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Yearly simulation of the interaction between an ice rink and its refrigeration system: A case study

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ABSTRACT

A transient model of airflow and heat transfer in an indoor ice rink and a quasi-steady model of its refrigeration system have been coupled and used to simulate their response to the time dependent ambient conditions and operating schedule for a typical meteorological year. The results for two different cases show that it is possible to reduce significantly the time of operation of the compressors and the energy consumption of the refrigeration system by simultaneously reducing the ceiling emissivity and increasing the secondary coolant temperature without affecting the quality of the ice.

Simulation annuelle de l'interaction entre une patinoire à glace et son système frigorifique: étude de cas

Mots clés : Énergie ; Consommation ; Installation frigorifique ; Modélisation ; Immeuble ; Patinoire

1. Introduction

Ice rinks are large buildings which consume a great amount of energy and generate considerable emissions contributing to the greenhouse effect. The thousands of such buildings in North America offer a big potential for improvement since most of them are old, oversized and were not designed for high energy efficiency (Nichols, 2009). However, the available methods for the calculation of their annual energy consumption are not as developed as those for conventional buildings (Bellache et al., 2006). The large dimensions of these buildings, the simultaneous need of refrigeration and heating, as well as time-dependent schedules for ventilation, lighting and resurfacing of the ice are some of the reasons for this situation.

Readily available fast computers have prompted some researchers to use Computational Fluid Dynamics (CFD) for the analysis of phenomena taking place in ice rinks. Thus,

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Nomenclature

T_{a_out}	outlet air temperature from the condensers (°C)
T_{a_in}	inlet air temperature to the condensers (°C)
T_{ice}	ice surface temperature (°C)
T_{b_out}	outlet brine temperature from the refrigeration system (°C)
T_{b_in}	inlet brine temperature to the refrigeration system (°C)
T_{int}	temperature at the interface between ice and concrete slab (°C)
P_{cd}	condensation pressure (kPa)
P_{ev}	evaporation pressure (kPa)
Q_{ice}	heat flux from AIM to BIM (kW)
Q_R	refrigeration load (kW)
Q_R^*	refrigeration load per unit in operation $= Q_R/N_C$ (kW)
N_C	number of compressors in operation
Δt	timestep (min)

Abbreviations

BIM	Below Ice Model
AIM	Above Ice Model
REFSYS	Refrigeration System

Yang et al. (2000) used a CFD code to evaluate air quality in an ice rink but did not calculate heating and refrigeration loads. Later, Bellache et al. (2005a,b) developed a steady state 2D CFD model which predicts velocity, temperature, absolute humidity distributions and calculates the fluxes toward the ice due to convection from the air, to condensation of vapor and to radiation from the walls and ceiling. It was improved by Bellache et al. (2006) by including transient phenomena, heat transfer through the ground and gains from lights as well as the effects of resurfacing and dissipation of pump work in the coolant pipes. This model was validated with experimental results by Ouzzane et al. (2006). The great disadvantage of these CFD based models is the considerable calculation time required to obtain results for even a typical 24-h period. Shortly afterwards, Daoud and Galanis (2006) and Daoud et al. (2007) proposed an alternative method to CFD which requires considerably less calculation time and computer memory and provides good results for an entire year. This model (AIM) takes into account the transient phenomena and 3D geometry of the building as well as the resurfacing and occupation loads but assumes that the temperature of the brine entering the concrete slab under the ice is constant. Heat fluxes toward the ice surface were calculated and are in good agreement with measurements.

Following this work, Seghouani et al. (2009) developed a complementary model to AIM, called BIM, which calculates the transient heat fluxes to the brine pipes from the soil under and around the foundation of the ice rink. The two models AIM and BIM have been successfully coupled under TRNSYS software and permit a wide range of parametric studies (such as determining the effect on energy consumption of the climate, the ice sheet thickness, the emissivity of the ceiling, the brine temperature entering the concrete slab, etc.).

In parallel, Seghouani and Galanis (2009) developed a quasi-steady state model of the air-cooled refrigeration system of the ice rink (REFSYS) based on a combination of thermodynamic relations, heat transfer correlations and data available in manufacturers' catalogues. Results such as the monthly energy consumption, the number of compressors in operation as well as the heat rejected by the condensers were shown among numerous other outputs. This model requires as an input the time-dependent refrigeration load and the corresponding brine temperature entering the evaporators. These were taken from a data file calculated by the AIM + BIM models. However, REFSYS did not take into account the complete interaction between the ice rink and its refrigeration system since the refrigeration load was not recalculated when the refrigeration system was unable to cool the brine to the assumed temperature of the brine used by the AIM and BIM models.

The purpose of the present paper is to integrate the previous works by coupling the refrigeration system (REFSYS) and the ice rink (AIM and BIM) models. The resulting model constitutes a complete simulation tool of an ice rink and its refrigeration system, able to predict the annual energy consumption by the compressors and the ventilation system, the ice surface temperature evolution as well as the potential for heat recovery.

2. Ice rink description

Fig. 1 shows a schematic representation of the ice rink under study. A short description is presented here since more details are available in our previous publications (Seghouani et al., 2009). The ice surface is 61 m long and 25.9 m wide. The spectator stand is heated by 8 radiant heaters which are controlled by an electronic thermostat with a set point equal to 15 °C; its hysteresis is ±0.2 °C and its nocturnal setback is 7 °C. Seven inlets supply a stream of ventilation air. Its flow rate is 4270 L s^{-1} except during resurfacing of the ice when it is increased to 10,384 L s^{-1}to evacuate the combustion gases of the resurfacing vehicle. The air exits through 4 outlets on the walls. Heat gains from lighting are 10 W m^{-2} above the ice and 5 W m^{-2} above the stands; those due to the presence of the audience are also taken into account while the number of spectators is specified according to a weekly schedule. The ice resurfacing is done several times per day, lasts 12 min and is modeled as a 1 mm film of hot water at 60 °C. Its frequency, specified in the schedule mentioned above, is higher in the evenings and weekends. The ground structure beneath the ice rink comprises horizontal layers of ice (2.5 cm), concrete (15 cm), thermal insulation (10 cm), sand (20 cm) and, finally, soil. The total depth included in the calculation domain is 4 m. The secondary coolant used to maintain the ice at the desired temperature is calcium chloride brine (concentration 20% by mass). It is supplied from a header located at the west end of the ice sheet and circulates in the concrete slab within 74 uniformly distributed, four-pass polyethylene tubes. An electrical heater of 8 kW is activated in the sand layer when the ground temperature at a depth of 4 m is below 4 °C to prevent