parametric analysis showed that overall energy costs calculated by the tool deviate up to 8% from the IDA-ICE dynamic simulation results, while the discrepancy of refrigeration and AHU heating energy demands calculated by the tool compared with the IDA-ICE results were up to 10%. The AHU cooling demand results of the tool were calculated with a larger discrepancy, which reached up to 30% compared with the IDA-ICE results. This large inaccuracy can be explained by the instant variations in indoor air conditions, which are ignored by the steady state analysis of the calculation tool. The demand response control actions to control humidity and CO_2 levels in the occupied zone, which are influenced by instant outdoor and occupancy conditions, cannot be considered by the steady-state approach of the calculation tool.

The results verify that the tool can be practically applied with reasonable accuracy in preliminary design phases for evaluating energy consumption and its costs. There is a large difference between the quantity of the real and the official primary energy (E-value) factors. This is due to the high energy consumption of refrigeration processes in such buildings. This is a substantial point for investors to be aware of, particularly when they apply for energy performance certificates.

The calculation tool considers all the main factors that need to be taken into account in calculating energy consumption, such as the indoor air temperature at different elevations (the so-called temperature gradient), resurfacing times per day/week, players' and spectators' presence, surface emissivity, different AHU configurations, and utilizing condensation heat. The main feature of the calculation tool is its simplicity, so that every non-expert individual can apply it easily. The only thing non-expert users need to obtain is some basic understanding about technical terms and their meanings, such as temperature gradient or condensation heat.

The applicability of the calculation tool is limited to relatively cold and dry climates with short and mild summer, similar to northern hemisphere latitudes of 50° and higher. In warm and humid climates with a long summer season, latitude 30° and lower, the effect of outdoor weather would increase, and the argument about process domination would not be true. Therefore, the tool results for those areas would not be valid. Another limitation is the emissivity factor of the coating. Although this factor can theoretically be any value from 0 to 1, the options were limited by the developers of the tool to either low (0.2) or high (0.8) options. This can decrease the accuracy of the results if the user's choice is different, but he/she is forced to choose the option that is closer to his/her desired value. Lastly, a similar approach can principally be extended to other process-dominated premises, such as swimming pools.

Acknowledgments

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Author Statement

The work initiated by co-authors Lauri Leppä and Sander Toomla by creating the preliminary version of calculation tool, measurements, and preliminary version of the computer model. Then they left unfinished in the early stage. The correspondent author, Mehdi Taebnia continued to carry out the literature review, analyzing heat transfer and thermodynamic principals of the process and developing the calculation tool based on that, implementing computer simulations, analyses of the simulation results, extracting measured data and writing of this manuscript, responding to comments of the editors and reviewers and editing the manuscript. The co-author Professor Jarek Kurnitski contributed to the development of the research idea, selection of the research methodology, provided supervision and revision for this research article.

Declaration of interests

☒ The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: