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Optimal insulation for ice rink floors

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ABSTRACT

This paper presents an economic analysis to estimate the optimal insulation levels required for indoor ice rinks floors. The analysis is carried out using a thermal ice rink floor system model integrated into EnergyPlus to determine the energy performance of ice rink facilities under different design and operation conditions. Moreover, an ice rink facility is first simulated using a modified version of EnergyPlus that includes the developed ice rink floor system model. The ice rink facility simulation model is then calibrated using monthly utility data. The calibrated ice rink facility simulation model is used to determine economically optimal insulation levels to be placed beneath an ice rink floor in order to reduce energy use and cost for operating ice rink arenas in selected US climates. Finally, the results of sensitivity analyses are utilized to develop a simplified calculation method to determine optimal ice rink floor insulation level as a function of both climatic and economic parameters.

1. Introduction

Ice rinks are highly energy intensive facilities in the entertainment industry. According to a DOE report, a typical indoor ice rink arena in Massachusetts had an average electric consumption of 730,000 kWh costing up to $70,500 for just a 7–8 month season [1]. Another study showed that energy costs to operate an ice rink facility in Canada for 8 months are on average $86,000. Unfortunately, it is difficult to accurately estimate refrigeration load and energy use of ice rinks using commonly used building energy simulations due to lack of detailed models of ice rink floor systems [2].

Indeed, very limited thermal analysis of indoor ice rinks has been reported in the literature. Most of the published studies typically have focused on indoor air quality including ventilation requirements and systems [3–7]. Only few models suitable for thermal analysis of ice rink facilities including ground-coupled floor systems have been developed [8,9]. In particular, Mun and Krarti [9] have developed an ice rink floor thermal model, based on the conduction transfer function method. In particular, the model utilizes conduction transfer function (CTF) method to account for thermal interactions between brine tubing system and ice floor structure in order to estimate the refrigeration load associated with the ice-making equipment. The model has been validated against experimental data obtained under laboratory testing conditions.

In the analysis carried out in this paper, the ice rink thermal model developed by Mun and Krarti is integrated into EnergyPlus, a detailed energy simulation program, and is considered to estimate the energy performance of various design and operating strategies for ice rink floor. Specifically, an energy model for an actual ice rink floor facility is developed in the analysis. Using the ice rink thermal model, the impact and the cost-effectiveness of floor insulation to reduce energy use and cost for operating ice rinks are investigated. In particular, the optimal insulation level to be placed beneath an ice rink floor is determined for select US climates. In this study, the optimal level of insulation is estimated using the life cycle cost (LCC) analysis method.

2. Ice rink floor system model

For this study, a numerical model is developed to determine the temperature distribution inside various layers of ice rink floor including ground medium for slab-on-grade floor constructions as outlined in Fig. 1.

For the ice-making or charging periods (i.e., when the ice rink floor opens at the beginning of the season) as well as ice-melting or discharging period (i.e., when the ice rink is closed at the end of the season and ice is melted), solid and liquid phase-change processes occur at the ice/water layer above the ice rink floor. The phase-change problem can be formulated using Eq. (1) [10]:

$$\rho \frac{\partial H}{\partial t} = \nabla(K\nabla T) \quad (1)$$

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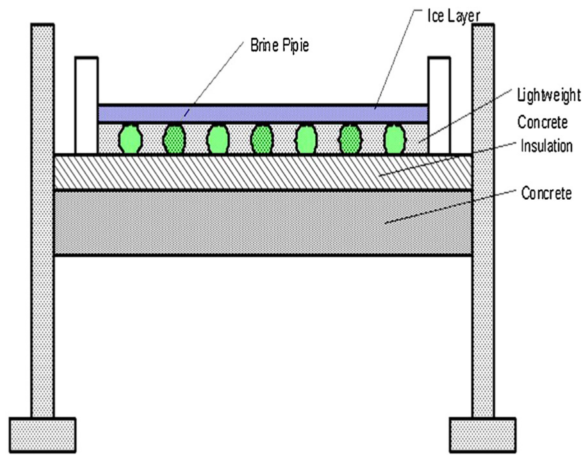

Fig. 1. Ice rink floor model with its ice/water layer, insulation, concrete, and ground medium.

The enthalpy defined as the heat content per unit mass, can be estimated for water as following:

$$H = \begin{cases} C_{ice}T; & T \le Tm - e \\ C_{ice}(T_{m-e}) + \left[\dfrac{C_{ice}+C_{water}}{2} + \dfrac{H_{fg}}{2e}\right](T - T_m + e); & Tm - e < T < Tm + e \\ C_{water}T + (C_{ice} - C_{water})Tm + H_{fg}; & T \ge Tm + e \end{cases} \quad (2)$$

where

- H: enthalpy (J/kg)
- C_{water}: water heat capacity (4200 J/kg K)
- C_{ice}: ice heat capacity (2100 J/kg K)
- H_{fg}: phase change enthalpy (333,700 J/kg)
- T_m: phase change temperature (0 °C for water)
- e: temperature interval for freezing area (0.5)

To solve Eq. (1), the apparent heat capacity method is utilized [11,12].

Beneath the ice rink floor, the dominant heat transfer in the concrete, insulation, and ground medium. A Two-dimensional analysis is carried to solve the heat conduction beneath the ice rink floor [10]:

$$\frac{\partial}{\partial x}\left(k\frac{\partial T(r,t)}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T(r,t)}{\partial y}\right) = \rho c_p \frac{\partial T(r,t)}{\partial t} \quad (3)$$

where

- T: temperature [°C]
- r: vector space of x or y [m]
- c_p: specific heat [J/kg °C]
- k: thermal conductivity [W/m °C]
- ρ: density [kg/m^3]
- t: time [s]

The boundary conditions for Eq. (3) depend on the specific ice rink configuration. Typically, the symmetry line (middle of the ice rink floor) can be modeled as an adiabatic surface. The upper water/ice layer surface can be modeled using a third boundary condition since a convective heat transfer occurs between the indoor air and the water/ice surface. A control volume approach and pure implicit finite difference technique is used to solve the heat conduction Eq. (3) with associated boundary conditions [13].

In order to integrate the solution of the ice rink floor model defined by Eqs. (1)–(3) in whole-building simulation tool, conduction transfer function (CTF) technique with heat source/sink is used as expressed by Eq. (4) [9]:

$$q''_{i,t} = \sum_{m=1}^{M} X_m T_{i,t-m+1} - \sum_{m=1}^{M} Y_m T_{o,t-m+1} + \sum_{m=1}^{k} F_m q''_{i,t-m} + \sum_{m=1}^{M} W_m q_{source,t-m+1} \quad (4)$$

where

- X_m: inside CTF coefficient, $m = 0, 1, \ldots, M$.
- Y_m: cross CTF coefficient, $m = 0, 1, \ldots, M$.
- T_i: inside face temperature
- T_o: outside face temperature
- F_m: flux CTF coefficient, $m = 0, 1, \ldots, k$.
- W_m: QTF inside term for the heat source/sink, $m = 0, 1, \ldots, M$.
- q'': surface heat balance including radiation from other surface, solar radiation, and convection heat flux on the inside surface

Mun and Krarti [9] developed and validated the ice rink floor CTF solution integrated within EnergyPlus using laboratory experimental data.

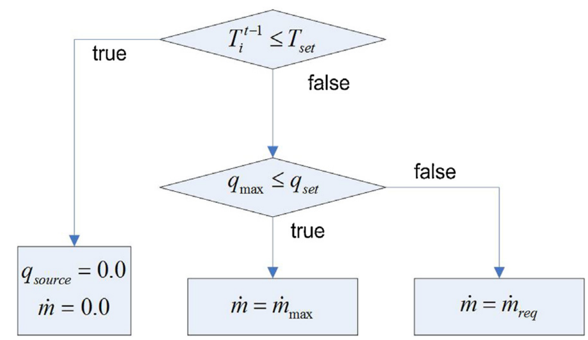

Fig. 2. Ice surface temperature control system algorithm.

Ice rinks can be used in a variety of sports and activities with specific requirements for ice quality and ice surface temperature. For figure skating, for instance, ice should be soft for better gripping needed for various motions. However, for ice hockey, players prefer hard ice surfaces to help with speed. Thus, for ice hockey, the top of the ice surface is usually kept at lower temperature than that for figure skating. Usually, when the air temperature above the ice surface is 7.2 °C, it is recommended that the ice be kept at −6.4 to −5.5 °C to be satisfactory for hockey, −4.4 to −3.3 °C for figure skating, and −3.3 to −2.2 °C for recreational skating [14]. The ideal temperatures for an ice rink facility vary with the quality of water used for ice making as well as the activity type. To maintain desired surface temperature settings, ice surface control is considered to be the most effective operating strategy for new rink arenas. In this control strategy, the brine flow rate is varied to reach the desired setpoint temperature for the ice rink surface. Fig. 2 illustrates the basic algorithm for the ice surface control strategy.

If the ice surface temperature of the previous time step, T_i^{t-1}, is lower than the setpoint temperature, T_{set}, the refrigeration system is turned off. Therefore, no heat rejection occurs from the ice rink floor structure, q_{source}, to the brine and the ice surface temperature increases. However, when the ice surface temperature of the previous time step, T_i^{t-1}, is higher than the setpoint temperature, T_{set}, the refrigeration system is turned on. In this case, the maximum heat flux to the structure from the brine, q_{max}, and the required heat flux to reach the setpoint temperature, q_{set}, are compared. The maximum heat flux to the structure from the brine, q_{max}, is the value of heat flux with the maximum brine mass flow rate, $\dot{m}_{max}$.