Transient Energy Flux Analysis in an Indoor Ice Rink Using a CFD Approach

O. Bellache[a,*], M. Ouzzane[a], R. Sunyé[a] and N. Galanis[b]

[a] *CANMET Energy Technology Centre, Natural Resources Canada, Varennes QC Canada J3X 1S6*
[b] *THERMAUS, Faculté de génie, Université de Sherbrooke, Sherbrooke QC Canada J1K 2R1*

Abstract

An innovative method for energy flux calculations in an indoor ice rink has been developed and tested by comparing its predictions with measured values in a Montreal (Canada) municipal ice rink and a reduced-scale test bench. Based on a CFD simulation of the velocity, temperature and humidity fields it was used to predict the daily heat flux profiles into the ice by convection, radiation, condensation and resurfacing operations for a representative configuration and average steady periodic meteorological conditions. The heat loads from the ground and dissipation in the floor pipes were also calculated. The effects of climate and some design parameters (ceiling emissivity, below floor insulation thickness and sub-floor heating) and operating conditions (average brine temperature, operating schedule and ice thickness) on the daily loads were evaluated and analyzed.

Keywords: refrigeration loads, numerical prediction, measurements, CFD, ice rink

Nomenclature

A	Area (m^2)
C	Absolute humidity (kg water/kg dry air)
$c_{\varepsilon 1}$	Turbulent model constant
$c_{\varepsilon 2}$	Turbulent model constant
$c_{\varepsilon 3}$	Turbulent model constant
c_μ	Turbulent model constant
c_d	Turbulent model constant
C_p	Specific heat (kJ/kg .K)
G	Stress production rate of k
G_B	Buoyancy production rate of k
g	Gravitational acceleration (m/s^2)
h_o	Convective heat transfer coefficient (W/m^2. K)
h_{ig}	Specific enthalpy of sublimation (J/kg)
H_{sys}	Hydraulic head (m)
k	Turbulent kinetic energy (m^2/s^2)
$\dot{m}$	Mass flow rate (kg/s)
M	Mass (kg)
P	Static pressure (N/m^2)
Pr	Molecular Prandtl number
Pr_t	Turbulent Prandtl number

q_{cd}	Conductive heat flow rate (W)
q_{cv}	Convective heat flow rate (W)
q_ℓ	Heat flow rate due to condensation (W)
q_r	Radiant heat flow rate (W)
q_s	Heat flow rate due to resurfacing (W)
q_p	Heat flow rate due to pumping (W)
Q	Volume flow (L/s)
S_ϕ	Source term in equation 1
T	Temperature (°C)
T_s	External wall surface (°C)
t	Time (s)
$\vec{V}$	Velocity vector (m/s)
U, V	Components of mean velocity (m/s)
V_f	Volume of flood water (m^3)
X, Y	Cartesian coordinates (m)

Greek letters

β	Thermal expansion coefficient (K^{-1})
β^*	Coefficient of mass expansion
Γ_ϕ	Diffusion coefficient in equation 1
ΔT	Temperature difference (K)
ε	Dissipation rate of turbulent kinetic energy (m^2/s^3)
ϕ	Field variable in equation 1
μ	Fluid viscosity (N.s/m^2)
μ_t	turbulent viscosity
ρ	Density (kg/m^3)
σ_k	Diffusion Prandtl number for turbulence energy
σ_ε	Diffusion Prandtl number for dissipation rate

Subscripts

b	brine
c	ceiling
f	flood water
i	surface element
in	inlet
ou	outlet
p	pump
w	walls

1. Introduction

Thousands of indoor ice rinks operating in North America are used for spectator events such as ice sports (hockey, curling, figure skating, speed skating) and for ice shows. Although they have different sizes and configurations, all ice rinks have some common features: large dimensions; significant refrigeration load; simultaneous need for heating; refrigeration and ventilation; high energy consumption; and, considerable emissions of greenhouse gases (GHG) due to this high energy consumption and to the loss of synthetic refrigerants. The indoor air conditions (air temperature and humidity) and the motion of the ventilation air through these large buildings often lead to uncomfortable thermal conditions for the spectators, fog formation, an increase of the refrigeration load and the deterioration of ice conditions.

Lavoie et al. [1] established that, in Québec, ice rinks offer the potential for annual energy savings of 1300 GWh and for annual reductions of GHG emissions of about 0.55 Mtons of equivalent CO_2. According to that study, the current performance of ice rinks is poor due to the lack of new and improved design and operation guidelines that take into consideration the interaction between the ice-making, heating and ventilation systems.

A major difficulty in modeling the phenomena occurring in ice rinks is due to their complexity. The model must take into account heat transfer through the envelope and heat gain from the ground, air motion, vapour diffusion and condensation on the ice sheet, heat transfer by radiation between all internal surfaces, heat transfer by convection between the indoor air and all internal surfaces, sensible and latent heat gains from the spectators, conduction in the ice and floor as well as heat generation by the lights, the resurfacing operations and the secondary coolant pump work. A second difficulty in modeling the interrelated phenomena occurring in ice rinks is due to their size, irregular form and lack of internal partitions. These buildings can not be treated with the one-air-point model which assumes that the temperature is uniform because of the existence of important temperature gradients, especially in the vertical direction, and the associated buoyancy induced air movement. Furthermore, this model requires values of the heat and mass transfer coefficients, which are not easy to specify accurately, especially for zones which are not separated by solid partitions. The use of a Computational Fluid Dynamics (CFD) code [2] for the simultaneous calculation of all unknown variables eliminates the uncertainty resulting from the assumptions necessary for the application of zonal methods. Such simulations provide more detailed descriptions of the flow field. Specifically, they determine the spatial distribution of air velocity, air temperature and absolute humidity over the ice sheet. They also provide the spatial distribution of the ice, wall and ceiling temperatures as well as the corresponding distributions of the heat fluxes into the ice sheet and into the secondary coolant.

Nielsen et al [3] and Davidson [4] were among the first to apply CFD methods for airflow studies in ventilated rooms. However, they ignored the interaction between the indoor and outdoor environments and neglected heat transfer by radiation between the interior surfaces of the room. Chen & Van der Kooi [5] proposed a methodology using a

CFD code to calculate time independent airflow patterns and a separate load calculation code, which solved the algebraic room energy balance equation using transient temperature differences. This approach, which used two computer programs, is not economical for hour-by-hour air conditioning load calculations. Li [6] extended the application of CFD codes in the field of HVAC by including heat conduction through the walls and radiation transfer between their internal surfaces. This formulation was then applied by Li et al. [7] to study the radiative effects on flow and temperature fields in a room ventilated by displacement. Yuan et al [8] and Yang et al. [9] used a commercial CFD code to study ventilation and air quality in gymnasia and skating rinks respectively. Bellache et al. [10, 11] used the same commercial CFD code to analyze two-dimensional steady-state simultaneous heat and mass transfer in ice rinks. These studies have shown that the influence of the warm ventilation air on the thermal comfort of spectators and on the refrigeration load is significant. Later, Bellache et al [12] extended this CFD model to account for transient conditions. They used it to calculate the evolution of the ice rink thermal loads (conductive, convective, radiative and latent) over a 24 hour period for different locations.

In the present paper, a transient model which uses a single computer program to calculate the daily profiles of all the refrigeration load components reaching the secondary coolant of an ice rink is described. A major simplification which consists in solving the transient equations of energy and humidity conservation, using a CFD computed average hydrodynamic field as suggested by Chen & Van der Kooi [5], is described and justified. Next, comparisons of its predictions with measured data from a reduced-scale ice sheet test bench and a full-scale ice rink are presented. Finally, this model is used to evaluate the effect of several design and operating parameters on the components of the refrigeration load transmitted to the secondary refrigerant circulating under the ice.

2. Modeling of the problem

A typical, ice rink, 64 m long with average spectator capacity owned and operated by the city of Montreal (Canada) was selected for the present study. Fig. 1 shows a schematic representation of its cross-section (X-Y plane) parallel to the north and south walls at Z = 32 m (centre ice position). Seven rows of stands run the whole length of the building on one side and a 1.5 m walkway encircles the ice surface. Dasher boards, extended vertically with protective transparent barriers, are used to separate the corridor and stands from the ice surface. Infrared heaters provide spot heating of the players' and scorekeepers' benches which are situated on the west side of the building symmetrically with respect to the plane at Z = 32 m. The spectator area infrared heaters are distributed regularly over the length of the building. They are activated by motion sensors and deactivated by a thermostatic control when the air temperature over the stands reaches 15 °C. The ice rink has three different ventilation systems: (1) a 100% exhaust/make-up system for the locker rooms under the stands, (2) an intermittent system for the stands that is activated for about 15 minutes at every ice resurfacing operation and (3) exhaust fans over the ice sheet interlocked with the infrared heater controls.

A 24-hour operating schedule, which specifies occupancy, lighting, resurfacing as well as possible stoppages of the refrigeration and heating operations during unoccupied periods (e.g. between midnight and approximately 4 a.m. depending on the next morning activities) described in [12] contributes to the transient character of the conditions in the building. Calcium chloride brine is used as the secondary coolant to maintain the ice surface temperature at the desired value. It follows from this description that the building and its heating, cooling and ventilation systems are symmetrical relative to the plane at Z = 32 m. Assuming spectators are uniformly distributed throughout the length of the stands and the differences of outdoor conditions due to solar radiation and wind direction are negligible, it follows that, at the Z = 32 m plane, there will be no velocity components and no appreciable heat or mass transfer in the Z direction. The velocity, temperature and humidity distributions in this plane are therefore functions of X, Y and time and can be calculated from the corresponding transient, two-dimensional conservation equations.

The airflow within the building is turbulent, as evidenced by all the studies mentioned previously. Numerous turbulence models exist and several have been tested during these earlier works. The detailed results of these tests have been presented elsewhere [13]. The principal conclusion is that the standard k-ε model [14] is very stable and gives satisfactory predictions compared to experimental and numerical results from the literature. Chen [15] has also compared the performance of five turbulence models in predicting natural convection, forced convection and mixed convection in rooms. He used experimental data for validation and concluded that the performance of the standard k-ε model is good. Therefore this model was selected for the present investigation.

The partial differential equations modeling the air movement, heat transfer and vapour diffusion for transient conditions can be written in the following general form

$$\frac{\partial(\rho\phi)}{\partial t} + \text{div}(\rho\bar{V}\phi - \Gamma_\phi\nabla\phi) = S_\phi \tag{1}$$

The appropriate values or expressions for ϕ, Γ_ϕ, and S_ϕ in each of the seven conservation equations in the region occupied by air are defined in Table 1. It should be noted that, in this region, these seven partial differential equations are elliptic, nonlinear and coupled, since the air density in the gravity force expression varies linearly with both the temperature and the absolute humidity:

$$\rho = \rho_o\left[1 + \beta(T - T_o) + \beta^*(C - C_o)\right] \tag{2}$$

On the other hand, within the solids in the calculation domain (ceiling, walls, protective barriers, ice, concrete floor, insulation and ground) the velocity is zero. For the purpose of this study it is also assumed that these solids are impermeable to vapour and do not include any heat sources. Therefore their temperature is the only unknown quantity and can be calculated from the two-dimensional transient conduction equation.

The external surfaces of the building are subject to variable, site-dependant meteorological conditions. Since the objective of this study is to obtain "typical" daily profiles of refrigeration load components, rather than corresponding maximum values, it was decided to use average meteorological conditions except for the model validation. The hourly meteorological data provided by the U.S. Department of Energy [16] was therefore used to calculate average daily profiles of sol-air temperatures [17] for the roof, east and west walls for winter (December-February), spring (March-May), summer (June-August) and autumn (September-November). In an earlier paper [12] we have presented the meteorological data for Montreal as well as for three North American sites with very different climatic conditions (Edmonton, Houston and Pittsburgh). The data used for the calculations was based on the average daily dry bulb and horizontal sol-air temperature.

By combining radiation and convection fluxes in a single term, the energy balance for any element of the external surfaces is:

$$q_{cd,o} = h_o A \left(T_{sol-air} - T_s \right) \tag{3}$$

The dry bulb temperature and humidity of the ventilation air entering the building when resurfacing takes place are the corresponding outdoor values, evaluated as explained in the previous paragraph. The horizontal and vertical components of its velocity at the supply diffusers are both equal to 0.27 m/s. The relative pressure at the return air grilles is assumed to be zero.

The ground at a depth of 2 m is assumed to have a constant temperature. Heat fluxes across the underground surfaces of the computational domain are evaluated as proposed in [17].

On the inside surfaces of the ice-rink (walls, ceiling, stands and ice) the velocity components, concentration gradient and turbulent kinetic energy are equal to zero. The corresponding temperature is related to that of the air within the flow domain by an energy balance which takes into account conduction through the solid, convection to the inside air as well as net radiation fluxes between the inside surfaces. Thus the conductive heat flux for each element of the inside surfaces is

$$q_{cd,i} = q_{cv,i} + q_{r,i} + q_{s,i} + q_{\ell,i} \tag{4}$$

The last two terms denote the heat flux due to resurfacing operations and the latent heat due to vapour condensation (mass transfer to the ice sheet) and are only taken into account for the ice surface. The conductive and convective heat fluxes q_{cd} and q_{cv} are calculated by the CFD code while the net radiative heat flux q_r between surface i and all the N inside surfaces of the envelope is calculated using Gebhart's method [18]. The calculation of q_r requires the emissivity of all N elements of the ice rink envelope and the N^2 radiation view factors between these surfaces. A specially developed subroutine implementing Gebhart's method was written and incorporated into the CFD code. This subroutine has been

described and partially validated by Bellache et al. [11]. Its predictions are compared with experimental results in section 5 of the present paper.

The total loads due to vapour condensation, resurfacing and heat dissipation in the pipes under the ice are calculated assuming quasi-steady conditions. In particular, the formulation from ASHRAE [19] has been used to calculate the resurfacing heat flow on the ice surface. Thus:

$$q_{\ell,i} = \dot{m}_{in}(C_{in} - C_{ou})h_{ig} \tag{5}$$

$$q_{s,i} = \rho V_f \left[4.2(T_f - 0) + 334 + 2(0 - T_{ice}) \right] \tag{6}$$

$$q_{p,i} = \rho g Q H_{sys} \tag{7}$$

$q_{\ell,i}$ and $q_{s,i}$ are attributed to the ice surface (cf. equation 4) while $q_{p,i}$ is attributed to the brine.

The horizontal surface through the centerlines of the floor pipes is assumed to be isothermal. This assumption is based on the small center-to-center spacing of the floor pipes as well as on the small increase of brine temperature (~1°C) between the supply and return headers. For the purposes of the present study, the temperature at this centerline position is assumed to be equal to the average brine temperature in the floor pipes.

The resurfacing heat load is assumed to be uniformly distributed over the ice surface and over a time interval of 10 minutes. When resurfacing takes place the total heat reaching the brine generally exceeds the capacity of the refrigeration plant. During these time intervals the energy balance for the brine can be approximated as follows:

$$M C_p \frac{d\overline{T}}{dt} = \sum_i (q_{cd,i} + q_{p,i}) \tag{8}$$

The summation on the right hand side includes all the conductive fluxes reaching the isothermal surface from above (ice side) and below (ground side). The heat capacity term on the left hand side includes contributions from the brine and the concrete in the appropriate finite volumes.

When resurfacing is not taking place the capacity of the refrigeration plant is greater than the total heat transferred to the brine. During these time intervals, the space average of the brine temperature is assumed to be constant. Therefore, the corresponding energy balance is:

$$\dot{m} C_p \Delta T = \sum_i (q_{cd,i} + q_{p,i}) \tag{9}$$

The total power input due to lighting is 28 kW and the maximum number of spectators is fixed at 200. The heat generated per spectator is taken as 100 W [17] and is assumed to

be uniformly distributed over the stands. The ground is, in some cases, heated by electrical resistances with a power of 8 kW.

3. Calculation procedure

The numerical solution of the system of coupled partial differential equations has been carried out using the finite volume method, a staggered grid and the hybrid scheme for discretization of the convection terms. The SIMPLE algorithm was used for the pressure correction and the solution was obtained iteratively using a commercial CFD code [20].

The calculation domain was first subdivided into zones corresponding to the physical discontinuities appearing in Fig. 1. In the present case there are 27 zones in the X direction and 30 zones in the Y direction. The number of grid points, or finite volumes, within each of the resulting 810 subdivisions of the domain has been varied to ensure that the results are independent of their number. The discretization grid was always non-uniform with higher density near the air inlets and the solid surfaces where velocity and temperature gradients are high. Numerical tests were carried out with 100x50 and 200x100 grid points in the X and Y directions, respectively. The number of iterations was also varied. The results are essentially identical. Therefore, the results reported throughout this paper were calculated with 100x50 grid points and 10000 iterations. Analogous tests were conducted with different time steps. The results presented here were all calculated with a time step of 600 seconds. Finally, the simulation was run for two days with identical 24-hour meteorological conditions in order to insure that the calculated "typical" daily profiles of the refrigeration loads are periodic. Under-relaxation was often necessary to achieve convergence, which was declared when the cumulative residuals for each of the conservation equations were less than 10^{-6}.

4. Discussion and simplification of the model

Chen and Van der Kooi [5] proposed a methodology for the solution of combined problems of energy analysis and indoor flow which takes into account the interaction between the indoor and outdoor environments but neglects heat transfer by radiation between the interior surfaces of the room. They used time-independent air flow patterns calculated by a CFD code with the k-ε turbulence model for typical situations. They then solved the algebraic room energy balance equation with a load calculation code which used transient temperature differences between the air near the inside surfaces and the middle point of the room. This temperature difference was evaluated from the solution of the transient differential energy equation obtained by the CFD code with the predetermined time-independent velocity distribution. Iterations between these two equations were necessary to achieve convergent results. This simplification is valid if the hydrodynamic field is not influenced by the internal and external conditions. The same approach was used here, i.e. it was assumed that the velocity field is constant over a period of 24 hours. It was determined by solving the hydrodynamic equations using the average external and internal

conditions for the 24 hour period under consideration. On the other hand, the temperature field and the heat fluxes were calculated using the time dependent energy equations and the corresponding variable meteorological and operating conditions. This simplification reduces considerably the calculation time since it is not necessary to solve the Navier-Stokes equations at every time step. In order to validate this approach we have carried out several simulations with the adopted CFD code for turbulent mixed convection in the selected ice rink. The simulations results for the extreme (6 ºC and 20.6 ºC) and average (15.2 ºC) external temperatures of the typical spring day for Pittsburgh are presented here. Figs. 2a, 2b, 3a and 3b show that the velocity profiles at different locations above the ice surface are practically independent of the external temperature. Even more important is the fact that the heat load values due to radiation, condensation and convection are essentially the same whether calculated with the velocity field corresponding to the actual (Table 2) or the average ambient temperature (Table 3).

These results indicate that the flow field in the ice rink is essentially independent of the sol-air temperature. Therefore it only needs to be calculated once, for the average sol-air temperature. This flow field can then be used for the solution of the transient energy equation in order to determine the refrigeration load profile over the entire 24-hour period. This simplification reduces the computer processing time approximately by half. It is especially appropriated for the present study as numerous simulations were required for the sensitivity analysis.

For the present purposes, the two-dimensional transient numerical model described earlier was validated with measured data from the reduced-scale ice sheet test bench and a selected full-scale ice rink.

5. Validation with measurements from the reduced-scale test bench

Full-scale tests of buildings are often prohibitively expensive and time-consuming. Small-scale model studies are usually used in order to resolve these difficulties, and to allow the testing of a particular concept in the laboratory. The principal purpose of the reduced-scale ice sheet test bench is to validate the numerical model previously described under controlled conditions while restricting the unavoidable environmental and space conditioning effects of an existing building. This ice rink model indicated that radiation heat exchange is the dominant thermal load transmitted to the ice. In order to establish the reliability of the model, and in particular that of the radiation subroutine, it was important to test it under extreme conditions (i.e. high and low emissivity).

The test bench was then designed to represent a small part of a full-scale ice rink. It is a completely self-contained unit that contains a 1010 mm x 2100 mm x 127 mm concrete slab embedded with 31.8 mm plastic pipes in the same manner as would be found in a full-scale ice rink. A total of 12 thermocouples (T/C's) are embedded in the concrete near the centre of the slab. The piping is connected in a single circuit with a flow rate of a typical ice rink – the only difference being that the test stand circuit is 21 m long rather than the 122 m

common in ice rinks. The concrete slab was constructed with threaded inserts to facilitate its placement in the test stand frame. It is surrounded by polyisocyanurate insulation with 178 mm underneath and 152 mm around the perimeter. The bottom insulation contains 4 equally spaced thermocouples to measure the temperature profile through the insulation. The heat transfer loop consists of a pump, a heat exchanger and manual flow control valves. Loop instrumentation consists of two high-precision, dual-output Coriolis meters measuring density and mass flow rate, one gear type positive displacement flow meter and 6 high precision 4W-RTD's with thermowells. A 1.6 m high enclosure composed of lightweight, high emissivity (blackbody), thin-wall fluted (corrugated) plastic sheeting material surrounds the space above the ice slab. The 'enclosure skin' temperature is measured with 20 gauge thermocouples embedded in the 'flutes' of the enclosure sheeting. The lightweight enclosure was built to minimise infiltration of air and moisture from the ambient environment.

The experimental results presented here are based on average values obtained under steady state conditions. Fig. 4 shows the vertical temperature distribution near the geometric center of the ice sheet for walls with high emissivity ($\varepsilon_w = 0.98$). The symbols denote the measured data whereas the solid line represents the numerical results. The 0 mm height in this figure corresponds to the concrete-ice interface. The ice thickness is 25.4 mm. Both measured and numerical data reveal that the radiation exchange between surfaces plays a predominant role in the establishment of the vertical temperature profiles in the enclosure since air movement in the test bench is negligible. The agreement between the experimental and numerical results is very satisfactory throughout the domain (insulation, concrete, ice and air).

In the case of low wall emissivity ($\varepsilon_w = 0.05$), Fig. 5, the calculated temperatures are in good agreement with the experimental results in the solid parts of the test bench (ice, concrete and insulation). On the other hand, the predicted air temperature above the ice surface differs from the experimental profile. This difference may be in part attributed, to the uncertainly regarding the value adopted for the emissivity of the walls.

In the case of the reduced-scale test bench, the ice sheet load is the sum of the convective flux between the air and the ice plus the net radiation flux between the ice and the inside surfaces of the test bench. Its experimental and calculated values were compared in order to test the capability of the model to simulate the heat transfer mechanisms that take place in the test bench. The experimental values were determined by subtracting estimates of the power dissipated by the pump and the heat gain in the external piping loop from the enthalpy change of the brine. Thus, for an ambient temperature of 8 °C, a brine mass flow rate of 36.7 g/s and a brine inlet temperature of -10.9 °C, the experimental value of the ice sheet load was estimated to be 331 W. Under steady state conditions this ice sheet load is equal to the conductive heat flux in the ice. The numerical model calculated the later using a two-node approximation for the temperature gradient in the ice. For the operation conditions defined previously, the numerical estimate of this quantity was equal to 288 W. In view of the experimental uncertainty and the different hypotheses used to obtain these two values we consider that their agreement is satisfactory.

6. Validation with data from the full-scale ice rink

Measurements were conducted in the selected Montreal ice rink in order to validate the numerical model and to better understand the physical processes affecting the cooling load of the ice sheet and the comfort of the occupants. The actual weather data from a nearby meteorological station was used for these simulations.

The main objective of the present study is the calculation of the ice sheet heat load. This includes the contributions of convection between the air and the ice, net radiation exchanges between the ice and the other surfaces of the envelope, condensation of vapour and resurfacing (cf. Equation 4). Therefore, particular attention was given to the comparison of calculated values of this variable with corresponding results derived from direct measurements. Fig. 6 shows the evolution of the total heat flux into the ice from above (due to convection, condensation, radiation and resurfacing) measured by four flux meters installed between the ice and the concrete floor, as well as the corresponding calculated profile over a 24 hour period. During that day, five resurfacing operations were performed between midnight and 7:00 PM. They are shown in the graph by the pronounced peaks of very short duration. The transient response is very well predicted by the simulation. Furthermore, the predicted heat flux is close to three of the four measured values (the fourth flux meter did not seem to work properly since it gives very low values). The maximum heat flux into the ice is equal to approximately 300 W/m^2.

The daily average value of the heat flux from above the ice sheet was evaluated from the measured quantities by three different methods.

a. Its first estimate was obtained by integrating the readings of the four heat flux sensors installed under the ice sheet. The corresponding daily mean values for January 7, 2005 are: 90.7 W/m^2, 87.8 W/m^2 108.6 W/m^2 and 53.3 W/m^2. As mentioned before, one of the four sensors gives significantly lower values than the other three and its readings have therefore been disregarded.

b. The second estimate was obtained by calculating the enthalpy change of the brine ($\dot{m}Cp.\Delta T$) and by subtracting the pumping power and the calculated ground gain. Thus, for January 7, 2005 this method gives a value of 96.4 W/m^2.

c. The third estimate is obtained from the measurement of the total electrical consumption of the refrigeration system. The consumption of the operating compressors is obtained by subtracting the corresponding values for the pump and ventilators from this measured quantity. The result of this first operation is multiplied by the efficiency of the electrical motors and the COP of the refrigeration system (obtained from the data supplied by the manufacturer for the observed operating conditions) to yield the heat absorbed by the refrigerant in the evaporators. The required value of the heat flux into the ice is finally obtained by subtracting from this quantity the pumping power and the calculated piping heat gain. This method gives a value of 99.9 W/m^2 for January 7, 2005.

The corresponding result calculated by the model is equal to 89.7 W/m^2. This value is lower than the estimates based on methods b and c (by about 7% and 10% respectively) but is between two of the three acceptable values obtained from the heat flux sensors. In view of the complexity of the phenomena taking place in the ice rink, the hypotheses incorporated in the model, and the differences between the values obtained from measured quantities, we believe that these results confirm the validity of the model.

7. Sensitivity analysis

The objective of this section is to show how the cooling load of the ice sheet is affected by climatic conditions as well as by changes in design and operating characteristics. In order to accomplish this sensitivity analysis, we used the validated model to simulate the behaviour of a building with the same characteristics of the ice rink presented in Fig. 1 (including the operating schedule), but with different climatic conditions, ice thicknesses, sub-floor insulations, ice surface temperatures, ceiling emissivities and coolant temperatures. The sensitivity analysis was carried out with different values of these parameters. Three site/season combinations have been selected to analyze the effects of the climate: spring and summer in Pittsburgh and winter in Edmonton. The selected design characteristics include two options for the sub-floor insulation (0.0 cm and 7.6 cm thicknesses), two options for the sub-floor conditions (with and without heating) and three ceiling emissivities (0.9, 0.5 and 0.1). The operating characteristics include three ice thicknesses (2.5, 5.1 and 7.6 cm); two operating schedules (weekday and weekend) that take into account lighting, resurfacing, ventilation and occupancy; and three average brine temperatures (-7, -9 and -11 °C). Instead of carrying out 648 simulations (the total combinations of the 18 values of the seven parameters) only 198 simulations were executed. This reduction was justified by the fact that the heat fluxes to the secondary coolant from above the ice sheet are independent of the conditions below the concrete slab (i.e., floor insulation thickness and ground heating) and the heat fluxes to the secondary coolant from below are independent of the conditions above (i.e., schedule, meteorological conditions, ice thickness, ceiling emissivity). Therefore only 36 combinations of meteorological conditions, floor configurations, insulation thicknesses and average coolant temperatures were required to determine the effect of these parameters on the heat flux from below. On the other hand, 162 combinations of schedules, meteorological conditions, ice thicknesses, average coolant temperatures and ceiling emissivities were required to determine the effect of these parameters on the heat flux toward the coolant from above. It is noted that each simulation generated a 24 hr load profile.

The analysis in this section uses the scenario which corresponds to meteorological conditions for Pittsburgh in spring (daily average of the sol-air temperature = 15.2 °C) as the base case. For this base case there is no sub-floor heating, the ice and sub-floor insulation thicknesses are respectively 5.1 cm and 7.6 cm, the average brine temperature in the floor pipes is –9°C, the ceiling emissivity is 0.9 and the weekend operating schedule applies. The effects of all the seven previously mentioned parameters on the daily cooling load of the ice sheet (by convection, radiation, condensation and resurfacing operations)

and on the daily cooling load of the brine (heat flux from the ground and heat gain due to friction in the floor pipes) were evaluated. For each simulation only one parameter was modified while all others retained the value specified in the description of the base case.

7.1 Effects of brine temperature

The impact of the brine temperature variation on the cooling load for the base case is presented in Fig. 7. It is noted that the convective, radiative and ground loads increase significantly as the average brine temperature decreases. The values of the latent heat load due to vapour condensation and solidification remain the same (within ± 0.2%) for the three cases. The resurfacing and friction loads are not affected by the variation of this parameter. The total daily heat load is 2.250, 2.410 and 2.530 kWh/m^2 respectively for average brine temperatures of -7 ℃, -9 ℃ and -11 ℃. Since the quality of the ice is strongly related to its temperature, it is important to determine how this value is affected by the brine temperature. Calculated results, not included here for lack of space, show that when the brine temperature increases so does the ice temperature while their difference decreases.

7.2 Effects of the ceiling emissivity

Fig. 8 shows that the effect of ceiling emissivity (ε_c) on the cooling load is quite strong; the total ice load is reduced by 25% (from 2.410 to 1.800 kWh/m^2) while the radiation load is lowered by 63% (from 1.045 to 0.390 kWh/m^2) when the emissivity decreases from 0.9 to 0.1.This important reduction in the radiation load confirms that most of the heat comes from the ceiling and validates the interest of adopting low emissivity ceilings. The temperature difference (TD) between the ice and the brine decreases from 4.1 to 2.9 ℃ with a reduction of ε_c from 0.9 to 0.1. The benefit of the decrease in the TD is either a colder and better quality ice (for the same brine temperature) or a further reduction in the energy consumption of the refrigeration system (given that the brine temperature set-point could be higher for the same ice temperature).

The convective heat flow between the air and the ice is proportional to the temperature difference between the ice sheet and the air over it. Since the reduction of ε_c results in a colder ice and the temperature variation of the air over the ice is negligible, the convective heat flow between the air and the ice increases slightly when ε_c decreases.
The latent load due to the condensation and solidification of water vapour from the air onto the ice remains the same (within ± 2%) for the three cases. The load of the brine from the ground and pump (dissipation in the floor pipes) is essentially unaffected by ε_c.

7.3 Effects of the ice thickness

The total daily load of the ice sheet decreases slightly with an increase of the ice thickness (Fig. 9); it is reduced by 6.6% (from 2.484 to 2.320 kWh/m^2) when the ice thickness triples (from 2.54 to 7.62 cm). This is due to a small reduction of the radiation and convection loads, which come from above the ice.

These results could lead to a wrong interpretation of the effects of this parameter on the ice load; because one could assume that thicker ice is associated with lower cooling loads. A short explanation is therefore required. The computer model calculates the total ice sheet heat load by maintaining the brine temperature at a fixed value (-9 ºC) for the three cases presented in this section and therefore an increase in the ice thickness results in a higher ice surface temperature, as illustrated by Table 4. The increase in ice surface temperature leads to the decrease of radiative and convective heat fluxes, and of the total ice cooling load. However this increase of the ice temperature is undesirable since it implies a deterioration of the ice quality. Therefore increasing the ice thickness while maintaining constant the brine temperature is not recommended.

Table 4 shows that the temperature difference (TD) between the ice and the brine is lowered by 25% (from 4.8 ºC to 3.6 °C) when the ice thickness is reduced to a third (from 7.62 to 2.54 cm). The benefit of this decrease in the TD is the improvement in the performance and energy efficiency of the refrigeration system given that the brine temperature set point can be higher for a given ice temperature.

7.4 *Effects of the operating schedule*

Fig. 10 shows that the difference between weekends and weekdays is essentially due to the lower number of resurfacing operations in the latter case. The daily resurfacing load is significantly lower during weekdays (0.352 kWh/m^2) than during weekends (0.470 kWh/m^2) and compensates for the small increases of the convection and radiation loads which are due to a slight decrease of the average ice temperature. This temperature reduction is a direct result of the reduction in the daily number of resurfacing operations (9 on weekdays, 12 on weekends). The total load is 2.408 kWh/m^2 on weekends and 2.339 kWh/m^2 on weekdays. It is important to note that the load into the brine from the ground is totally unaffected by the operating schedule.

7.5 *Effects of the climate*

Fig. 11 illustrates the effects of the climate by comparing the base case (i.e., spring in Pittsburgh where the daily average of the sol-air temperature is 15.2 °C) with winter conditions in Edmonton (where the daily average of the sol-air temperature is -8.7 °C) and summer conditions in Pittsburgh (where the daily average of the sol-air temperature is 27.6 °C). Three of the four daily loads are lowest in winter and highest in summer, due to the corresponding changes of the outdoor dry-bulb temperature and absolute humidity. The influence of the climate on ground heat gains is very small while daily loads from resurfacing and friction are independent of climatic conditions. Overall, the influence of the climate on the total daily ice sheet load is very high. The total daily load increases by 46% (from 1.980 to 2.887 kWh/m^2) with a change in the climate from winter in Edmonton to summer in Pittsburgh.

7.6 Effects of thermal insulation under the concrete slab

Fig. 12 shows the effects of the insulation of the concrete slab on the daily loads. The influence of the insulation on ground heat gains is quite significant while daily loads from convection, radiation, condensation and resurfacing are not affected by this parameter. Overall the insulation of the concrete slab has a small effect on the total load.

7.7 Effect of sub-floor heating

The effect of this parameter is very small, the total load rising from 2.408 kWh/m^2 in the absence of heating to 2.462 kWh/m^2 when the ground is heated (an increase of 1.7%).

8. Conclusion

Turbulent flow with simultaneous mass diffusion in a ventilated ice rink coupled to conduction through its envelope and radiation between the inside surfaces of the latter has been modeled using the standard k-ε turbulence model and a modified version of Gebhart's method.

The transient model and numerical code used for the solution of the coupled partial differential equations were validated by comparing their predictions with measured values in a municipal ice rink situated in Montreal (Canada) and in a reduced-scale test bench. An important simplification which consists in computing the flow field only once daily was introduced and justified. This leads to a significant reduction of the CPU time required for a 24 hour simulation.

The simplified numerical code was used to evaluate the daily heat flux profiles into the ice and the brine for a representative ice rink and steady periodic meteorological conditions. 198 simulations were carried out in order to establish the influence of different design and operational parameters on the thermal loads of the ice and the brine. Results were obtained for three different climatic conditions, three ice thicknesses, two thicknesses of the sub-floor insulation, three values of the ceiling emissivity, two operating schedules and three brine temperatures. The main results are:

- The net radiative flux into the ice is the highest contributor to the total daily ice sheet cooling load.
- The total daily ice sheet cooling load is reduced by 25% when the ceiling emissivity decreases from 0.9 to 0.1.
- An increase of the ice thickness by 2.5 cm results in a slight decrease of the total cooling load and an approximately 1.6 °C increase of the ice surface temperature.
- The operating schedule has a significant effect on the load due to resurfacing operations which is proportional to the number of such operations. However, since this load represents only a small part of the total daily cooling load, its effect on the latter is rather small.

- The effect of climate is important; an increase of the average daily sol-air temperature from -8.7 °C to 15.2 °C results in a 18% increase of the total cooling load.
- Sub-floor insulation and heating have a very small effect on the total cooling load; they are, however, necessary in very cold climates to prevent freezing and heaving of the ground.

References
[1]   M. Lavoie, R. Sunyé, D. Giguère, Potentiel d'économies d'énergie en   réfrigération dans les arénas du Québec, Report prepared by the CANMET Energy Technology Center (Canada), 2000.
[2]   Q. Chen, J. Srebric, A procedure for verification, validation, and reporting of indoor environment CFD analyses. HVAC&R Research 8 (2) (2002) 201-216.
[3]   P.V. Nielsen, A. Restivo, J.H. Whitelaw, Buoyancy-affected flows in ventilated rooms. Numerical Heat Transfer 2 (1979) 115-127.
[4]   L. Davidson, Ventilation by Displacement in a Three-Dimensional Room: A Numerical Study. Building and Environment 24 (4) (1989) 363-372.
[5]   Q. Chen, J. Van der Kooi, Accuracy-A Program for Combined Problems of Energy Analysis, Indoor Airflow and Air Quality. ASHRAE Transactions 94 (1994) 196-214.
[6]   Y. Li, Simulation of Flow and Heat Transfer in Ventilated Rooms, Transactions of the Dept. of Mechanics/ Applied Comp. Fluid Dynamics, Royal Institute of Technology, Stockholm (Sweden), 1992.
[7]   Y. Li, L. Fuchs, M. Sandberg, Numerical prediction of air flow and heat radiation interaction in a ventilation: Numerical Predictions. Energy and Buildings 20 (1) (1993) 27-33.
[8]   X. Yuan, Q. Chen, A. Moser, P. Suter, Numerical simulation of air flows in gymnasia. Indoor Environment 1 (1992) 224-233.
[9]   C. Yang, P. Demokritou, Q. Chen, Ventilation and Air Quality in Indoor Ice Skating Arenas. ASHRAE Transactions 160 (2000) 339-346.
[10] O. Bellache, M. Ouzzane, N. Galanis, Numerical prediction of ventilation patterns and thermal processes in ice rinks. Building and Environment 40 (2005) 417-426.
[11] O. Bellache, M. Ouzzane, N. Galanis, Coupled conduction, convection, radiation heat transfer with simultaneous mass transfer in ice rinks. Numerical Heat Transfer Part A 48 (2005) 219-238.
[12] Bellache, O., R. Sunyé, N. Galanis, M. Ouzzane, D. Giguère, Two-Dimensional Transient Model of Airflow and Heat Transfer in Ice Rinks (1289-RP). ASHRAE Transactions, Part2 112 (2006) 1-11.

[13] O. Bellache, Modélisation des écoulements avec transfert de chaleur et de masse dans un milieu réfrigéré (application à l'aréna), Thèse de Doctorat (Ph.D), Université de Sherbrooke (Canada), 2004.

[14] B.E. Launder, D.B. Spalding, The numerical computation of turbulent flows. In Computer methods in applied mechanics and energy 3 (1974) 269-289.

[15] Q. Chen, Comparison of Different k-ε Models for Indoor Air Flow Computations, Numerical Heat Transfer, Part B 28 (1995) 353-369.

[16] ENERGYPLUS. US Department of Energy. www.eere.energy-gov/building.

[17] ASHRAE handbook - Fundamentals, Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 2005

[18] R. Siegel, J.R. Howell, Thermal radiation heat transfer, Hemisphere, 1981.

[19] ASHRAE handbook - Refrigeration, Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc, 2002.

[20] D.B. Spalding, The Phoenics reference manual. CHAM TR/200a (PIL), TR/200b, 1991.

Figure Captions

Fig. 1. Schematic representation of the selected ice rink.

Fig. 2b. Profiles of the horizontal velocity component at position X = 30 m.

Fig. 3a. Profiles of the vertical velocity component at position Y = 1m.

Fig. 3b. Profiles of the vertical velocity component at position Y = 3m.

Fig. 4. Vertical temperature profile in the middle section (X= 0.7 m, ε_w =0.98, Tb = -10.9 ºC)

Fig. 5. Vertical temperature profile in the middle section (X=0.7 m, ε_w =0.98, Tb = -10.9ºC).

Fig. 6. Profiles of heat flux into the ice from above (Montreal, January 07 2006).

Fig. 7. Impact of the brine temperature on the cooling load.

Fig. 8. Effects of ceiling emissivity.

Fig. 9. Effects of ice thickness.

Fig. 10. Effects of operating schedule.

Fig. 11. Effects of the climate.

Fig. 12. Effects of the thermal insulation under the concrete slab.