MODELING ANALYSIS OF ICE RINK PERFORMANCE

by

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Abstract

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Modeling Analysis of Ice Rink Performance

Thesis directed by Professor Moncef Krarti

A new EnergyPlus module is developed to model indoor ice rink facilities. The study outlined in the thesis presents the justifications of needs for an ice rink module for EnergyPlus, the modeling and integration approaches of the module within EnergyPlus, as well as the results of a series of sensitivity analyses carried out using the developed module. The new EnergyPlus module calculates the refrigeration load needed to maintain the ice rink surface temperature at the desired setpoint. Moreover, the module updates the zone heat balance to determine its impact on HVAC loads. The sensitivity analyses investigate the impacts of several design and operating parameters on the energy performance for an ice rink facility. An energy model for the ice rink facility is developed based on an existing ice rink located in Old Bridge, NJ. The results from the sensitivity analyses confirm that ice rink facilities are energy-intensive buildings due to the significant refrigeration loads. Moreover, the EnergyPlus based analyses indicate that a significant improvement of the energy performance of ice rink facilities can be achieved using easy to implement operation and design measures.

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Chapter I: Introduction

1.1 Introduction

In North America, ice rink facilities are used for different entertainment events such as figure skating, hockey games, and recreational skating. Ice rink arenas are designed to maintain the rink's floor surface at a temperature below freezing while maintaining thermal comfort in the facility. In addition, ice rinks typically operate continuously for 8 months to 11 month per year [1,2]. Therefore, these facilities consume a significant amount of energy per year leading to high cost of operation. According to the international ice hockey federation (IIHF) technical guides, the utility expenses can amount to $180,000/year for an ice rink facility that houses a 30-mx60-m ice pad [2]. The annual energy consumption for ice rink facilities average around 750 kWh/year which corresponds to triple the energy use intensity of office buildings [3]. A DOE report states that a standard indoor ice rink in Massachusetts consumes 730 MWh for 7-8 months operation costing $70,500 during 2010[4]. Moreover, most energy efficient Canadian ice rinks are reported to consume 800 MWh/year that is one third of 2400 MWh/year used by the least energy efficient facilities [5]. The average cost for operating an ice rink for 8 months is estimated to be $86,000 in Canada [6]. According to a CanmetENERGY report, ice rinks in Quebec consumes around 1,525,000 kWh/year with an average energy intensity use (EUI) is 500 kWh/m^2. In addition, the same report states that refrigeration consumes 50% of annual electricity consumption and that ice rinks tend to be the most energy intensive buildings in small Canadian towns with a population up to 25,000 [7,8]. In an EnergyStar document, it was reported that indoor ice rink source EUI ranges between 0.09 GJ/m^2 to 8.41 GJ/m^2 [9]. Another study shows that Swedish ice rinks consumes on average 1185 MWh/year and refrigeration accounts for 43% of electricity consumption [10,11,12].

Due to the impacts of several factors, such as climate and ice rink size, existing literature indicate different and wide energy consumption ranges for ice rink facilities. However, all reported data agree that ice rink facilities are energy intensive buildings that are expensive to maintain. Therefore, energy efficient design and operation of ice rinks are highly recommended.

Like other buildings, energy consumption and operation costs are the main concerns for facilities that houses indoor ice rinks. Therefore, it is important to consider having a well-insulated and airtight structure as well as high performance energy systems including refrigeration plant, mechanical ventilation, air de-humidification, HVAC, and lighting systems [2]. The well insulated and airtight structure would reduce the heat transfer to the outside, reduce infiltration and noise pollution. Installing a ceiling in the ice rink zone may increase maintenance costs due to indoor condensation. Indeed, the ceiling surface temperature becomes low due to radiant heat transfer from the ice rink surface. Thus, cold ceiling surface can cause condensation of any humid air that comes in contact with its surface resulting in moisture and condensation problems. To solve indoor condensation problems, installing low-emissivity ceiling would reduce the heat flow between the ice rink and the ceiling surfaces. In addition, low emissivity ceilings provide other advantages such as reducing HVAC thermal loads and reflecting light which would lower the need for electrical lighting requirements [2,13]. On the other hand, over insulating the roof would promote lower roof temperature and hence air condensation. As for ice pad structure, there are various floor types that require different specifications. In general, a typical ice rink floor consists of ice sheet, concrete, and insulation above the slab foundation. Some of the new ice rinks are constructed with heated slab to prevent ground from freezing with the potential of incurring structural damages [13]. Table 1.1 lists typical specifications for an ice rink floor:

Table 1.1: Ice Pad structure specifications [2,13]

Parameter	**Value range**
Ice Thickness	25mm-30mm
Concrete Thickness	100mm – 150mm
Refrigeration pipe spacing	Indirect Refrigeration Systems: • High flow rate refrigerant: 90-100mm • Low flow rate refrigerant: 20mm-40mm Direct system: 100mm
Refrigeration pipe diameter	Indirect Refrigeration Systems: • High flow rate refrigerant: 20-32 mm • Low flow rate refrigerant: 6mm Direct system: 12-22mm

Distance between the refrigeration pipes center and the bottom of the ice pad.	25-50mm
Thermal Insulation	100 m

Figure 1.1 shows the common floor section of an ice rink. There are different type of floors depending on whether the floor is used for other sports or not. In addition, some floors have heating pipes in the foundation layer to provide freeze protection [2, 13]. These floor layers are used for the analysis in the rest of the chapter.

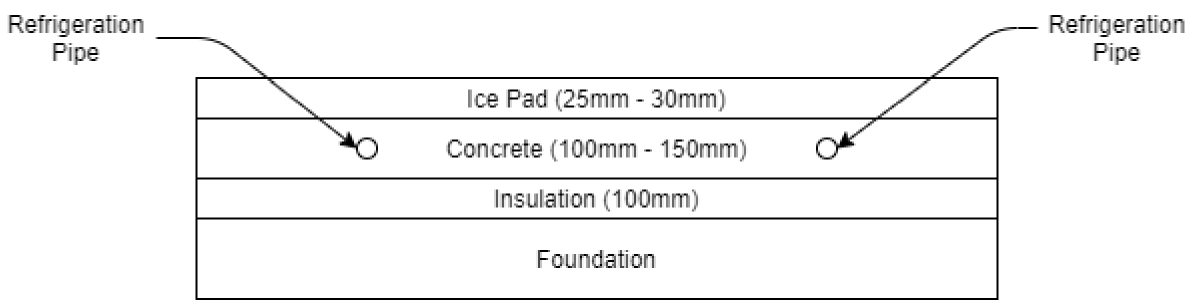


Figure 1.1: Typical Ice Rink Floor foundation.

Different ice rink activities require specific floor surface temperatures. In particular, hockey training activities and games require a -4°C and -5°C surface temperature, respectively. Moreover, figure skating training and competition activities require -3°C and -4°C, respectively [2]. To maintain ice rink surface at the desired temperature settings, refrigeration systems are needed. A standard ice rink would require 300-350 kW refrigeration capacity [2]. For indoor ice rink arenas, the refrigeration plants can be either direct or indirect. Fora direct refrigeration system, heat exchanger piping located under the ice pad acts as the evaporator. On the other hand, for an indirect system, the piping under the ice rink is connected to a separate evaporator. The direct system is more efficient than the indirect system but is more expensive. Factory made refrigeration units can be considered for indirect systems, but custom design is needed for direct systems [2]. For both systems, heat recovery can be used to save energy as it can provide 75% to 100% of the ice rink facility's heating needs [13]. Typically, condensers reject significant amount of heat that can exceed the facility's heating requirements. In most cases, at least 2 compressors are needed for each indoor ice rink arena. One compressor is responsible for normal load operation and the other one is to support the first system during the initial ice freezing phase when higher refrigeration capacity is needed. In addition to the refrigeration systems, ice resurfacer machines are used to maintain the quality of ice surfaces by making them smooth. The resurfacing operation simply consists of flooding the ice surface with hot water and use blades to shave any additional layer of ice formed from the frozen hot water. The ice resurfacer machine provides flood water between 55°C and 80°C. For a 30-mx60-m ice rink, the ice resurfacing machine would need between 0.4 m^3 and 0.7m^3 of water tank.

For the HVAC energy needs specific to ice rink facilities, space heating dominates the cooling thermal loads even for hot climates [2]. Indeed, the "free" radiative cooling effect provided by the ice rink surface can be significant and may require heating even during the summer months and in hot climates such as that of Miami. In addition to heating, ventilation is required to maintain healthy indoor air quality associated with pollutant emissions from people, building materials, and gas or propane resurfacers [2]. Moreover, dehumidification is often needed to maintain low relative humidity levels and avoid condensation. Excess moisture can cause undesirable consequences such as mold, corrosion, and fog. According to IIHF, to avoid fog, the relative humidity should not exceed 60% and 90% at 20°C and 5°C ice rink indoor air temperature, respectively [2]. Dehumidification can be provided mainly through two methods including air conditioning with dehumidification and heat recovery systems or use of desiccant wheels. In the first method, the air is cooled below the dew point and it condenses as the air passes over the cooling coil. In the second method, air is dehumidified as it passes over an absorbent material that has the ability to extract moisture [2].

1.2 Methodology

The ice rink represent a challenge when modelling it. In previous studies, the building HVAC was modeled separately in a building simulation software and the ice rink loads were provided by refrigeration consultants or analytical model [14,15**Error! Reference source not found.**]. Since the ice rink facilities are energy intensive, there is a need to have the ice rink as feature in whole building simulation software. Therefore, the goal of this project is to develop, integrate and test an ice rink feature in EnergyPlus. The ice rink feature reads the user inputs and calculates the refrigeration load to reduce the ice rink surface temperature to the desired setpoint assigned by the user. For every time step, the refrigeration load is calculated and EnergyPlus updates the zone heat balance. This process is similar to the existing EnergyPlus feature called low temperature radiant system.

In this thesis, the ice rink modeling approach is described in Chapter 2 including its implementation details into EnergyPlus ice rink model was described. The energy predictions of the ice rink module, as implemented in EnergyPlus, are validated against utility data obtained for an existing ice rink facility as outlined in Chapter 3. Then, a series of sensitivity analyses is conducted as outlined in Chapter 4 to better assess the impacts of various design and operation parameters on the energy performance of ice rink facilities. Finally, recommendations for future improvements are made as part of Chapter 5 to enhance the modeling capabilities of the developed ice rink module in EnergyPlus.

Chapter II: Modeling Approach

In this section, the model of the ice rink implemented in EnergyPlus is described in detail based on the reported literature. First, the design method based on the time to freeze the ice rink water is used to estimate the capacity of the refrigeration system. Then, the ice rink thermal loads are determined based on the refrigeration needs and the control strategies as outlined in the following flowchart.

2.1 Freezing

The ice rink model uses the time to freeze method [13] to calculate the capacity of the required refrigeration system. This calculation occurs at the beginning of the simulation process taking in consideration the time to freeze the ice and maintaining the ice surface in different conditions. Equation (2.1) represents the equation used to calculate the freezing capacity.

$$q_{freezing} = C_{loss} * \rho * V_w * \frac{(C_{p,w}T_w) + q_{fusion} + \left(C_{p,ice}(0 - T_{set})\right)}{H_{freeze} * SecInHour} \tag{2.1}$$

Where,

- $q_{freezing}$, is the capacity of the refrigeration system. (J/s; Btu/s)
- $C_{p,w}$ is the specific heat of water. (kJ/(kg.K);Btu/lb.°F)
- q_{fusion}, is the latent heat of the freezing water. (kJ/Kg; Btu/lb)
- $C_{p,w}$, is the specific heat of ice. (kJ/(kg.K);Btu/lb.°F)
- T_{set}, is the ice surface temperature setpoint. (°C; °F)
- H_{freeze}, is the hours to freeze the water. (hours)
- C_{loss}, is a correction coefficient to account for any thermal losses.
- $SecInHour$, number of seconds in an hour.

- V_w, volume of water in the ice rink. (m^3;ft^3)

To obtain the maximum refrigerant mass flow rate, equation (2.2) is used:

$$\dot{m}_{ref,max} = \frac{q_{freezing}}{C_{p,refrig} * \Delta T} \quad (2.2)$$

Where,

- $\dot{m}_{ref,max}$, is the maximum refrigerant mass flow rate. (Kg/s; lb/s)
- $C_{p,refrig}$ is the specific heat of the refrigerant. (kJ/(kg.K);Btu/lb.°F)
- ΔT, is the difference between the refrigerant inlet and outlet design temperatures. (°C; °F)

Figure 2.1 shows the inputs and outputs of the freezing function. This function requires the heat exchanger pipe length, diameter and spacing. These inputs are all entered by the user when building the model.

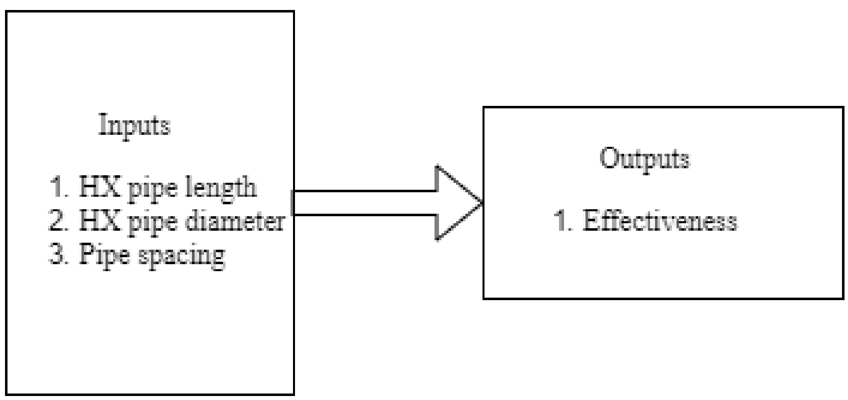


Figure 2. 1: Freezing function inputs and outputs

2.2 Effectiveness (ε)

As mentioned in chapter I, heat exchanger tubes are commonly placed in the chilled concrete layer under the water layer to freeze it. The ice rink module is using effectiveness-NTU heat exchanger approach [16-18]. The heat transfer between the water/ice and the refrigerant flowing within the heat exchanger tubes can be determined from the following equation (2.3):

$$q = \dot{m}_{ref} C_{p,refrig} (T_{refrig,i} - T_{refrig,o}) \tag{2.3}$$

Where,

- q is the actual heat transfer between the refrigerant and water/ice. (J/s; Btu/s)
- $\dot{m}_{ref}$ is the refrigerant mass flow rate. (Kg/s; lb/s)
- $T_{refrig,i}$ is the inlet temperature of the refrigerant. (°C; °F)
- $T_{refrig,o}$ is the outlet temperature of the refrigerant. (°C; °F)

According to the second law of Thermodynamics, the maximum heat transfer can be obtained from the following equation (2.4) [1,16-18]:

$$q_{max} = \dot{m}_{ref} C_{p,refrig} (T_{refrig,i} - T_s) \tag{2.4}$$

Where,

- q_{max}, is the maximum possible heat transfer. (J/s; Btu/s)
- T_s, is the temperature at the source location. In the case of the ice rink, the source location is the construction layer within which the heat exchanger pipes are located. (°C; °F)

The effectiveness of the heat exchanger, ε, is the ratio of the actual heat transfer rate to the maximum rate as shown in equation (2.5).

$$\varepsilon = \frac{q}{q_{max}} \tag{2.5}$$

Also, it can be calculated using the number of transfer units, NTU, using equation (2.6):

$$\varepsilon = 1 - e^{-NTU} \tag{2.6}$$

The NTU term can be expressed as follows in equation (2.7):

$$NTU = \frac{\pi k Nu_D L N_c}{\dot{m}_{ref}} \tag{2.7}$$

Where,

- k is conductivity of the refrigerant. (W/(m·K); Btu · in/h·ft^2 ·°F)
- L is the length of the heat exchanger tube. (m; ft)
- N_c is the number of circuits. (dimensionless)
- Nu_D is the Nusselt number. (dimensionless)

In this model, it is assumed that the heat exchanger tubes are installed parallel to the shorter side of the ice rink floor which is its width dimension. Therefore, the number of circuits N_c can be calculated using the following equation (2.8).

$$N_c = \frac{L_{rink}}{D+S} \tag{2.8}$$

Where,

- L_{rink}, is the length of the ice rink. (m; ft)
- S, is the spacing between heat exchanger pipes. (m; ft)

For the refrigerant flowing inside the tubes, the Nusselt number can be calculated using the following equations depending on the flow regime:

For laminar internal flow (Reynolds number $(Re_D) < 2300$) :

$$Nu_D = \frac{hD}{k} = 3.66 \tag{2.9}$$

Where,

D is the tube diameter. (m; ft)

h is the convection coefficient and can be obtained from the flow correlations between the Nusselt number and flow properties.

For turbulent internal flow ($Re_D \geq 2300$):

$$Nu_D = 0.23 * Re_D^{\frac{4}{5}} * P_r^{1/3} \quad (2.10)$$

Where P_r is the Prandtl number of the refrigerant and can be calculated from equation (2.11):

$$\mathrm{P_r} = \frac{\mu C_p}{k} \quad (2.11)$$

Where μ is the viscosity of refrigerant (mPa·s ;lb/ft·s).

The Reynolds number associated to the refrigerant flow inside the tubes can be calculated using the following equation (2.12):

$$Re_D = \frac{4.0\dot{m}_{ref}}{\pi\mu D * N_c} \quad (2.12)$$

Figure 2.2 shows the inputs needed for the ice rink feature to calculate the effectiveness of the heat exchanger. For EnergyPlus to calculate the effectiveness, the user must enter the heat exchanger pipe length, diameter an spacing.

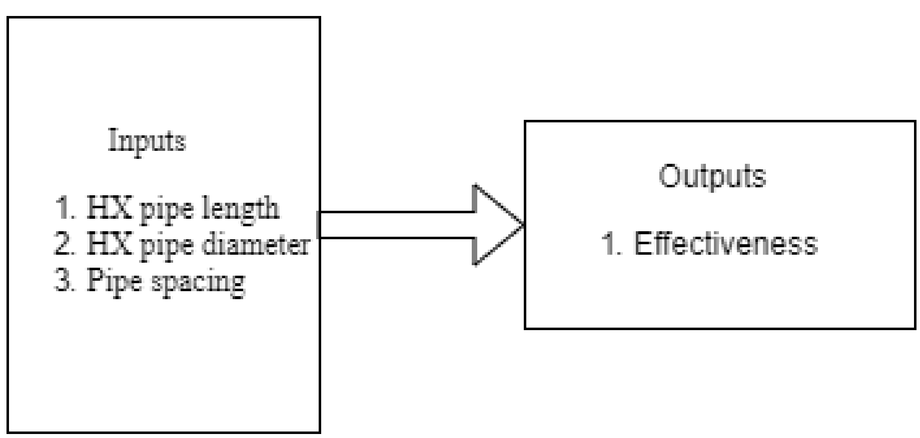


Figure 2. 2: Input and output data required for the Effectiveness calculation algorithm

2.3 Ice Rink load Calculation

EnergyPlus calculates heat loads through heat balance equations performed on the inside and outside of each building surface [19]. The heat balance on the outside surface follows this equation (2.13) [19]:

$$q"_{\alpha sol} + q"_{LWR} + q"_{conv} - q"_{ko} = 0 \tag{2.13}$$

Where,

- $q"_{\alpha sol}$ is the absorbed direct and diffuse solar radiation heat transfer (short wavelength). (J/s; Btu/s)
- $q"_{LWR}$ is the net long wavelength radiation heat exchange with air and surrounding. (J/s; Btu/s)
- $q"_{conv}$ is the convective heat transfer with outside air. (J/s; Btu/s)
- $q"_{ko}$ is the conduction heat flux through the surface. (J/s; Btu/s)

As for the heat balance on the inside surface, it is calculated from equation (2.14) [19]:

$$q"_{LWX} + q"_{SW} + q"_{LWS} + q"_{ki} + q"_{sol} + q"_{conv} = 0 \tag{2.14}$$

Where,

- $q"_{LWX}$ is the net longwave radiant heat transfer between zone surfaces. (J/m^2; Btu/ft^2)
- $q"_{SW}$ is the net short wave radiation flux to surface lights. (J/m^2; Btu/ft^2)
- $q"_{conv}$ is convective heat transfer. (J/m^2; Btu/ft^2)
- $q"_{sol}$ is transmitted solar radiation flux absorbed at surface. (J/m^2; Btu/ft^2)
- $q"_{ki}$ is the conduction heat transfer through wall. (J/m^2; Btu/ft^2)
- $q"_{LWS}$ is the longwave radiation heat transfer from equipment in a zone. (J/m^2; Btu/ft^2)

Like the low temperature radiant system, the ice rink module calculates the refrigeration load necessary to maintain the ice rink floor at the desired temperature setpoint [16-18]. Once the heat balance equations are updated with the refrigeration load, the ice rink module reads the surface temperature from the previous time step and use it in the current time step calculations.

In the case of an ice rink, the conduction part of the heat balance is modified to accommodate for the presence of heat sink as indicated by Eq. (2.15).

$$q"_{i,t} = \sum_{m=1}^{M} X_m T_{i,t-m+1} - \sum_{m=1}^{M} Y_m T_{o,t-m+1} + \sum_{m=1}^{k} F_m \, q"_{i,t-m} + \sum_{m=1}^{M} W_m \, q_{source,t-m+1} \tag{2.15}$$

Where,

- $q"$, is the surface heat balance including radiation from other surfaces, solar radiation and convection heat transfer inside the surface. (J/m^2; Btu/ft^2)
- X_m, is the inside CTF coefficient, m = 0,1,…,M
- T, is the surface temperature. (°C; °F)
- Y_m, is the cross CTF coefficient, m = 0,1,…,M.
- F_m, is the flux CTF coefficient, m=0,1,…,k.
- W_m, is the QTF inside term for the heat source/sink, m=0,1,…,M.
- i, refers to inside.
- t, is the current time step.
- o, refers to the outside.

The coefficients in Eq. (15) are calculated from the material properties of the surface layers including, thickness, density, specific heat, and conductivity.

There are two algorithms used to model an ice rink. Heat fluxes and temperatures through the heat sinks can be obtained from the first algorithm. As for the second algorithm, it calculates the temperatures at the heat sink. By arranging the equation (2.15), the inside and outside surface temperatures at a time step can be obtained from Eq. (2.16) and Eq. (2.17).

$$T_{i,t} = -\frac{1}{X_1}\sum_{m=2}^{M} X_m T_{i,t-m+1} + \frac{1}{X_1}\sum_{m=2}^{M} Y_m T_{o,t-m+1} + -\frac{1}{X_1}\sum_{m=1}^{k} F_m\, q"_{i,t-m} + \frac{q"_{i,t}}{X_1} - \frac{1}{X_1}\sum_{m=1}^{M} W_m\, q_{src,t-m+2} + \left(\frac{Y_1}{X_1}\right) T_{o,t} + \left(\frac{W_1}{X_1}\right) q_{src,t} \tag{2. 16}$$

$$T_{o,t} = \frac{1}{Y_1}\sum_{m=2}^{M} X_m T_{i,t-m+1} + \frac{1}{Y_1}\sum_{m=2}^{M} Y_m T_{o,t-m+1} - \frac{q"_{i,t}}{Y_1} + \frac{1}{Y_1}\sum_{m=1}^{k} F_m\, q"_{i,t-m} + \frac{1}{Y_1}\sum_{m=1}^{M} W_m\, q_{src,t-m+1} + \left(\frac{X_1}{Y_1}\right) T_{i,t} + \left(\frac{W_1}{Y_1}\right) q_{src,t} \tag{2.17}$$

Where,

- q_{src}, heat transfer from the heat source/sink.

The equations above can be simplified into Eq. (2.18) and Eq. (2.19) with the source temperature, T_s, can be obtained from Eq. (2.20):

$$T_i = C_a + C_b T_o + C_c q_{source} \tag{2.18}$$

$$T_o = C_d + C_e T_i + C_f q_{source} \tag{2.19}$$

$$T_s = C_g + C_i T_i + C_j T_o + C_h q_{source} = C_k + C_l q_{source} \tag{2.20}$$

Where,

- T_s, is the temperature at source (°C; °F)
- C_a, is the coefficients associated with the heat balance at the inside surface including solar, long wave radiation heat exchange and conduction history terms. (°C; °F)
- C_b, is the current cross Conduction Transfer Function term.
- C_c, is the coefficients associated with the current heat source/sink. (m^2 °C/W; ft^2 h °F/Btu)

- C_d, is the coefficients associated with the heat balance at the outside surface including solar, long wave radiation exchange, and conduction history terms. (°C; °F)
- C_e, is the current cross Conduction Transfer Function term.
- C_f, is the coefficients associated with the current heat source/sink. (m^2 °C/W; ft^2 h °F/Btu)
- C_g, is the sum of temperature and source history terms at the source/sink location. (°C; °F)
- C_h, is the coefficients associated with the current heat source/sink. (m^2 °C/W; ft^2 h °F/Btu)
- C_i, is the inside term for the current inside surface temperature.
- C_j, is the CTF outside term for the current outside surface temperature.

Another two coefficients that are used in the heat sink load equations are $C_k\ and\ C_l$ which are defined as follows:

$$C_k = \frac{C_g + C_i(C_a + C_b C_d) + C_j(C_d + C_e * C_a)}{1 - C_e C_b} \tag{2.21}$$

$$C_l = \frac{C_h + C_i(C_c + C_b C_f) + C_j(C_f + C_e C_c)}{1 - C_e C_b} \tag{2.22}$$

The ice rink module utilizes the CTF coefficients in the following equations (2.23-2.27) to obtain the required refrigeration load to maintain the operation of the ice rink. Specifically, $q_{setpoint}$ is the refrigeration load required to bring the surface temperature T_i to the setpoint temperature T_{set} and can be obtained from Eq. (2.23):

$$q_{setpoint} = \frac{1 - (C_b * C_e) * T_{set} - C_a - (C_b * C_d)}{C_c + C_d C_f} \tag{2.23}$$

$\dot{m}_{ref,req}$ is the required mass flow rate of refrigerant to maintain T_{set} can be calculated from equation (24):

$$\dot{m}_{ref,req} = \frac{q_{setpoint}}{\varepsilon \mathrm{C_p}(T_{referig,in} - T_{src})} \tag{2.24}$$

The following Eq. (25) is used to calculate the maximum heat flux from the refrigerant q_{max}:

$$q_{max} = \dot{m}_{ref} C_{p,refrig}(T_{refrig,in} - T_{src}) \tag{2.25}$$

The following equation (26) calculates q_{src} which is the heat sink load that is used in updating the conduction part of the heat balance equations [i.e., Eq. (13) and Eq. (14)].

$$q_{src} = \frac{\varepsilon \dot{m}_{ref} \mathrm{C_p} \left(T_{refrig,in} - C_k\right)}{1.0 + \frac{\varepsilon \dot{m}_{ref} \mathrm{C_p} C_l}{Ap}} \tag{2.26}$$

Where,

$Ap,$ is the total surface area of the heat exchanger tubes and can be calculated using the following Eq. (27):

$$Ap = \pi * L * D * N_c \tag{2.27}$$

Figure 2.3 shows different inputs required by the ice rink load function to calculate the refrigeration load. Some of these inputs are calculated are obtained from EnergyPlus simulation and the rest from the user inputs. CTFs, effectiveness, T_i^{t-1}, T_{set} and $\dot{m}$ are examples of inputs obtained from other functions in EnergyPlus. On the other hand, user inputs, $T_{refrig,in}$, spacing and heat exchanger pipe length.

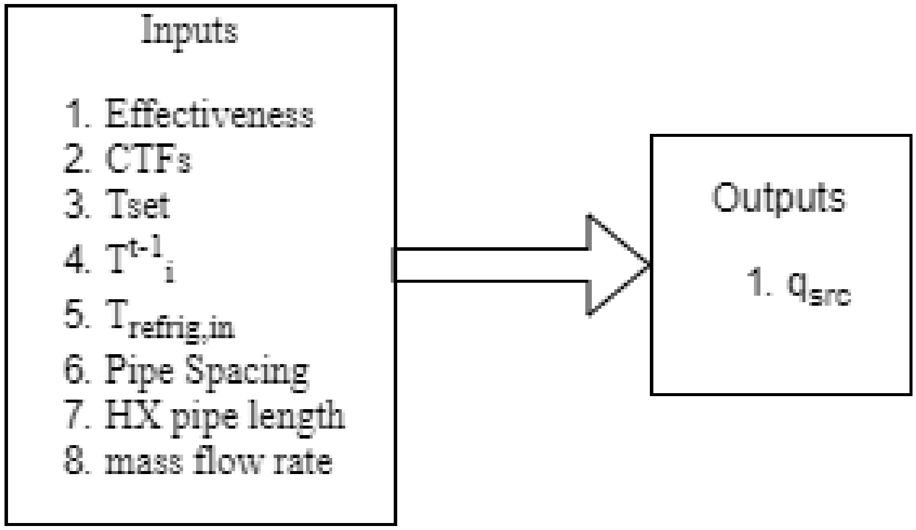


Figure 2. 3: Ice Rink Refrigeration Load inputs and outputs

2.4 Control Strategies:

2.4.1 Surface Temperature Control

Different ice rink activities require different ice temperatures. For example, skating requires the ice to be soft and gripping. On the other hand, ice hockey players prefer hard ice surface to allow for speeding. Hence, ice is maintained at lower temperatures for ice hockey compared to skating activities. Surface temperature control is considered the most effective and it is used in most of the new ice rinks [1]. In Fig. 2.4, if the ice rink is operating then operation flag is set 1 otherwise it is zero. If the ice temperature of the previous time step is lower than the ice setpoint temperature, then no refrigeration is required and hence no mass flowrate is needed in the heat exchanger tubes. However, when the ice temperature of the previous time step is higher than the ice setpoint temperature, the refrigeration system is operated and refrigerant flows in the heat exchanger tubes. The next step, the heat transfer required to maintain the ice setpoint and the maximum heat transfer from the refrigerant are calculated using Eq. (23) and Eq. (25). If the maximum heat flux is more than the required heat flux, then the mass flowrate is set to the required mass flowrate calculated from Eq. (24). Otherwise, the mass flowrate is set to the maximum allowed.

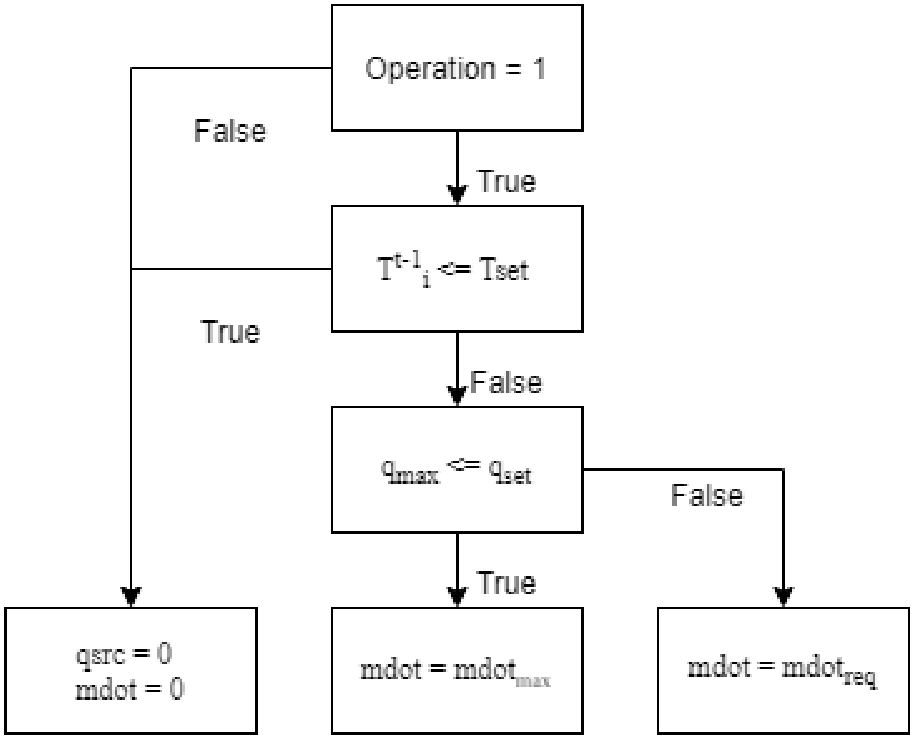


Figure 2. 4: Ice Surface Temperature Control Algorithm

2.4.2 Refrigerant Temperature Control

Refrigeration systems use primary and secondary refrigerants (coolants). Secondary refrigerants transport heat from the medium being cooled to the heat exchanger tubes. In the case of the ice rink, the medium being cooled or frozen is water. In the heat exchanger, the primary refrigerant absorbs heat from the secondary refrigerant. There are different types of secondary coolants used such as calcium chloride, glycol, and brine. Refrigerant temperature control algorithm is another method to control ice rink refrigeration systems. This strategy controls the secondary refrigerant outlet temperature which is the temperature of the refrigerant leaving the ice rink and flowing back to the heat exchanger. If the refrigerant outlet temperature of the previous time step, $T_{refrig,out}^{t-1}$, is lower than the refrigerant outlet setpoint, T_{set}, then the refrigerant mass flowrate is set to the minimum and refrigeration system is turned off. In addition, the inlet temperature of the refrigerant is set equal to the refrigerant temperature of the previous time step. If the refrigerant temperature of the previous time step is higher than the refrigerant temperature setpoint, the required mass flowrate is first compared to the maximum mass flowrate. If the required mass flowrate is higher, then the refrigerant mass flowrate is set equal to the maximum mass flowrate that can handled by the refrigeration system. Otherwise, the mass flowrate is set equal to the required mass flowrate. The refrigerant outlet temperature can be calculated by the following Eq. (28) [1]:

$$T_{refrig,out} = T_{b,in} - \frac{q_{src}}{\dot{m}C_{p,refrig}} \tag{2.28}$$

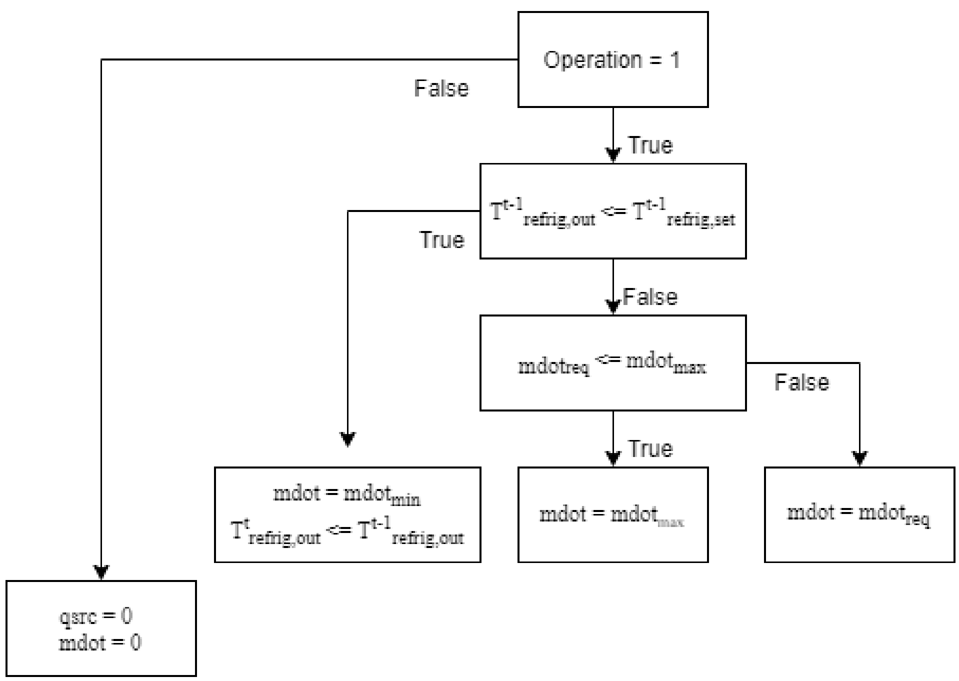


Figure 2. 5: Refrigerant Temperature Control Algorithm

2.5 Resurfacing Loads

As mentioned in the literature review, the resurfacer machine helps maintaining the ice surface quality by washing and shaving it. The resurfacer operation involves the utilization of warm water which causes part of the ice rink surface to melt. Therefore, the ice rink refrigeration system needs to account for the additional load caused by the resurfacer which can be determined using the following equation [13]:

$$Q_{resurf} = \frac{\rho_w V_{tank}(4.2\ T_{resurf}) + Q_{fusion} - (2.0\ T_i)}{SecInHour} \tag{2.29}$$

Where,

- Q_{resurf}, is the heat load caused by the resurfacing flood. (J/m^2; Btu/ft^2)
- V_{tank}, is the resurfacer machine tank capacity. (m^3;ft^3)
- T_{resurf}, is the water temperature of the resurfacer. (°C; °F)

The resurfacer water heating load can be calculated from the following equation:

$$E_{heating} = V_{tank}\rho_w C_p(T_{resurf} - T_{w,init}) \tag{2.30}$$

Where,

- $E_{heating}$, heating capacity to raise the water temperature used by the resurfacer. (J; Btu)
- $T_{w,init}$, the initial water temperature in the resurfacer tank. (°C; °F)

Figure 2.6 represents the inputs needed by the resurfacer equations. All the resurfacer related inputs are entered by the user. The required inputs are the resurface tank volume, the hot water temperature, the ice temperature, and initial water temperature.

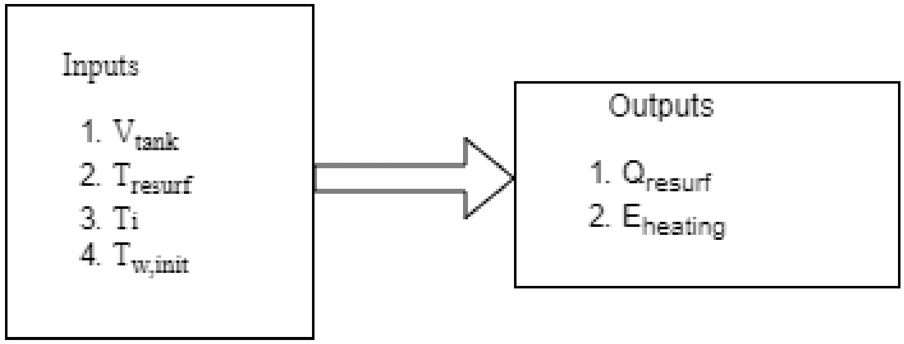


Figure 2. 6: Ice rink resurfacer equations inputs and outputs

Figure 2.7 shows the operation of the resurfacer in the ice rink code in EnergyPlus. If the ice rink is not operational then there is no need for resurfacing. For example, if the ice rink is set not to work in summer then the resurfacer will be off as well. When the ice rink is on, the resurfacer will run only based off its schedule. Once the resurfacer is on, EnergyPlus will calculate the energy needed to heat water and the heating load on the ice surface.

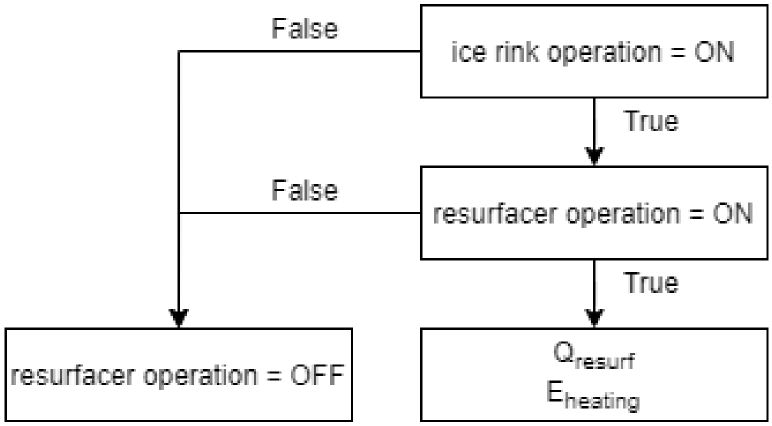


Figure 2. 7: Resurfacer algorithm in the ice rink feature

Figure 2.8 shows the effect of the resufacer operation on the ice rink surface temperature. Figure 2.8 shows the surface is maintained at -3°C and when the resurfacer is schedule to operate the ice rink surface temperature increases. Once the resurface operation is off, the refrigeration system brought down the surface temperature to its desired operational setpoint.

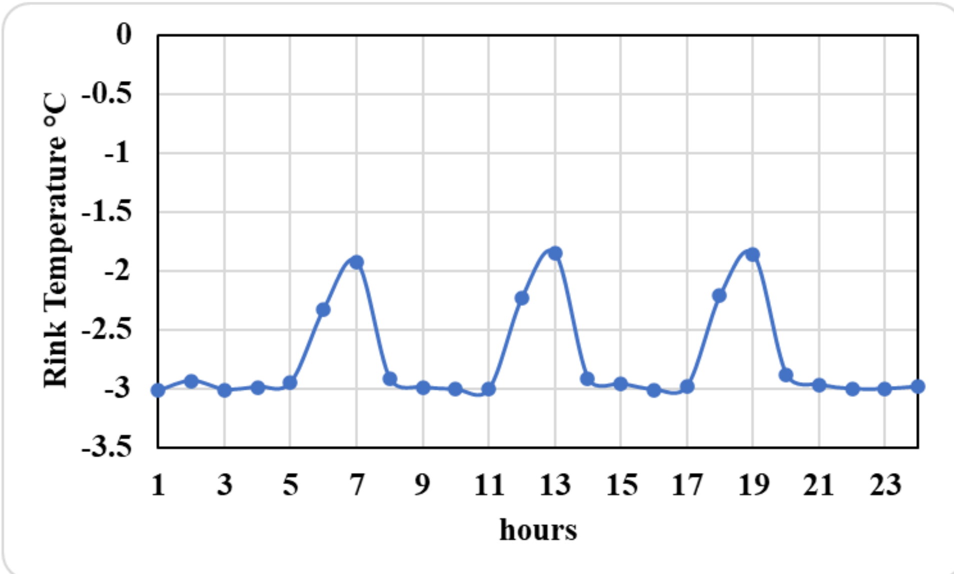


Figure 2. 8: Effect of the resurfacer use on ice rink surface temperature during a typical day

2.6 Input Data File:

This section describes the different inputs of the ice rink module as implemented in EnergyPlus input data file (IDF) [20]. In addition, this section outlines other necessary input data required to successfully run a simulation that includes an ice rink model.

Ice Rink Input Data

The following are the input variable of the new EnergyPlus ice rink module (object):

- Name: This field is a unique user assigned name for the ice rink object. It is used by other objects to reference this function.
- Availability Schedule Name: This field represents the name of the schedule of the ice rink operation. Acceptable values are 0 and 1 where 0 means that ice rink is off and 1

means it is on. EnergyPlus will not accept any other values and the default is 1 indicating that the ice rink is set to operate all the time.

- Zone Name: This field is the name of the thermal zone where the ice rink is located. Any building modeled in EnergyPlus must have at least 1 thermal zone to run the simulation.
- Surface Name: This field refers to the surface where the ice rink is assigned to. This surface is where the heat exchanger tubes are embedded. Unlike other similar objects such as low temperature radiant systems, the ice rink feature will accept only one surface.
- Tube Diameter: This field refers to the diameter of the secondary coolant heat exchanger tubes. These tubes are embedded in the surface that the ice rink is assigned to and are where ice rink refrigerant flows. The tube diameter is important and is used in different equations that are related to refrigeration load calculations such as effectiveness and circuits number.
- Tube Length: This field refers to the length of the heat exchanger tubes. However, this is not the total length of the tube.
- Ice Rink Control Strategy: This fields is to assist the user to specify the control strategy of the refrigeration system. Currently, two inputs are acceptable: either BTOC (Brine Temperature Outlet Control) and STC (Surface Temperature Control). The BTOC strategy controls the operation of the refrigeration system by adjusting the outlet temperature of the refrigerant. As for STC, it is controlling the refrigeration system by monitoring the surface temperature of the ice rink.
- Hours to freeze water: This field is the desired number of hours needed to freeze water to become ice at the start of the ice rink operation. The number of hours, entered by the user, is used to calculate the needed capacity for the ice rink's refrigeration system.

- Delta Temperature: This field represents the desired temperature difference between inlet and output of the refrigerant. This value is used to calculate the refrigeration system's capacity during the design calculations.
- Refrigerant Inlet Node Name: This field represents the name of the inlet node to the heat exchanger tubes embedded in the ice rink floor. Since the ice rink is part of a plant loop, this name is used in the branch description to define the demand side of the plant.
- Refrigerant Outlet Node Name: This field represents the name of the outlet node to the heat exchanger tubs embedded in the ice rink floor. This name is used in the branch description to define the demand side of the plant.
- Ice Rink Design Setpoint: This field represents the lowest setpoint temperature for the ice rink floor. This value is used to calculate the refrigeration capacity of the ice rink. This value does not represent the ice rink operation setpoint temperature instead it represents the worst-case scenario. For example, the refrigeration system can be sized to maintain the ice surface at a design setpoint of -7°C as a worst case scenario. Depending on the activity, the ice surface operational setpoint can be higher like -3°C, -4°C or -5°C.
- Rink Length: This field represents the ice rink length.
- Rink Width: This field represents the ice rink width.
- Water temperature: This field refers to the water temperature within the rink before freezing to ice. The water temperature is used in the design calculations to determine the refrigeration system capacity.
- Ice Thickness: This field represents the maximum ice thickness in the ice rink. This value is used in the design calculations to estimate the refrigeration capacity. The module only

accepts positive values. If no values entered, the default value is 0.0254m (1inch) which is the typical ice rink thickness.

- COP: This field represents the coefficient of performance of the refrigeration system. This field is used for static simulation of a chiller that cools the secondary coolant. However, this field is not required when a chiller is modeled as part of the plant. The default value of this field is 2.5.
- Ice Rink setpoint temperature: The field refers to the desired setpoint of the ice rink floor. This setpoint represents the operational setpoint temperature that the refrigeration system needs to maintain. This value cannot be higher than 0°C (32°F) and the default is -3°C (26.6°F).
- Ice Rink HX spacing: This field refers to the spacing between the heat exchanger tues embedded in the floor. This value is required to calculate the number of parallel circuits and is used to estimate the effectiveness of the refrigeration system. For the ice rink application, the tube spacing is typically on average 100 mm from the center of the tubes [13].

An EnergyPlus model can be built using a notepad file or the IDF editor. The IDF editor lists all available inputs for each object. Adding the ice rink to the list of objects was part of a new feature development in EnergyPlus. Figure 2.9 is a screenshot of the ice rink object in the EnergyPlus IDF. It shows the inputs discussed before and required to model an ice rink in EnergyPlus.

Field	Units
Name	
Availability Schedule Name	
Zone Name	
Surface Name	
Tube Diameter	m
Tube Length	m
Ice Rink Control Strategy	
Hours to freeze the water	hr
delta temperature	C
Refrigerant Inlet Node Name	
Refrigerant Outlet Node Name	
Refrigerant Outlet Temperature Schedule Name	
Ice Surface Temperature Schedule Name	
Unused	
Unused	
design setpoint	C
Rink Length	m
Rink Width	m
Water Temperature	C
Ice Thickness	m
COP	dimensionless
Ice Rink Setpoint Temperature	C
Ice Rink HX spacing	m

Figure 2. 9: Ice rink feature inputs

Construction:Internal Source Inputs:

In EnergyPlus, typical building envelope components such as floor, walls and roofs are modeled as construction objects [20]. However, the ice rink floor is modeled differently. Like the low temperature radiant system, the ice rink floor is modeled using the EnergyPlus object called Construction:Internal Source. This object is used when modeling surfaces with source/sink components. Not all the fields in the Construction:Internal Source object are needed to model the ice rink. Figures 2.10 & 2.11 shows the object location in the IDF and its inputs. The following are the required inputs to successfully run the simulation:

- Name: this field is the name assigned to the ice rink surface. It should be the surface name used for the ice rink surface name field.
- Source Present After Layer Number: This field represents the location of the source (i.e., heat exchanger tubes) by entering the number of the layers existing before the source. When the heat exchanger tues are embedded in a homogenous layer, the layer can be split into 2 layers.
- Outside layer: This fields refers to the layer located on the outside -opposite side of the modeled zone. In case of an ice rink located on the first floor, the outside layer should be facing the ground. According to EnergyPlus Input Output Reference [20], every construction should have at least one layer and it should not be film coefficient.

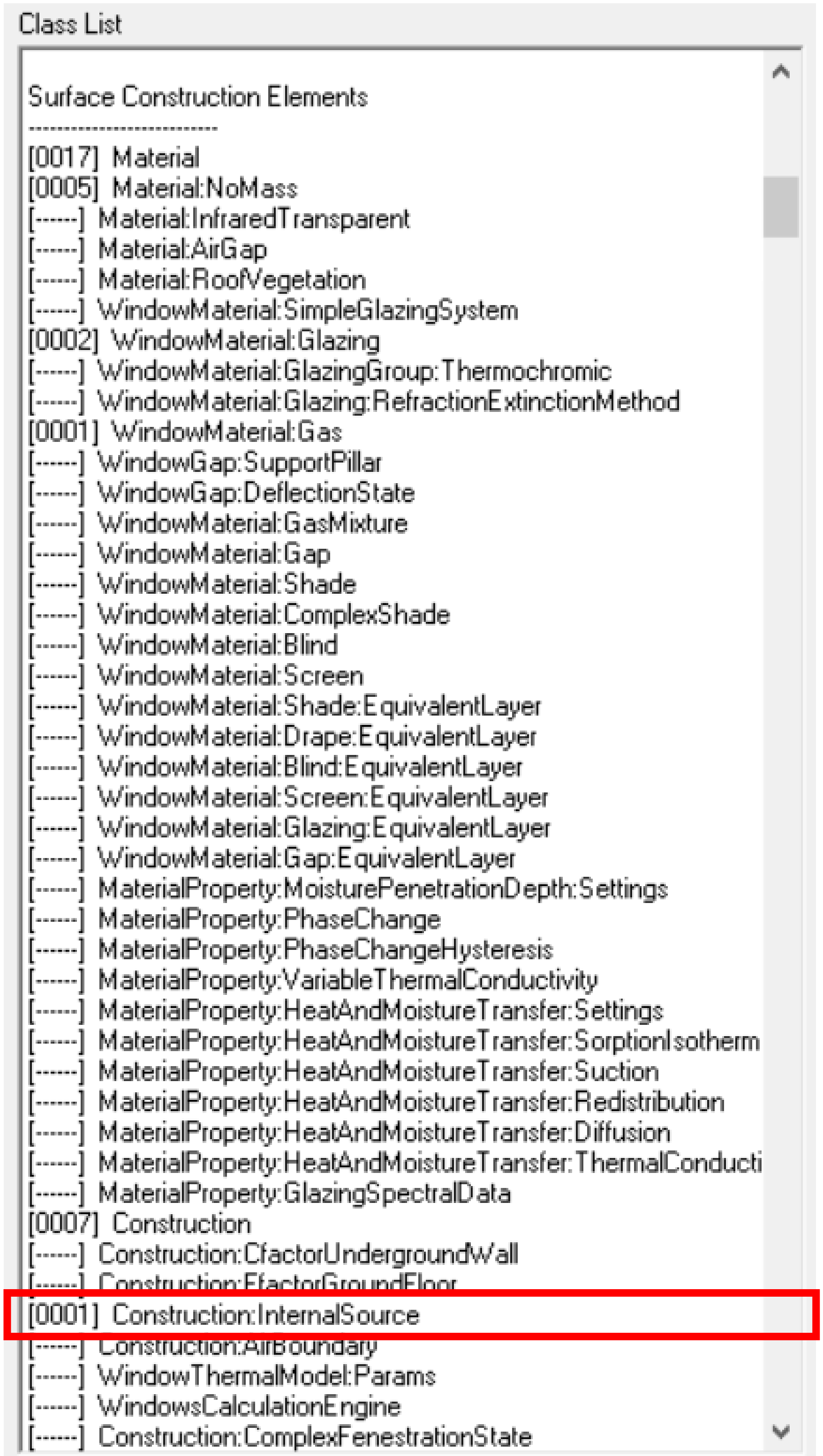


Figure 2. 10: Construction:Internal Source location in the class list

Field	Units	Obj1
Name		Slab Floor with
Source Present After Layer Number		3
Temperature Calculation Requested After Layer Numbe		3
Dimensions for the CTF Calculation		1
Tube Spacing	m	.09
Two-Dimensional Temperature Calculation Position	dimensionless	0
Outside Layer		CONCRETE - [
Layer 2		INS - EXPANDI
Layer 3		100mm -Chilled
Layer 4		10mm -Chilled [
Layer 5		Water
Layer 6		
Layer 7		
Layer 8		
Layer 9		
Layer 10		

Figure 2. 11: Construction:Internal Source inputs

Integrating an ice rink in an EnergyPlus plant:

Since the ice rink module relies on a refrigeration system, it is critical to include a chiller in the simulation. The chiller supplies the refrigerant needed for the heat exchanger tubes that maintain the ice rink at the desired setpoint temperature. To successfully run a simulation, the ice rink model needs to be integrated within a refrigeration plant. EnergyPlus models most HVAC systems using supply and demand loops. For an ice rink model, the supply loop can includes a chiller and the demand loop has the evaporator consisting of the heat exchanger tubes. If the chiller is connected to a cooling tower, another set of demand and supply loops can be added where the demand loop would be the chiller and the supply loop would be the cooling tower. To communicate between different components of a plant, EnergyPlus uses nodes to define the inlet and outlet of each component. These nodes help monitoring and calculating fluid properties at different locations of the loop. For example, EnergyPlus obtains the inlet refrigerant temperature at the inlet node of the ice rink and utilizes it in the refrigeration load calculations. Plant components are located on branches which, when connected to each other's, form a loop. Branch connections rely on mixers and splitters to connect between various components of a loop. A splitter splits the branch into one or more branches. For example, if a bypass is needed, the splitter would be used to split the refrigerant flow. On the other hand, a mixer connects different branch outlets into one branch. For example, different flows of refrigerant are combined at the entrance of the chiller. Similar to diagrams in EnergyPlus plant application guide[21], figure 2.12 represents a diagram of the ice rink supply and demand loop.

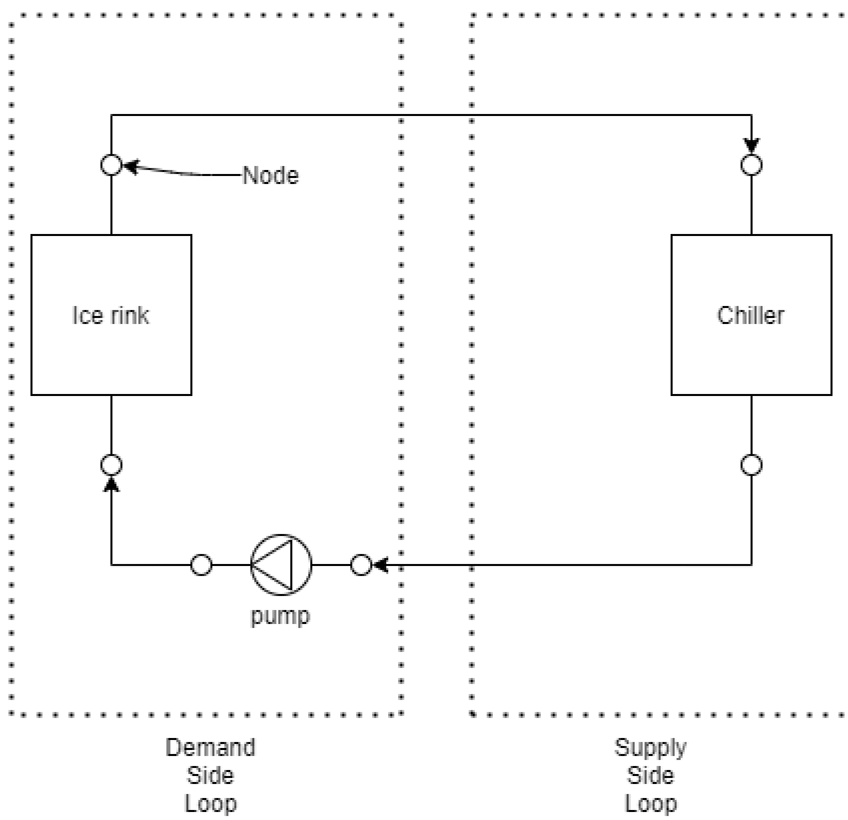


Figure 2. 12: Supply and demand side of the ice rink loop

Chapter III: Validation Analysis

3.1 Introduction

Since EnergyPlus is a building simulation program, it is necessary to make sure the ice rink yield results close to a real facility consumption. Therefore, part of the testing process was to build an EnergyPlus model that includes the ice rink feature. The model was built after an existing ice rink audit report. As discussed in the next section, some of the building information were unknown therefore realistic assumptions were made as necessary.

3.2 Description of Existing Ice Rink

The existing ice rink used for the validation analysis is located in the Old Bridge, New Jersey [22]. The ice rink facility has a total floor area of 2904 m^2 (31,250 ft^2). The ice rink size is unknown. For this analysis, it was assumed to be 30m x 60m, which corresponds to ice rink'size appropriate for hockey activities [2]. Specific thermal properties for the facility's envelope are not mentioned in the audit report including the R-values for the walls and roof. For this analysis, building envelope requirements from the 2018 International Energy Conservation Code [23] were used. Moreover, the dimensions for the windows are approximate and are based on visual inspection of the building. The ice rink facility is modeled using EnergyPlus using two thermal zones. One of the zones corresponds to the ice rink is arena. The other thermal zone represents the concession area. The main model input features for the ice rink facility used in the validation analysis is described in Table 3.1.

Table 3.1: Description of input features for the EnergyPlus model of ice rink facility used in the validation analysis

Building Component	**Value (Unit)**
Location	Old Bridge, NJ
Building dimensions	40 m x 84 m x 5 m
Thermal zones and their dimensions	Zone 1 - Ice rink arena: 40 m x 75 m Zone 2 – concession area: 9 m x 10 m
Wall constructions	Exterior walls: • 8” masonry block • R-11 • ½” Gyp board Partition wall: • ¾” Gyp board • 8” Structural clay tile • ¾” Gyp board
Roof Construction	• Metal roof frame • R-30 • ½” Gyp board
Fenestration	• Windows: U-0.65 & SHGC 0.25 • Door: Opaque Standard – U-2.1
HVAC	• Modeled as an ideal load air system.

	• Ideal load system is a simplified HVAC system that yield the required energy to condition the zone. • Ideal Load System is assumed to be 100% efficiency • The zone will be maintained at 22°C to meet indoor thermal comfort.
People	• 50 people in the ice rink zone. • 210 in the concession zone – based on the occupancy density listed in New Jersey building codes [24].
Lighting Power Density (LPD)	• 13 W/m^2 ($1.2W/ft^2$) • Schedule - Figure 3.1
Equipment Power Density	• 7 W/m^2 (0.65 W/ft^2) • Schedule - Figure 3.1
Ice rink floor	• 100mm concrete (4 in) • Insulation R-14 • 110 mm concrete • 0.0254 m of water/ice
Ice Rink inputs	• Size: 60 m x 30 m (196ft x 98.5ft) • Schedule: seasonal. Figure 3.1

	• HX pipe diameter: 0.012 m • HX length: 30 m • Control strategy: Surface Temperature Control (STC). • Ice design setpoint: -10 °C • Ice thickness: 0.0254 m • Design ice thickness: 0.0381 m • Refrigeration COP: 2.5 • Ice temperature setpoint: -3 °C • HX pipe spacing: 0.09 m

Figure 3.1 shows the different schedules for lights and equipment in the ice rink facility. This figure shows the utilization fraction. The fraction value is multiplied by the value listed in table 3.1 to get the actual usage in Watt. This schedule is the baseline schedule before starting the validation process. This value is also used in the baseline model used for the sensitivity analysis in chapter 4. The lighting and power densities in table 3.1 represent the design values and not operational values therefore the fraction does not reach 1 or 100% and instead average values were used to reflect actual consumption. Figure 3.1 shows that the lights and equipment fractions are higher when the ice rink facility is operating. Since the ice rink is not open 24/7 then the equipment fraction is assumed to be 0.5 to represent the average usage fraction per day. Like the equipment schedule, the lighting's fraction is to represent the average usage per day. However, it is slightly higher than equipment because it was assumed not all lights are turned off at night because maintenance, housekeeping, or security, etc. In summer months the fraction drops down to 0.1 to

reflect any consumption in summer due to maintenance or emergency lights, etc. Figure 3.2 shows the ice rink occupancy schedule for a typical day. Similar to the lighting and equipment schedule, figure 3.2 shows the occupancy fraction. It was assumed that the occupancy increases as well in the afternoon throughout the evening because typically events and training take place in the evening.

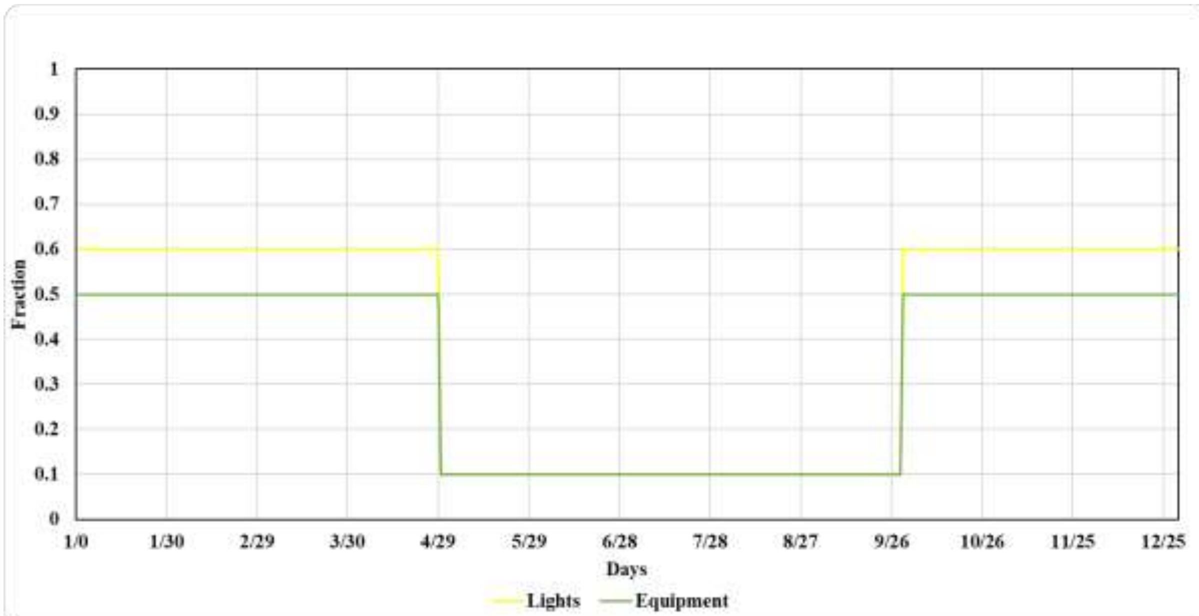


Figure 3. 1: Ice rink facility's lights and equipment schedule used before validation

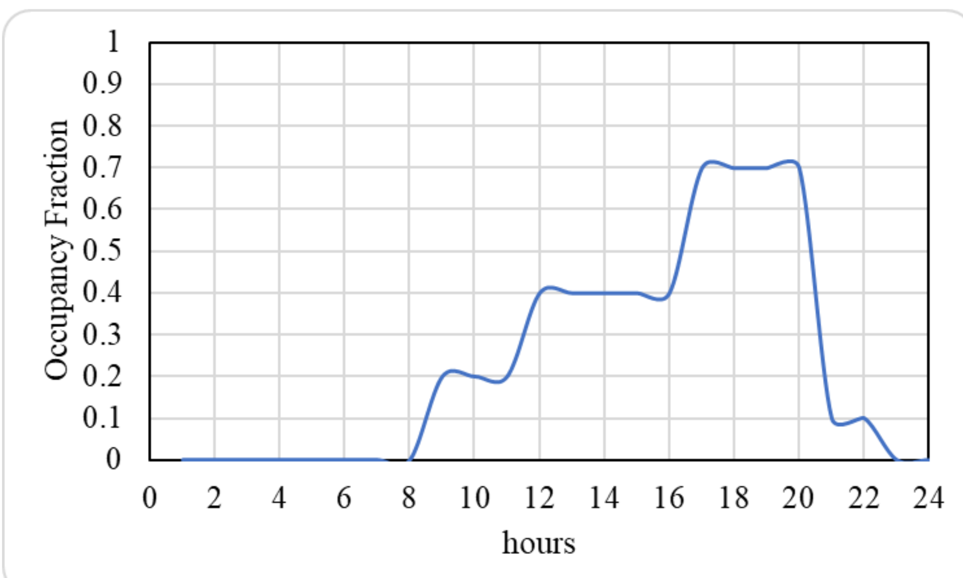


Figure 3. 2: Ice rink facility's occupancy schedule for a typical day used before validation

Figure 3.3 illustrates the model input using EnergyPlus object function (i.e., IDF editor) for the ice rink floor using the newly developed algorithm of the ice rink module.

Field	Units	Obj1
Name		IndoorIceRink
Unused		
Zone Name		Main Zone
Surface Name		Zn001:IceRink
Tube Diameter	m	0.012
Tube Length	m	30
Ice Rink Control Strategy		STC
Hours to freeze the water	hr	3
delta temperature	C	2
Refrigerant Inlet Node Name		InNode
Refrigerant Outlet Node Name		OutNode
Refrigerant Outlet Temperature Schedule Name		RefrigerantSetPtyr
Ice Surface Temperature Schedule Name		IceRink_ON
Unused		
Unused		
design setpoint	C	-10
Rink Length	m	60
Rink Width	m	30
Water Temperature	C	22
Ice Thickness	m	0.0381
COP	dimensionless	2.5
Ice Rink Setpoint Temperature	C	-3
Ice Rink HX spacing	m	0.09

Figure 3. 3: Ice rink using EnergyPlus input data file (IDF) editor

Figure 3.4 shows on/off schedule used in modeling the ice rink operation used in the baseline before starting the calibration. The schedule is set to follow a seasonal operation for the ice rink from October to April. A value of 1 means the ice rink is on and a value of 0 means it is off. This schedule is discussed in details in chapter 4 section 4.2.

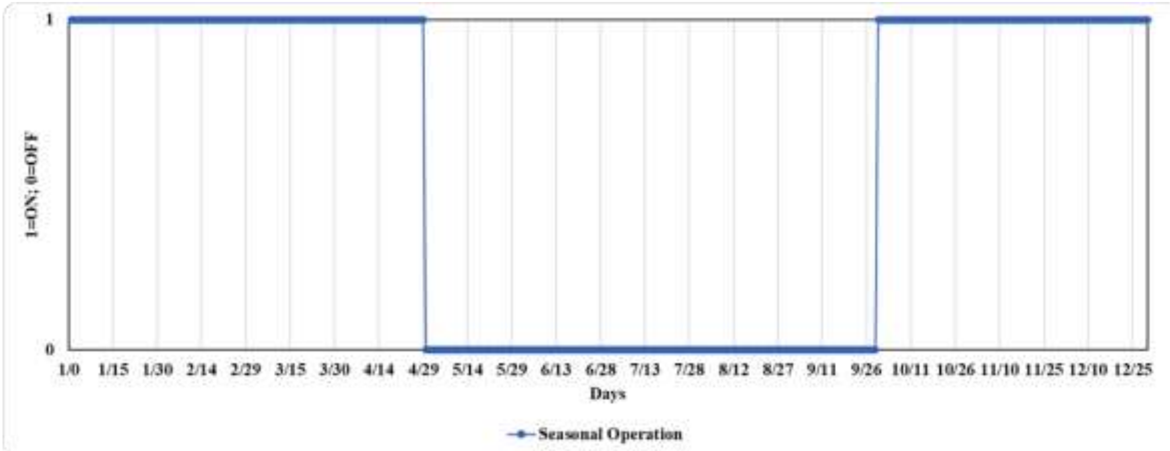
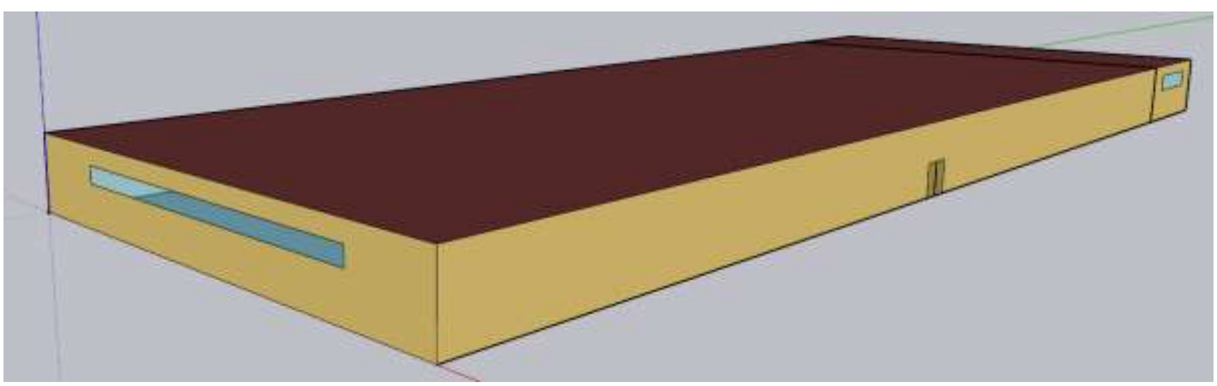
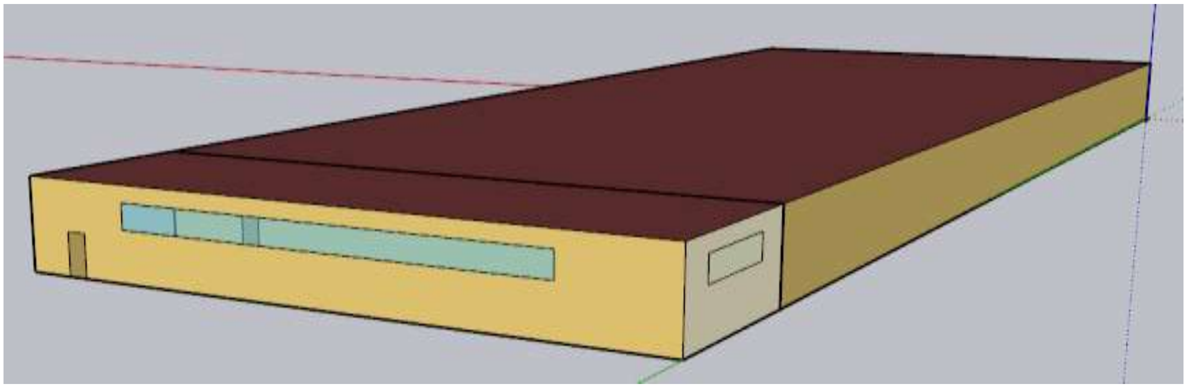


Figure 3. 4: Operation schedule for the ice rink used in the baseline

Renderings from different view angles of the modeled ice rink facility in EnergyPlus are illustrated in Figure 3.5. These 3-D rendering were obtained from SketchUp using OpenStudio plugin to help visualize the shape and the dimensions of the ice rink facility. Moreover, SketchUp was utilized to visualize the building envelope dimensions entered in EnergyPlus IDF editor as indicated in Figure3.6.



(a) South and East Façades



(b) North and West Façades

Figure 3. 5: **Renderings of the ice rink facility from various views**

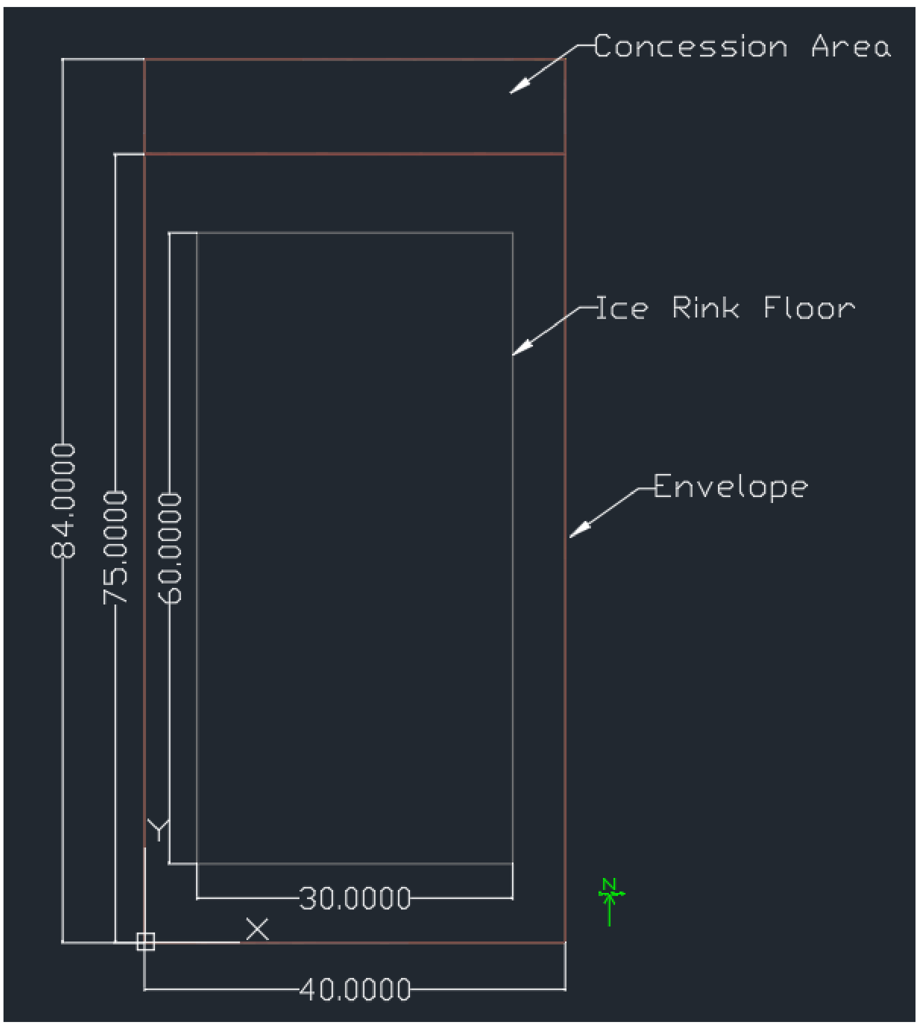


Figure 3. 6: Layout obtained from EnergyPlus CAD file (eplusout.dxf

3.3 Calibration Approach

As discussed in previous section, the EnergyPlus was built based off an existing ice rink in Old Bridge, NJ. Uniform schedules were used as discussed in previous chapter as a starting point for calibration. Once the simulation was completed, the results were significantly higher than the utility data provided in the audit report. Figure 3.7 shows the preliminary results of the simulation before the calibration process begins.

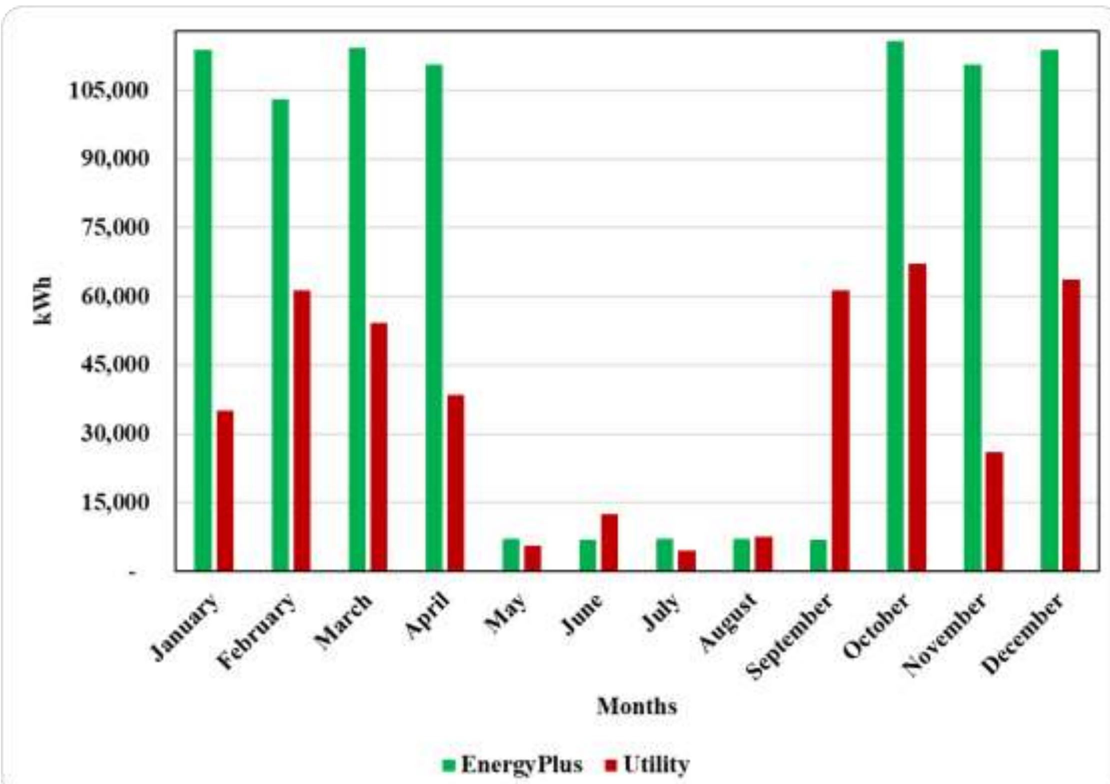


Figure 3. 7: Comparison of EnergyPlus model's predictions and utility data for monthly facility electricity consumption before validation

To calibrate the model, different schedules were modified. Since the refrigeration has a significant impact on results, the focus was on the ice rink operational schedule. Figure 3.8 represents the ice rink operational schedule after modification to reduce the energy consumption of the baseline. Lighting and equipment schedules were also modified.

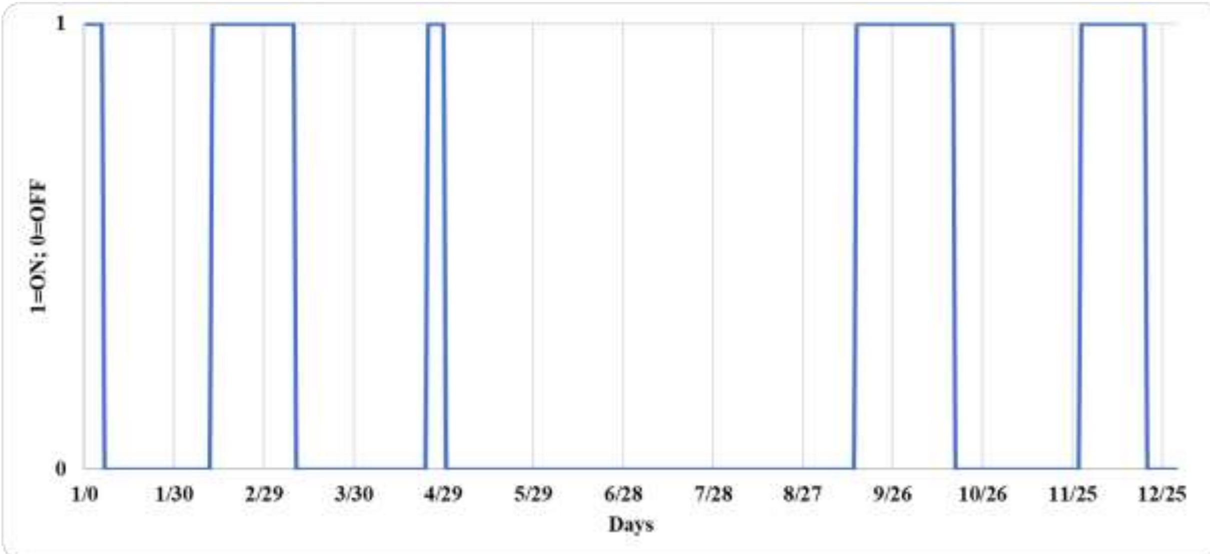


Figure 3. 8: Operation schedule for the ice rink used in the validation analysis

Fig. 3.9 compares the utility monthly electricity consumption of the ice rink facility to the EnergyPlus model predictions. EnergyPlus model predicts the facility's electricity consumption within 2% of the utility data as indicated in Figure 3.10. As expected, the electricity consumption is low during the summer period since the facility is not operational. Some discrepancies exist between utility and model predictions during operational months. For example, November and January electricity consumption is lower than expected.

The audit report did not provide a breakdown of electricity consumption. EnergyPlus model of the facility indicated that the refrigeration energy end-use represents about 40% of total electricity consumption which is close to 43% which is the average reported refrigeration end-use for a typical ice rink [10,11,12].

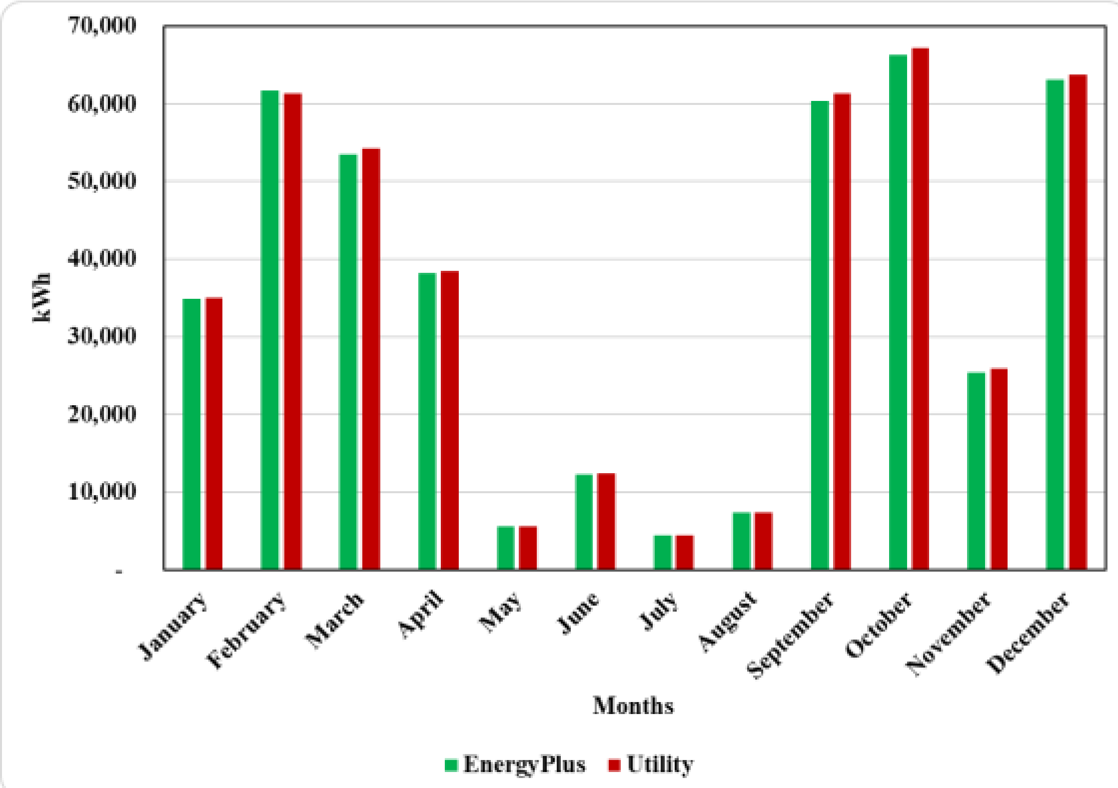


Figure 3. 9: Comparison of EnergyPlus model's predictions and utility data for monthly facility electricity consumption

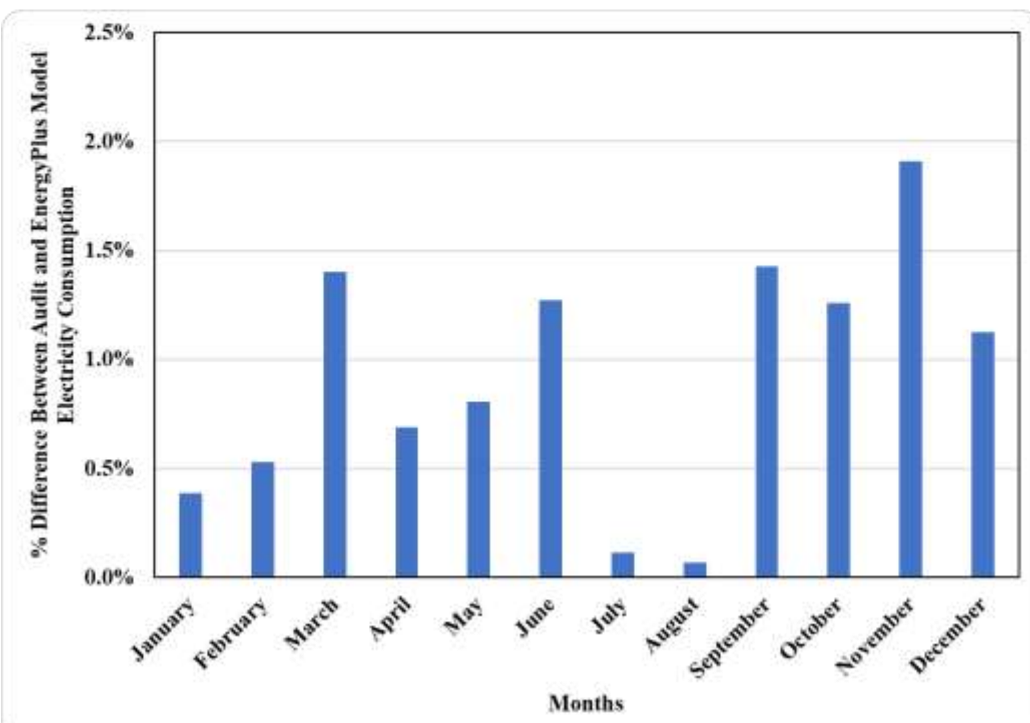


Figure 3. 10: Percent difference between EnergyPlus model's predictions and utility data for monthly electricity consumption of the audited ice rink facility

Figure 3.11 shows the natural gas consumption after calibration. The natural gas represent mainly consumption due to heating. Again there are some discrepancies in the utility data. Although the ice rink is not operating in May but yet the natural gas consumption is high. On the other hand, the ice rink should be running in October and November but yet the natural gas due to heating is low.

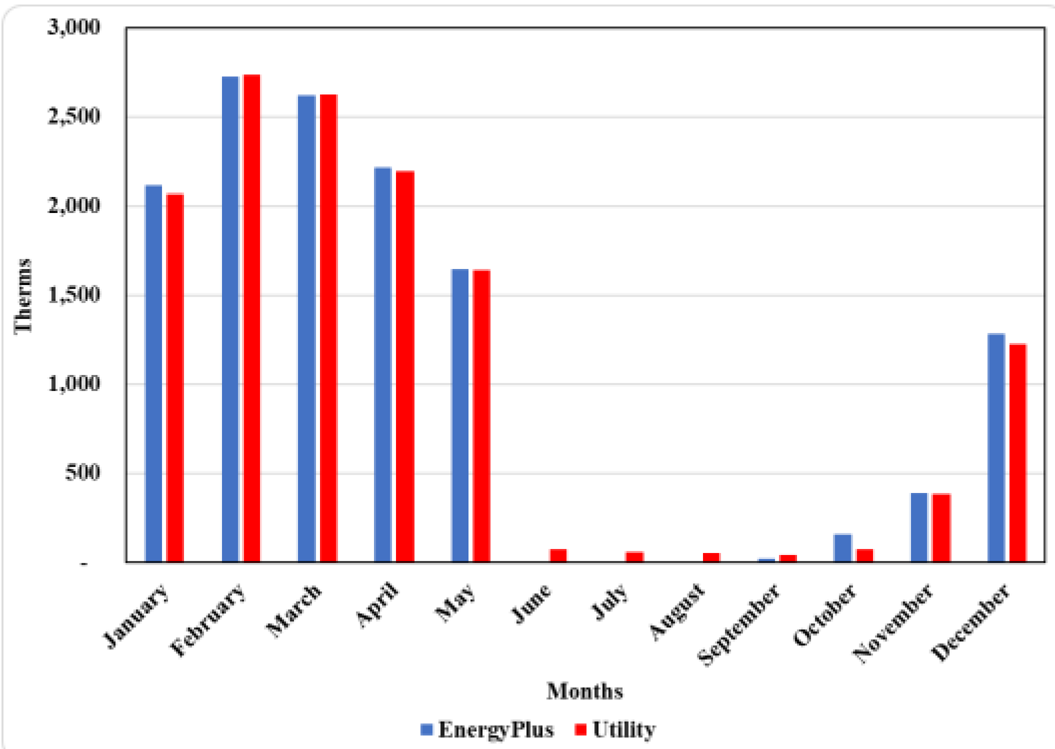


Figure 3. 11: Comparison of EnergyPlus model's predictions and utility data for monthly facility gas consumption

EnergyPlus simulation for the audited facility provided hourly data for various parameters. Example of the parameters are mean air temperature, rink floor surface temperature and refrigeration load as indicated in Figure 3.12. As noted in Table 3.1, the indoor air temperature and rink surface setpoints are set at 22°C and -3°C, respectively when the ice rink facility is occupied. Fig. 3.12 shows that both setpoints are met when the ice rink is operational.

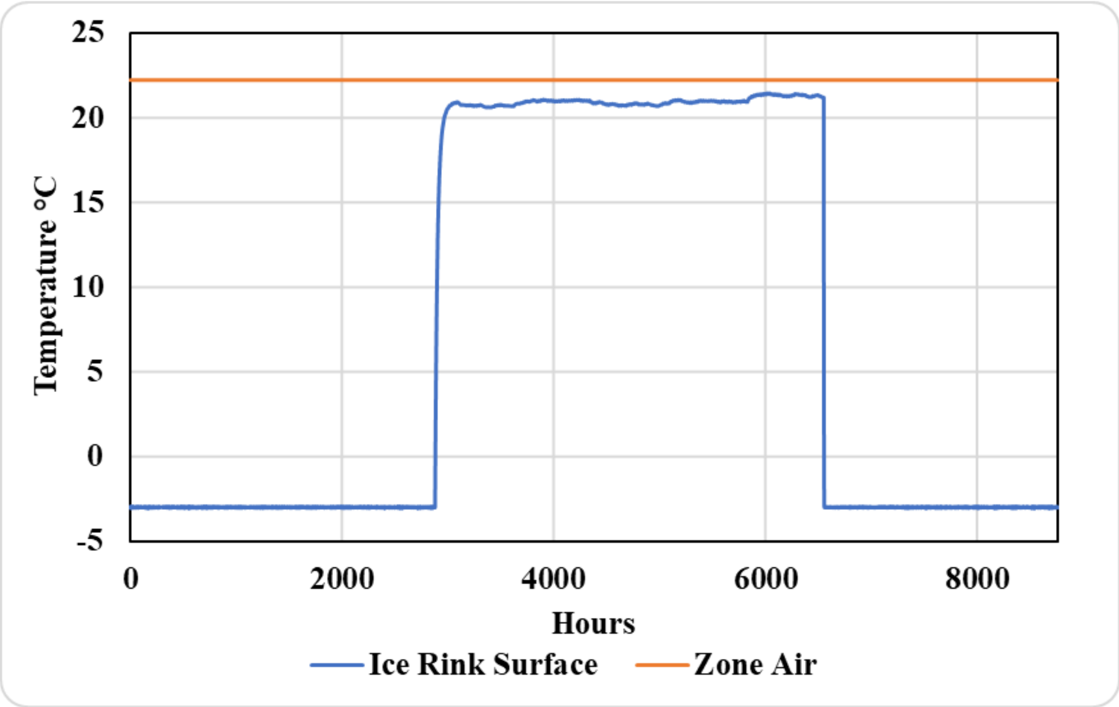


Figure 3. 12: Hourly variations of floor surface temperature and indoor air temperatures based on EnergyPlus the ice rink facility model's predictions

Based on EnergyPlus model's predictions, Figure 3.12 indicates that when the refrigeration system of the ice rink is shut-off, and the rink's floor surface temperature starts to rise to reach indoor air temperature. Specifically, Figure 3.13 shows that it takes around 46 hours for the ice to melt and water temperature to rise until it stabilizes at a temperature slightly lower than the indoor facility air temperature.

Furthermore, figure 3.13 shows the surface temperature when the ice rink is on and off. The ice rink is first maintained at the baseline ice rink operational setpoint which is -3°C. It also shows the zone air temperature maintained at the desired zone air setpoint. Once the ice rink is off, the surface temperature increases until it is close to the zone air temperature.

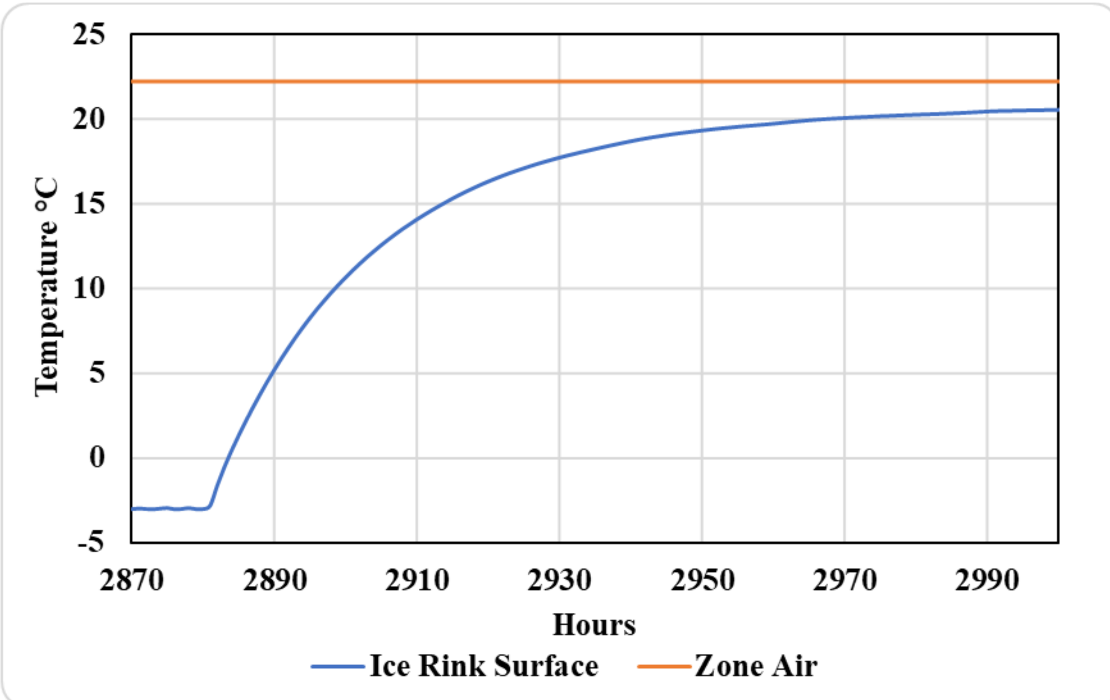


Figure 3. 13: Hourly variation of floor surface temperature and indoor facility air temperature during a period when the ice rink refrigeration system is shut-off

Chapter IV: Sensitivity Analysis

4.1 Introduction

The energy performance of ice rinks depends on several design and operation factors. In this chapter, a series of sensitivity analyses is conducted to assess the most important factors that affect a typical ice rink energy performance as well as to test the results from the new EnergyPlus module for ice rinks. The parameters used in this analysis include operational schedule, coefficient of performance (COP), roof thermal absorptance, location, indoor air temperature setpoint, ice rink surface temperature setpoint, and rink dimensions. For the sensitivity analysis, the seasonal operation schedule (refer to Figure 3.2) represents the baseline schedule. As discussed in chapter 4,0lighting and equipment energy consumption is constant throughout the analysis since their schedule did not change except for the annual operation results. After the evaluation of the impact of the individual parameters, a package of energy efficiency measures (EEMs) is considered to determine its impact of the energy performance of the ice rink facility.

4.1.2 Resurfacer Operation

As discussed in the literature review in chapter 1, the resurfacer helps maintain the quality of the ice surface. The audited facility report did not report any information about the resufacer operation. However, it is an important component of any ice rink operation. Therefore, it was added to the model used in the sensitivity analysis. The model used for the sensitivity analysis has the resurfacer scheduled to operate every 6 hours corresponding to 3 times per day when the facility is in operation. The modeled resurfacer has a water tank capacity of 0.55m^3 (145.3 gallons) and uses electricity to heat water up to 55 °C (131°F). Figure 3.10 shows the impact of the resurfacer on the floor temperature during one day. Every period when the resurfacer is used, the ice floor temperature decreases to -2°C. However, the refrigeration system is able to cool down rather quickly the floor surface temperature to the setpoint of -3°C.

4.2 Impact of Operation Schedule

The operation schedule can have a significant impact on the energy consumption of an ice rink facility. Typical ice rink operation is either seasonal or annual with a maintenance break [2]. Figures 4.1 represents respectively the annual and seasonal schedules used for the sensitivity analysis carried out in this section. A value of 1 in these schedules means that the ice rink and associated refrigeration system are operational on and a value of 0 means the ice rink's refrigeration systemis shut-off. Figure 4.1 (a) shows that for the annual operation schedule, the ice rink is used throughout the year except for the month of June for maintenance purposes. Figure 4.1(b) represents the seasonal operation schedule where the ice rink is used from the beginning of October until the end of April. In the summer, it is assumed that the ice rink is not operated. For both schedules, internal loads are modified from the base case settings to match the ice rink operation schedule. The baseline lights and equipment schedules were discussed in chapter 3. Figure 4.3 represents the lights and equipment annual schedule. The usage fractions used are equal to the baseline. However, the lights and equipment operation stays longer to accommodate the annual ice rink operation schedule.

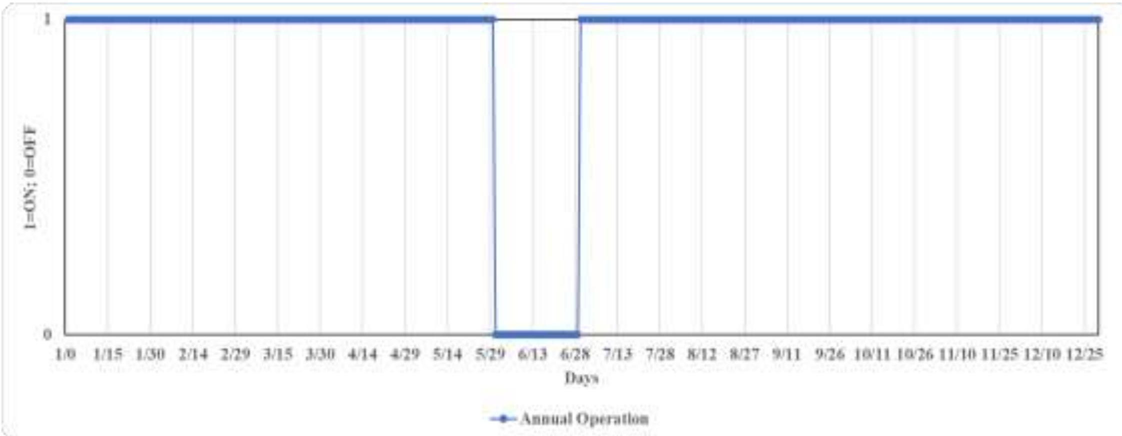


Figure 4. 1: **seasonal operation schedule for an ice rink facility**

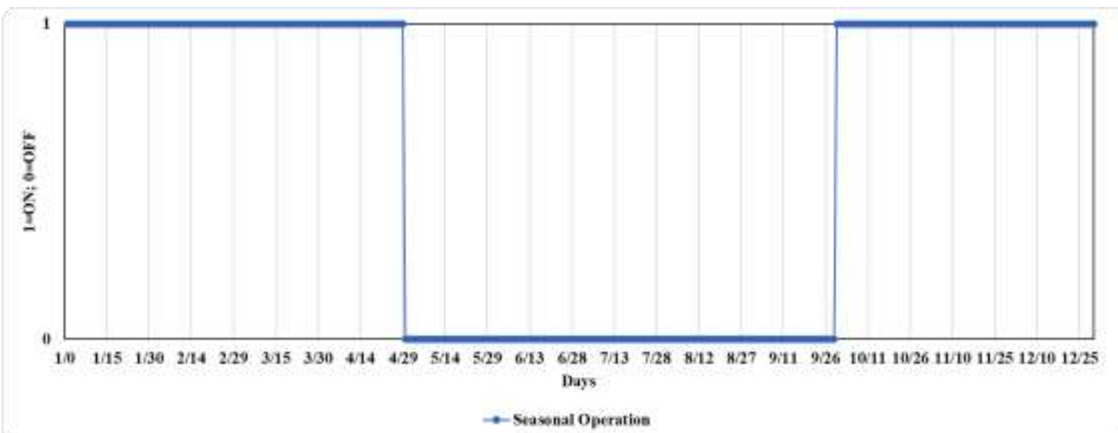


Figure 4.2: **seasonal operation schedule for an ice rink facility**

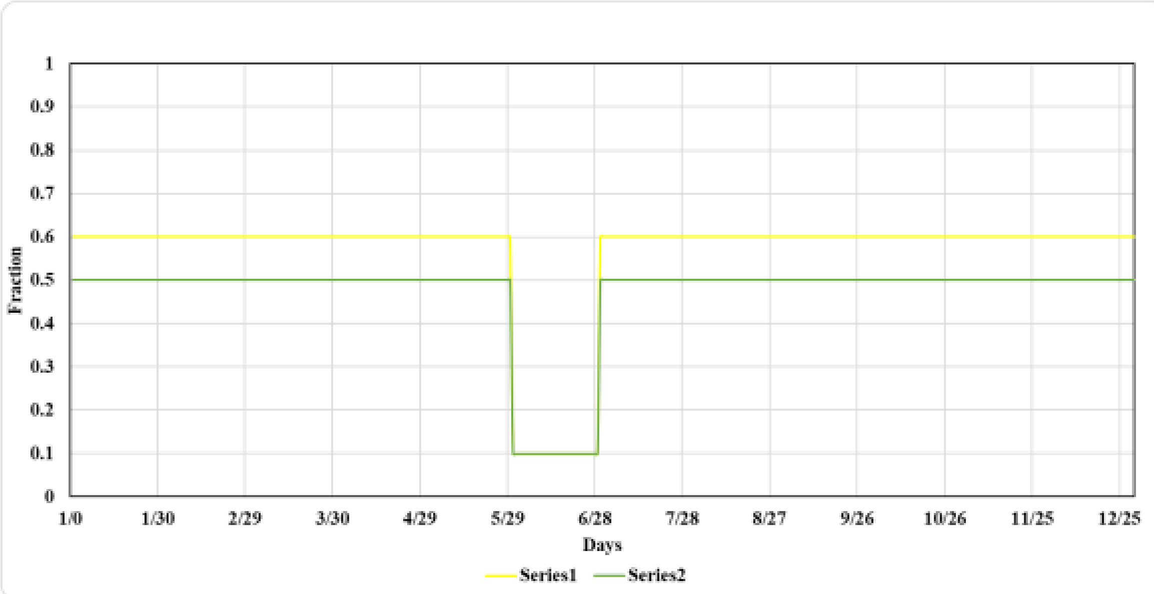


Figure 4. 3: Lights and equipment annual schedule

Figure 4.4 shows the source energy consumption for both seasonal and annual operation schedules. As expected, the ice rink facility consumes more energy when operated based on an annual schedule rather than on a seasonal schedule. Specifically, the ice rink facility consumed during one year 12,258 GJ during seasonal operation and 18,492 GJ during annual operation as shown in Figure 4.3, that is an increase of over 50%. The ice rink refrigeration system consumes annually 4,563 GJ and 7,231 GJ equivalent to 37% and 39% of total consumption for seasonal and annual operation, respectively. Since the ice rink floor is maintained at -3°C and the outdoor dry bulb temperature is lower than indoor temperature setpoint of 22°C during most of the year in as shown by Figure 4.4, heating consumes a significant amount of energy and is the second highest energy end use energy as indicated in Figure 3.3. In particular, space heating of the ice rink facility consumes annually 2,720 GJ and 3,803 GJ equivalent to 22% and 21% of total energy consumption for respectively, seasonal and annual operation schedules. The peak heating demand takes place in during January and equals to 342 GJ in for both operation schedules. Indeed, the ice rink and associated refrigeration system are operations for both schedules during the month of January. Space cooling needs occur mainly in the concession thermal zone where a higher occupancy level occurs during the summer months for the annual operation schedule with a peak demand of 15.4 GJ in August.

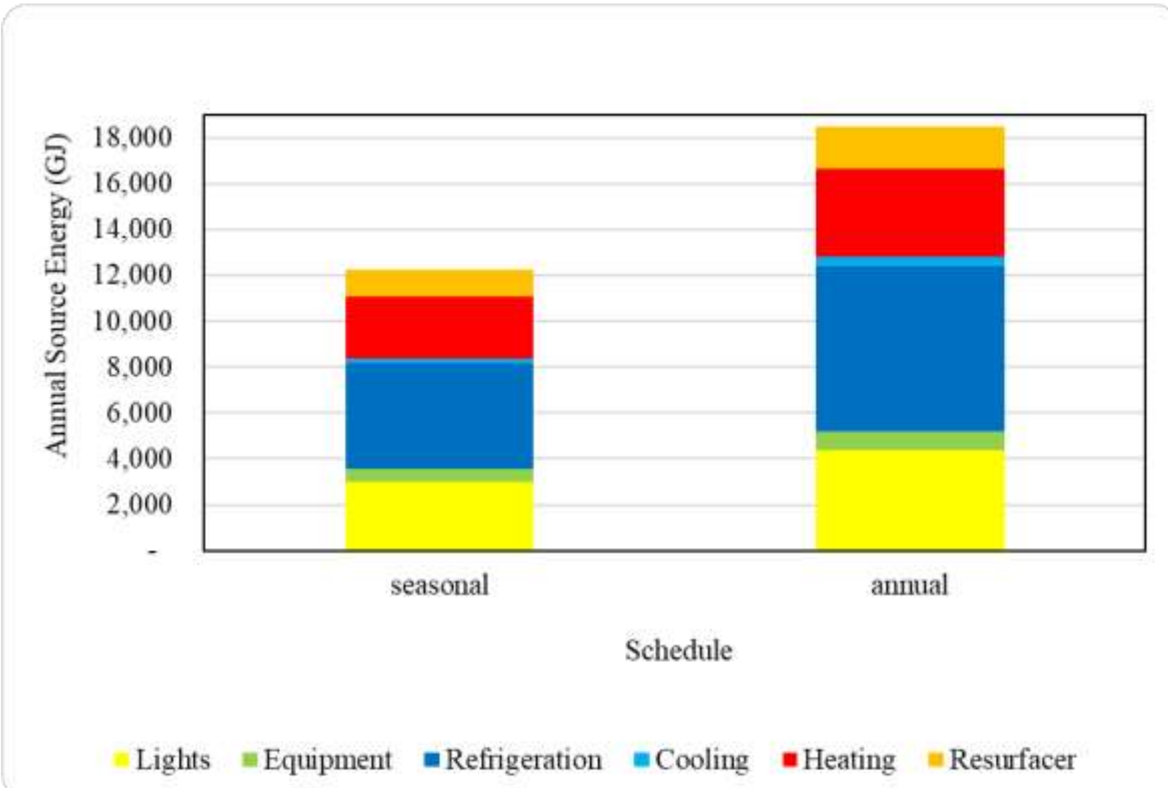


Figure 4. 4: Impact of operation schedule on source energy consumption for the ice rink facility

Since the resrufacer operates only when the ice rink is used, hence, its energy consumption is higher for annual operation. Specifically, the resurfacer consumes annually1,177 GJ and 1,865 GJ respectively, for both seasonal and annual schedules. Lighting's energy annual consumption is 3,018 GJ and 4,393 GJ, which is equivalent to 24% and 25% of total energy usage for respectively, seasonal and annual operation schedules. The equipment annual consumption is equal to 562 GJ and 791 GJ that is equivalent to 5% and 4% of total energy consumption for seasonal and annual operation, respectively. In general, energy consumption of all end uses increases due to the longer operation schedule. For seasonal operation schedule, the facility's source, and site energy use indices (EUIs) are 3.65 GJ/m^2 and 1.64 GJ/m^2. On the other hand, for annual operation, the source and site EUI are 5.50 GJ/m^2 and 2.42 GJ/m^2 for annual operation schedule.

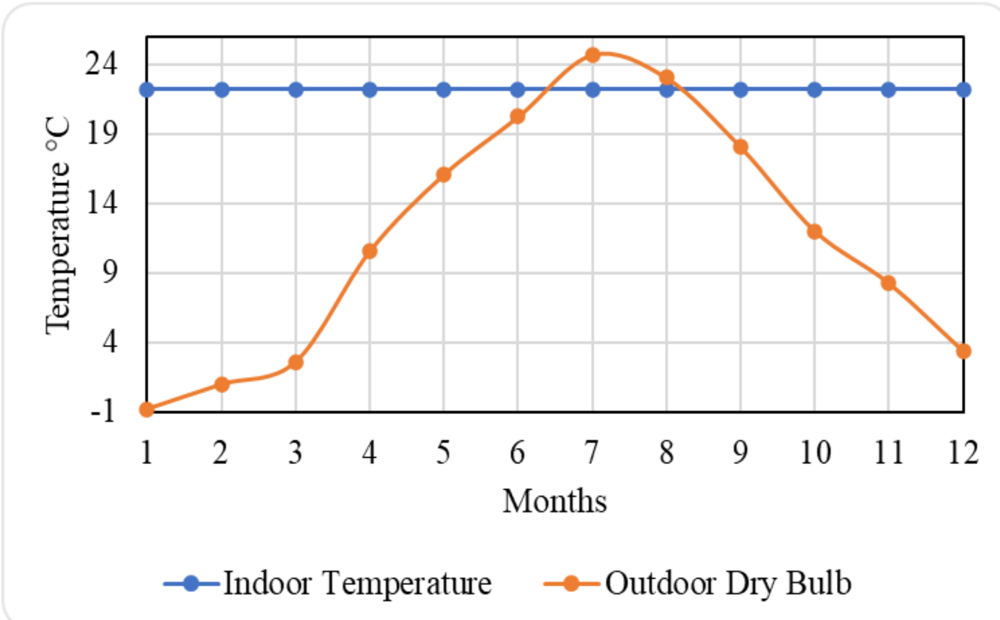


Figure 4. 5: Monthly variations of outdoor and indoor air temperatures when the ice rink facility is located in Newark, NJ

4.3 Impact of COP for the Refrigeration System

The efficiency of the refrigeration system is another important parameter that affects the the energy performance of the ice rink facaility. As mentioned in chapter III, the refrigeration system efficiency for the baseline ice rink facility's model is 2.5 based on the validation analysis. As part of sensitivity analysis considered in this section, the COP of the refrigeration system is increased to first 3.0 and then to 3.5. As seen in the analysis results shown in figure 4.6, the annual energy consumed by the ice rink facility 12,258 GJ, 11,497 GJ, 10,954 GJ when COP of respectively, 2.5, 3.0 and 3.5 is used. As expected, the results of this sensitivity analysis show that the refrigeration energy demand is significantly affected by the coefficient of performance. For instance, the refrigeration energy use is decreased to 3,259 GJ from 4,563 GJ, when the refrigeration system's COP is increased to 3.5 from 2.5. This 28.6% reduction of refrigeration energy end-use is equivalent to an 11% of source energy savings for the entire facility. Since the COP only affects the energy performance of the refrigeration system, other energy end uses such as those associated to resurfacer, HVAC and lighting , are not affected. The facility's source EUI is estimated to be 3.65, 3.42 and 3.26 GJ/m^2 for a COP of 2.5, 3.0 and 3.5, respectively. The facility site EUI is determined to be 1.64, 1.57 and 1.52 GJ/m^2 for a COP of 2.5, 3.0 and 3.5, respectively.

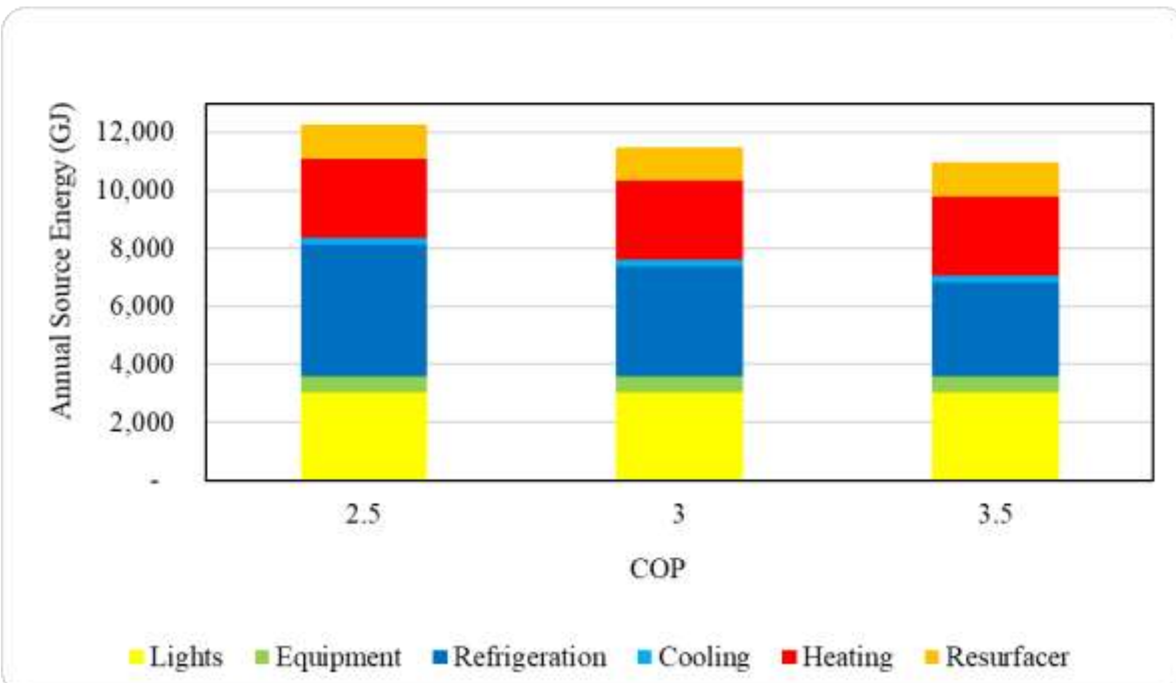


Figure 4. 6: Impact of the refrigeration system COP on source annual energy consumption for the ice rink facility

4.4 Impact of Ceiling

The thermal absorptance is a material property that the user inputs for layers of different building surfaces in an EnergyPlus model. This property represents the radiation that is absorbed by the material, which is also referred to as emissivity. On the other hand, reflectivity refers to how much light is reflected from the surface. If the material's reflectivity data is available, then thermal absorptance is equal to [19]:

$$Thermal\ Absorptance = 1 - reflectivity$$

The roof's temperature is higher than the ice surface and hence radiates heat to it. The ceiling radiation contributes about 30% of the ice rink refrigeration load [25]. Therefore, adding a low emissive surface will reduce the amount of radiation back to the ice surface. Therefore, in the sensitivity analysis presented in this section, the thermal absorptance (i.e., emissivity), of the inside layer of the ceiling, which is ½" gyp board, is varied to assess its impact on the energy performance of the ice rink facility including its refrigeration systems. Figure 4.7 shows the impact of ceiling absorptance on the ice rink facility's source annual energy consumption. The lower the ceiling's emissivity the lower is the energy used by the ice rink facility. This change in energy performance of the ice rink facility is mainly driven by the change in refrigeration loads and HVAC loads. The low emissivity ceiling affects the HVAC because it acts as a thermal breaker that resist the heat transfer between the roof and the zone. The facility's annual energy consumption is 12,258 GJ, 12,193 GJ, and 12,087 GJ for a thermal absorptance of 0.3, 0.6, and 0.9, respectively. According to the international ice hockey federation guide, having a low thermal absorptance ceiling surface in an ice rink facility can lead to energy savings [2]. Based the EnergyPlus simulation results, when the ceiling thermal absorptance of 0.9 is lowed to 0.3, annual energy savings of 0.6%, 1.3% and

5.2% due to ice rink refrigeration, space cooling, and space heating, respectively. The space heating is the most affected end use because it is considered the major HVAC load taking place during most of the operation schedule. Although decreasing ceiling thermal absorptance saves energy its overall impact is not significant. For instance, decreasing the ceiling thermal absorptance from the baseline's value of 0.9 to 0.6 or 0.3 saves respectively, 0.53% and 1.4% of annual energy used by the entire ice rink facility. The reported savings in literature is about 5% savings in total energy [25]. However, the literature does not provide information about the facilities used to obtain the reported savings. There are a lot of factors that can contribute to the total energy consumption which would affect the percentage of savings as well. The resurfacer, lighting and equipment energy consumption did not change since their operation is based on an operation schedule, which was not affected by the ceiling properties. The facility's site EUI is 1.64, 1.63 and 1.6 GJ/m^2 while its source EUI, is 3.65, 3.63 and 3.60 6 GJ/m^2 when the ceiling thermal absorptance is 0.3, 0.6, and 0.9, respectively.

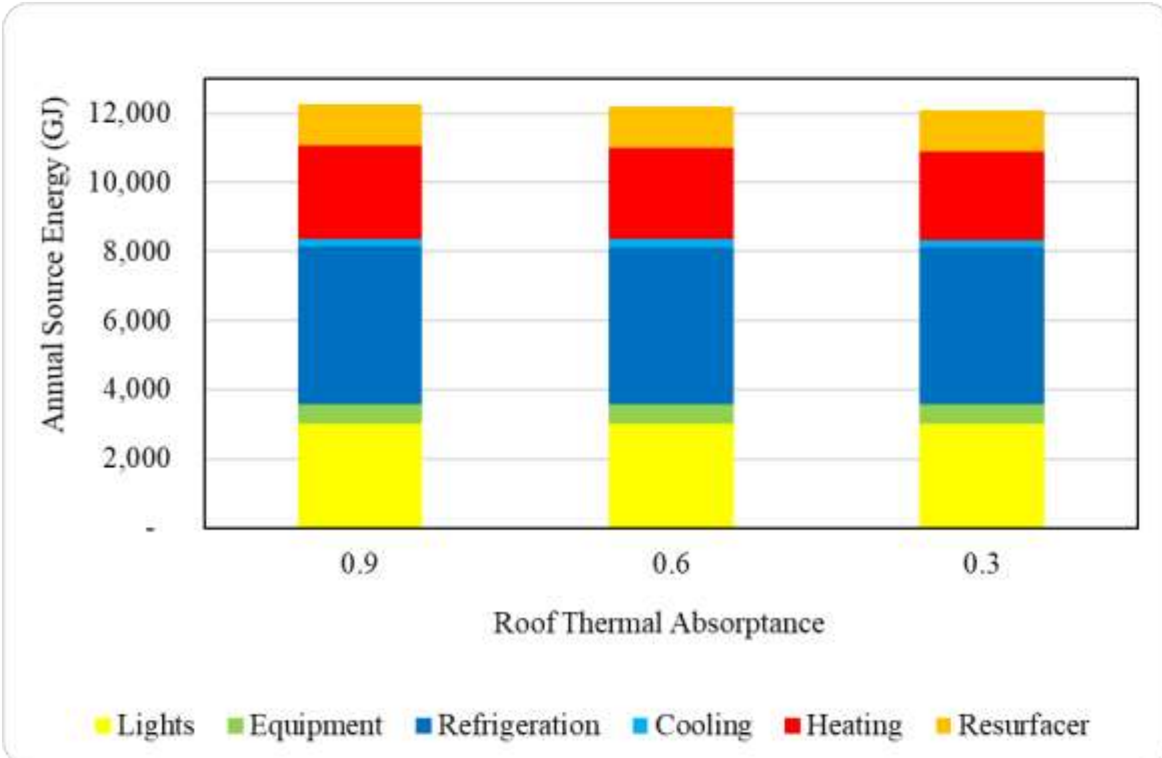


Figure 4.7: Effect of ceiling emissivity on source annual energy consumption for an ice rink facility

4.5 Impact of Zone Air Temperature Setpoint

The indoor air temperature setting can affect the HVAC energy use for any building. In this section, the impact of indoor air temperature setpoint is evaluated on energy performance for both the HVAC and ice rink refrigeration systems. The baseline indoor temperature for ice rink facility is set at 22 °C which complies with ASHRAE 55 [26]. For the sensitivity analysis, the indoor air temperature specific to the ice rink zone varied to 16°C, 22°C, and28°C when the ice rink arena is operating while the concession zone is maintained at 22°C. When the ice rink is not operational during summer, the indoor air temperature is set back to 22°C because the focus of this sensitivity analysis is to assess the facility's energy performance during the ice rink operation. Figure 4.8 summarizes the results of the impact of indoor air temperature setting on the annual source energy use for the ice rink facility. The source annual energy consumption is estimated to be 10,157, 12,258 and 15,273 GJ when the indoor temperature setpoint is set at 16°C, 22°C, and 28°C, respectively. As shown in Figure 4.8, the facility annual energy consumption increases as the indoor temperature setting increases. Indeed, heating demand increases for the ice rink zone when the temperature setpoint is increased and set to 28°C resulting to higher refrigeration load in order to maintain the ice rink surface temperature at -3°C. On the other hand, there is a significant potential for energy savings when decreasing the indoor temperature setpoint. For instance, setting the indoor air temperature within the ice rink arena to 16°C instead of 22°C, reduces the refrigeration and heating annual energy end-uses from 4,563 GJ to 3,511 GJ and 2,720 GJ to 1,492 GJ, respectively. Therefore, this temperature setting reduction achieves 17% savings in annual energy consumption for the entire ice rink facility. However, increasing the air temperature setpoint for the ice rink arena to 28°C results in 25% higher annual energy use for the facility. The cooling energy demand has not changed significantly given that the facility in , Newark,New Jersey is a heating dominated climate. Moreover, the ice rink surface contributes to lower the space cooling needs. The facility's site EUI is determined to be 1.22 GJ/m^2, 1.64 GJ/m^2 and 2.26 GJ/m^2 while its source EUI is found to be 3.02 GJ/m^2, 3.65 GJ/m^2 and 4.5 GJ/m^2 when air temperature for the ice rink arena is set at 16°C, 22°C, and28°C, respectively.

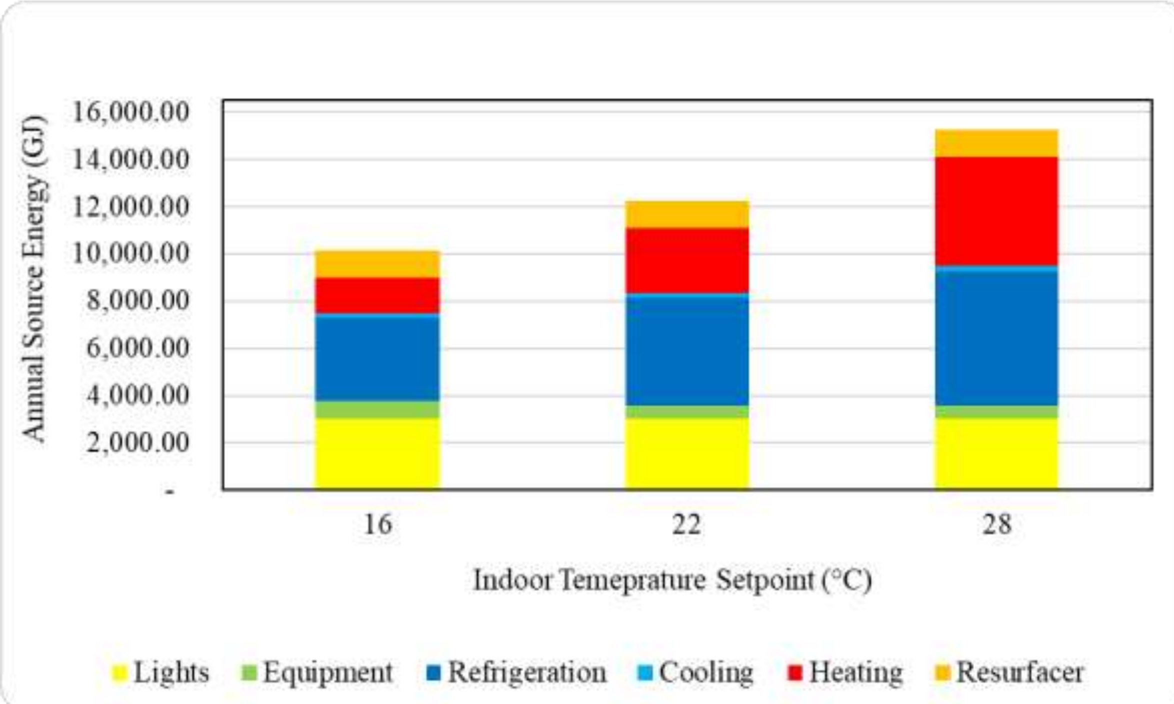


Figure 4. 8: Impact of indoor air temperature setpoint on annual source energy consumption for the ice rink facility

4.6 Impact of Climate

Climatic conditions affect the energy performance of any building including ice rink facilities. In this section, the energy performance of the ice rink facility is evaluated for different locations that represent different US climates. In particular, the locations are selected in this analysis to represent a diverse climate zones in the US. Table 4.1 shows the climate characteristics of each city used in this sensitivity analysis. As mentioned in Chapter III, the baseline ice rink facility is located in old bridge township near Newark, NJ, which is characterized by cold climate. Other locations considered in the analysis include, Miami, FL, Denver, CO, and Pheonix, AZ. These locations are selected to represent hot/humid cold/dry, and hot/dry climates, respectively.

Table 4.1: Heating and cooling degree days [27]

City	Heating Degree Days	Cooling Degree Days
Newark, NJ	4633	1270
Denver, CO	5667	721
Phoenix, AZ	912	4636
Miami, FL	113	4578

Figure 4.9 shows the annual source energy use for the ice rink facility when located at different locations. From the results of figure 4.9, the change in annual energy consumption is rather not significant for various climates. Indeed, the annual energy consumption is found to be 12,258, 12,383, 12,195, 12,204 GJ for Newark, Denver, Miami, and Phoenix, respectively.

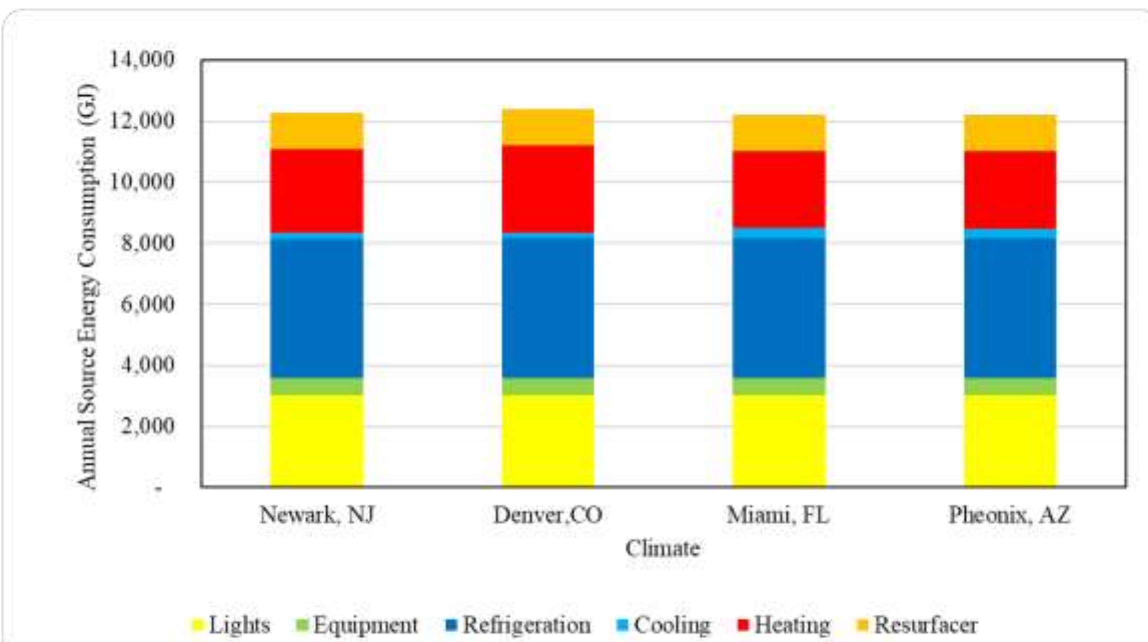


Figure 4.9 Effect of the climate on annual source energy consumption for the ice rink facility

Since the ice rink's operation schedule spans over winter months and not during the summer months, the heating load dominates the HVAC consumption. Therefore, when the ice rink facility is located in Newark, NJ and Denver, CO, the annual energy consumption is higher than when the location is Miami, FL, and Phoenix, AZ. Although Miami, FL, and Phoenix, AZ, have hot climates, it appears that the cooling effect of the ice rink surface is significant that heating rather than air conditioning is needed to maintain thermal comfort within the arena [2]. The annual heating energy end-use is 2,720, 2,843, 2,496 and 2,560 GJ for Newark, NJ, Denver, CO, Miami, FL, and Phoenix, AZ, respectively. The fact that when the ice rink facility is located Phoenix, AZ, has higher heating energy end-use than when it is located in Miami, FL, resulted in higher annual energy consumption. Although indoor conditions are set to be constant in the facility throughout the year, the ice rink refrigeration needs are slightly higher in hot climates due to the added thermal load most likely caused by the air infiltration. Indeed and according to IIHF ice rink guide [2], air infiltration can increase in ice rink refrigeration electricity consumption and can cause mold growth, metal corrosion, and lower ice quality. As indicated in Figure 4.8, space cooling energy end-use is not significant for all the climates due to the fact the ice rink is not running in summer months. With the seasonal operation schedule, the ice rink facility has the highest cooling needs, when it is located in Miami, FL, followed by Phoenix, AZ. Moreover, energy end-uses specific to lighting, resurfacer, and equipment remain the same for all the locations. Site and source EUIs for the ice rink facility are 1.64 GJ/m^2 and 3.65 GJ/m^2 for Newark, NJ, 1.68 GJ/m^2 and 3.69 GJ/m^2 for Denver, CO, 1.6 GJ/m^2 and 3.63 GJ/m^2 for Miami, FL, and 1.61 GJ/m^2 and 3.63 GJ/m^2 for Phoenix, AZ.

4.7 Impact of Ice Surface Temperature

As discussed in the literature review, there are different categories of ice rink arenas depending on the activities and sports commonly practiced in the facilities. Each activity type requires specific ice rink surface's temperature. Therefore, different surface temperature setpoints are considered for the ice rink in the sensitivity analysis presented in this section. The ice rink surface temperatures selected for the sensitivity analysis are -3°C, -4°C, and -5°C since they represent the recommended settings used for different ice rink activities [1,2]. The results of the analysis including the annual source energy end-use for various ice surface temperatures are illustrated in Figure 4.10. As expected, the results of Figure 4.10 indicates that the facility's annual source energy consumption increases as the ice surface temperature decreases.

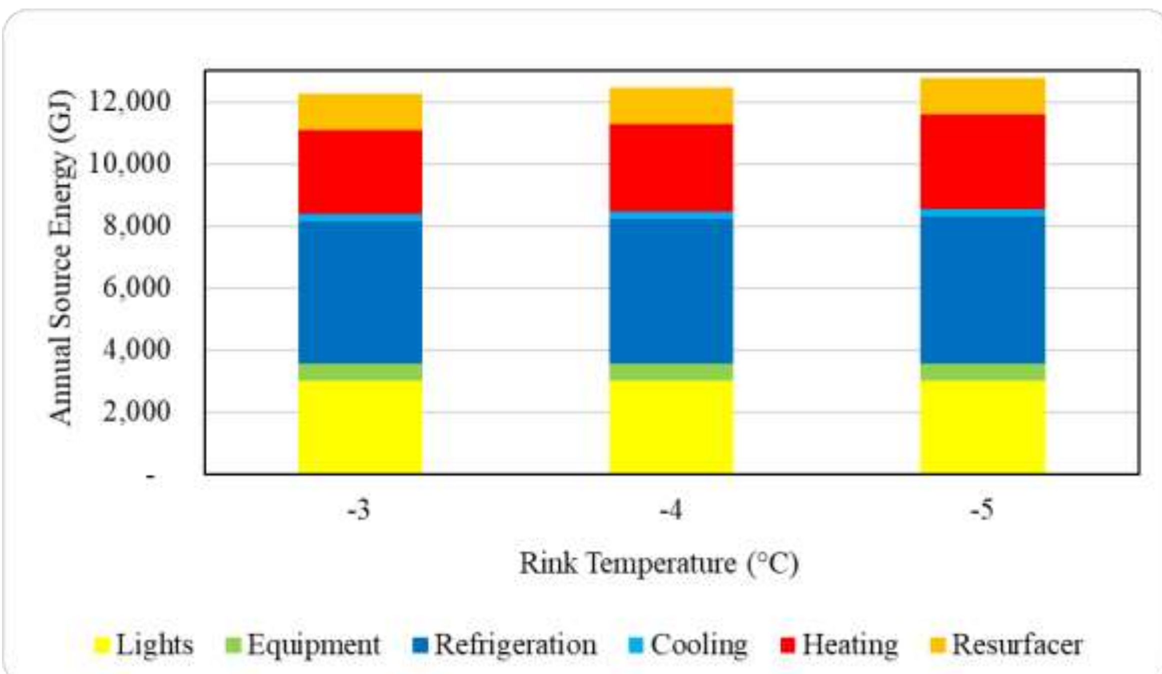


Figure 4. 10: Impact of ice surface temperature on annual source energy consumption for the ice rink facility

The ice rink facility consumes annually 12,258 GJ, 12,466 GJ and 12,763 GJ of source energy when the ice surface temperature is set at -3°C, -4°C, and -5°C, respectively. The ice rink refrigeration system is the main driver of this increase due to the higher refrigeration thermal load associated with the lower ice surface temperature setting. The annual source energy consumption specific to ice rink refrigeration is found to be 4,563 GJ, 4,643 GJ and 4,724 GJ when the ice surface temperature is set at -3°C, -4°C and -5°C, respectively. Similarly, the space heating needs increase with lower ice surface temperature. Specifically, the facility's heating source energy end-use is 2,720 GJ, 2,848 GJ, 3,064 GJ for ice surface temperature of -3°C, -4°C and -5°C, respectively. The space cooling demand did not change significantly with the ice surface temperature setting since ass noted earlier for seasonal operation schedule, the facility is not operated during the summer months. The overall ice rink facility's annual source energy consumption increases by 1.7% and 4.1% when the ice surface temperature is decreased from -3^{o}C to -4°C and -5°C, respectively. The site and source EUI values for the ice rink facility are estimated to 1.64 GJ/m^2 and 3.65 GJ/m^2 for -3°C, 1.69 GJ/m^2 and 3.71 GJ/m^2 for -4°C, 1.75 GJ/m^2 and 3.80 GJ/m^2 for -5°C.

4.8 Impact of Ice Rink Size

Since ice rink's refrigeration system consumes a significant fraction the energy used by the facility as outlined in the previous analyses, the size of the ice rink arena size can have ea major role in the overall facility's energy consumption. The size of the ice rink arena considered so far is 30-m x 60-m which is considered the largest size for ice rink arenas. In this section, a sensitivity analysis is carried out to assess the effect of the size of the ice arena on the facility's energy performance. Specifically, two additional sizes are considered including 26-mx56-m and 28-mx58-mm ice rink arenas [2]. Given that the ice rink refrigeration thermal load is directly related to the volume of the ice pad, the refrigeration energy end-use decreases as the size of the ice rink arena decreases. Figure 4.11 shows the annual source energy end-uses for ice rink facilities with various ice rink arena' sizes. Specifically, Figure 4.10 indicates that the annual source energy consumption is 12,258, 11,630 and 11,001 GJ when the ice rink arena's size is set to 30-mx60-m, 28-mx58-m and 26-mx56m, respectively. Since the presence of the ice rink area affects the HVAC loads for the facility, the space heating loads decreases when decreasing the size of the ice rink arena. Specifically, annual heating source energy end-use is estimated to be 2,720 GJ 2,506 GJ, and 2,304 GJ when the ice rink arena's size is set to be 30-mx 60-m, 28-mx58-m and 26-mx56-m, respectively. Compared to the baseline case of 30-mx60-m ice rink arena, the facility can incur annual source energy savings of 5.1% and 10.3% when the size of the ice rink arena is reduced slightly to 28-mx58-m and 26-mx56-m, respectively. Thus, downsizing even slightly the size of the ice rink arena can have a substantial energy savings. The facility's source and site EUI are estimated to be 1.64 GJ/m^2 and 3.65 GJ/m^2 for 30-mx60-m ice rink arena, 1.55 GJ/m^2 and 3.46 GJ/m^2 for 28-mx58-m arena, and 1.45 GJ/m^2 and 3.27 GJ/m^2 for 26-mx56-m arena

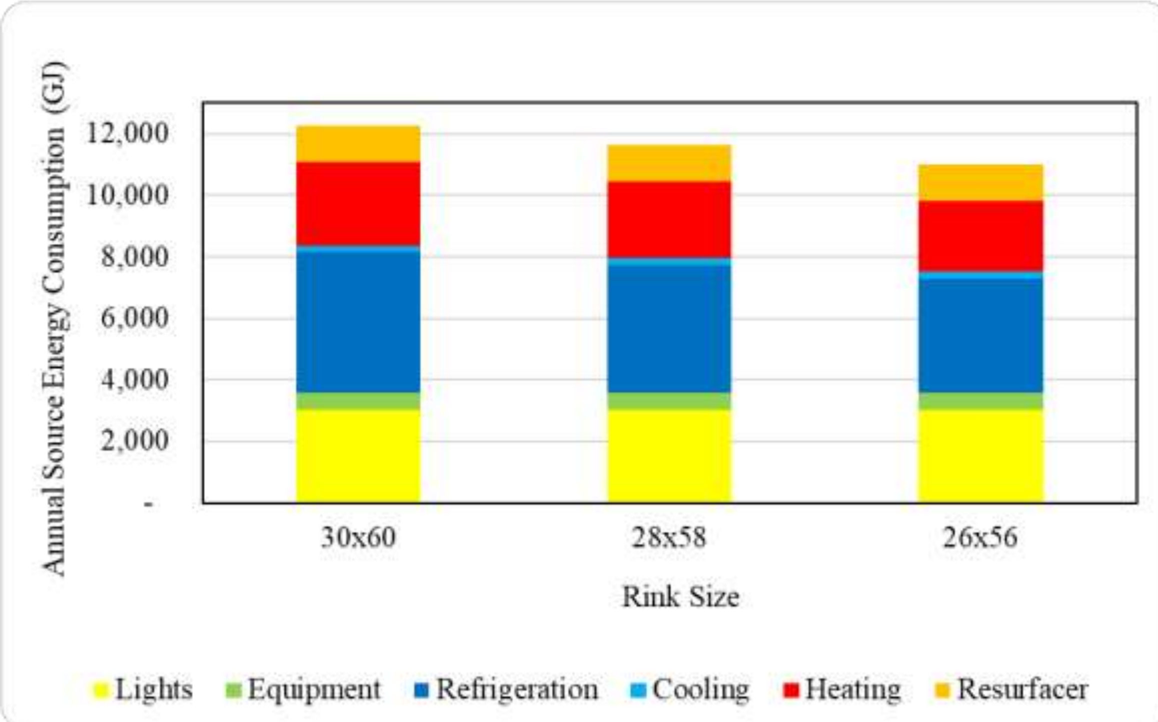


Figure 4. 11: Impact of the size of the ice rink arena on annual source energy consumption of the ice rink facility

4.9 Impact of Air Infiltration Rate

In this section, the impact of air infiltration rate is evaluated when the ice rink facility is located in different US climates. Figure 4.12 shows that increasing air infiltration rate results in an increase of annual energy consumed by the ice rink facility in all the climates. The increase in energy consumption due to higher air infiltration rate is mainly due to increased HVAC thermal loads to maintain the desired indoor air temperature. In particular, the refrigeration load specific to the ice rink floor is impacted by the indoor condition of the facility. Since the indoor air temperature did not change from the setpoint, 22°C, the refrigeration load did not change significantly due to variation in air infiltration rate. In this analysis, air infiltration rate of 0.5 ACH and 1.0 ACH are compared to the baseline ACH of 0.01 representing an airtight building. From the results of Figure 4.12, air infiltration rate has a higher impact for cold climates such those of Newark, NJ, and Denver, CO, than for hot climates of Miami, FL, and Phoenix, AZ. For instance, increasing the infiltration to 0.5 ACH, the total energy of consumption will increase 18% for Denver, CO, and Newark, NJ, 5% for Miami, FL, and 14% for Phoenix, AZ. Regardless of climate, space heating is required for the ice rink zone to maintain indoor thermal comfort. Figure 4.12 shows that increasing air infiltration rate led to higher cooling loads in the hot climates of Miami, FL and Phoenix, AZ. On the other hand, higher infiltration led to lower cooling loads in cold climates such as Newark, NJ and Denver, CO. The cooling load is mainly specific to maintain thermal comfort in the concession area. For instance, air infiltration rate of 0.5 ACH results in a 115% increase and 63% reduction in cooling energy end-use when the ice rink facility is located in Miami. FL and Denver, CO, respectively. However, the contribution and any change of the cooling is not significant on total energy consumption of the ice rink facility. In summary, air infiltration rate has more impacts on the energy performance of ice rink facilities located in cold climates than those in hot climates. In conclusion, these results confirms that the importance of building's air tightness [2].

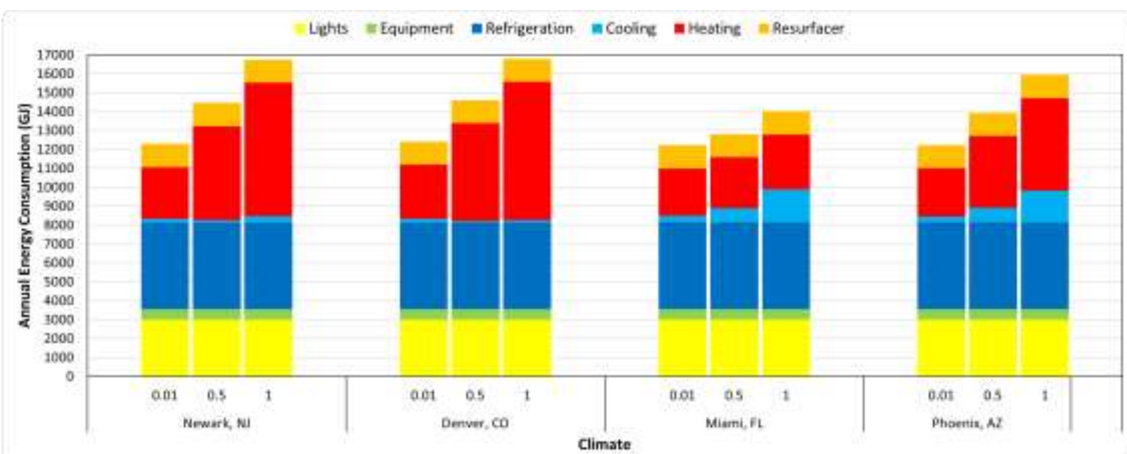


Figure 4. 12: Impact of infiltration on annual source energy consumption of the ice rink facility in different climates

4.10 Impact of Wall Thermal Insulation

In this section, the impact of increasing wall thermal insulation on the energy performance of ice rink facilities is evaluated. In this analysis, wall thermal insulations of R-0 and R-21 are compared to the baseline's R-11 when the ice rink facility is located in different climates. It is not recommended to over insulate the roof because it may cause condensation issues [2,15]. Therefore, increasing the roof insulation above code values was not evaluated in this analysis. Figure 4.13 shows the impact of adding wall thermal insulation on the annual energy consumption of the ice rink facility located in different climates. The analysis results indicate that i wall thermal insulation does not have a significant impact with the change in annual energy consumption is less than 1%. Although it is important to have a thermally insulated envelope, increasing wall insulation level does not have a significant impact on the ice rink facility's energy consumption. Figure 4.13 shows that in most climates, thermally insulated buildings have a lower energy consumption. However, a building with no insulation in Miami consumes less energy because its heating load significantly decreased. According to IIHF, maintaining high air tightness is more important than adding thermal insulation [2].

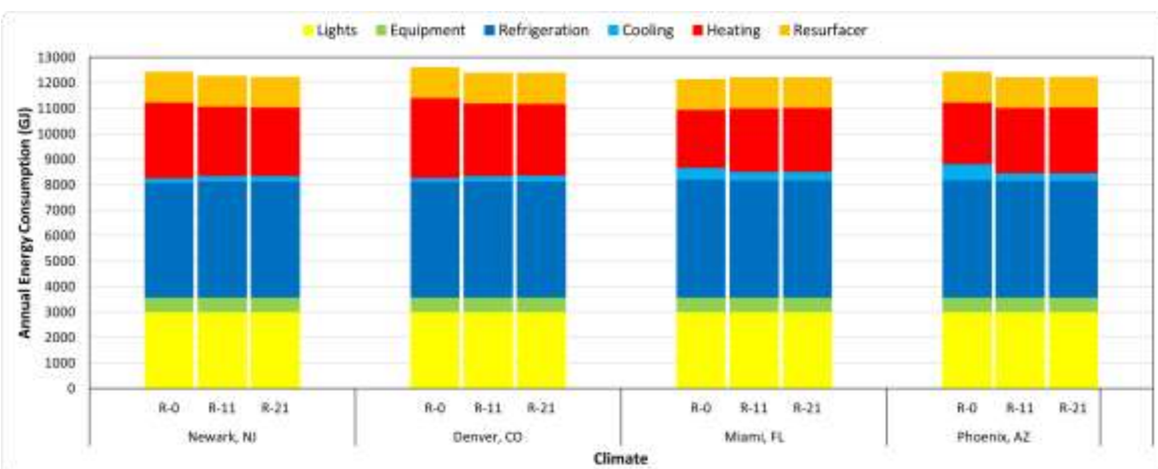


Figure 4. 13: Impact of wall thermal insulation on annual source energy consumption of the ice rink facility in different climates

4.11 Impact of Occupancy Level

In this section, the impact of occupancy level on the ice facility's annual source energy is investigated. The occupancy level for the baseline case is 210 in the concession area and 50 in the ice rink zone area based on the International Mechanical Code 2018. The location for the ice rink facility considered for this analysis is Denver, CO. To vary the occupancy level, the baseline values are adjusted using multipliers of 1.5 and 2. Figure 4.14 shows the impact of increasing occupancy on annual source energy of the ice rink facility. The total energy consumption increased to 12,530 GJ and 12,663 GJ for an occupancy multiplier of 1.5 and 2, respectively, correspondent to increases of 1.2% and 2.3% from the baseline. This change is mainly due to the increase in cooling loads specific to the concession area. Indeed, the cooling energy end-use increased 74% and 142% for the multiplier 1.5 and 2, respectively. On the other hand, heating energy end-use has slightly decreased due to higher occupancy level. The ice rink facility's site EUI is determined to be 1.68 GJ/m^2, 1.69 GJ/m^2 and 1.7 GJ/m^2 while its source EUI is found to be 3.65 GJ/m^2, 3.73 GJ/m^2 and 3.74 GJ/m^2 when the occupancy multiplier is 1, 1.5 and 2, respectively.

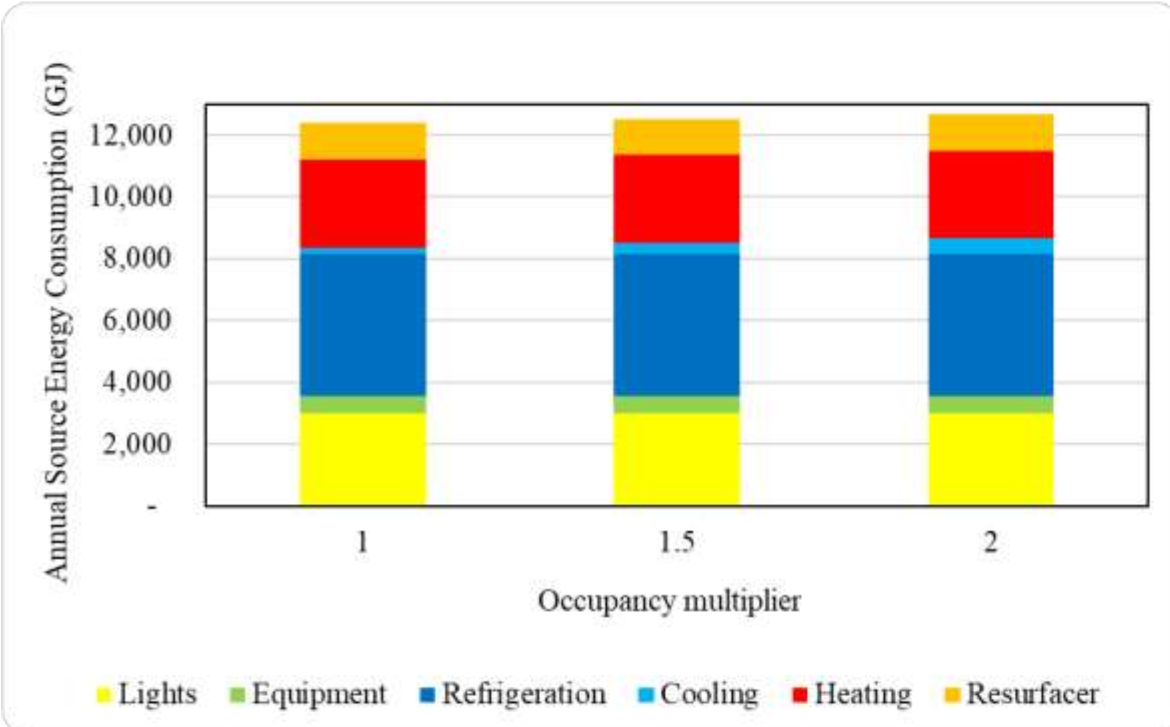


Figure 4. 14: Impact of occupancy on source energy consumption of the ice rink facility located in Denver, CO

4.12 Impact of Ice thickness

In this section, the impact on overall facility's energy consumption of the ice thickness for the ice rink is evaluated. As mentioned in literature, the ice thickness depends on the type of activity taking place in the ice rink. For the analysis, the baseline ice thickness is set to be 25 mm (1 inch) and the ice rink facility is located in Denver, CO. In this analysis, the baseline ice rink thickness is compared to 19 mm (0.75 inch) and 31.75 mm (1.25 inch). Figure 4.15 shows the impact of increasing the ice thickness on the total source energy consumption. The ice rink facility's energy consumption is 12,107 GJ, 12,383 GJ and 12,579 GJ for a 0.01905 m (0.75"), 0.0254 m (1") and 0.03175 m (1.25") ice rink thickness. Similar to the ice rink size, the ice thickness affects the ice volume within the rink and thus the refrigeration load. Therefore, the refrigeration energy end-use is the main driver for the change in the overall facility's energy consumption. The annual energy end-use attributed to refrigeration is decreased by 5% when reducing the ice thickness from 0.0254 m (1") to 0.01905 m (0.75"). On the other hand, the refrigeration energy end-use increases 6.2% when increasing the ice thickness from 0.0254 m to 0.03175 m (1" to 1.25"). The ice rink facility's site EUI is estimated to be 1.64 GJ/m^2, 1.68 GJ/m^2 and 1.69 GJ/m^2 while its source EUI is found to be 3.6 GJ/m^2, 3.69 GJ/m^2 and 3.74 GJ/m^2 when the ice thickness is 0.01905 m, 0.0254 m and 0.03175 m (0.75", 1" and 1.25"), respectively.

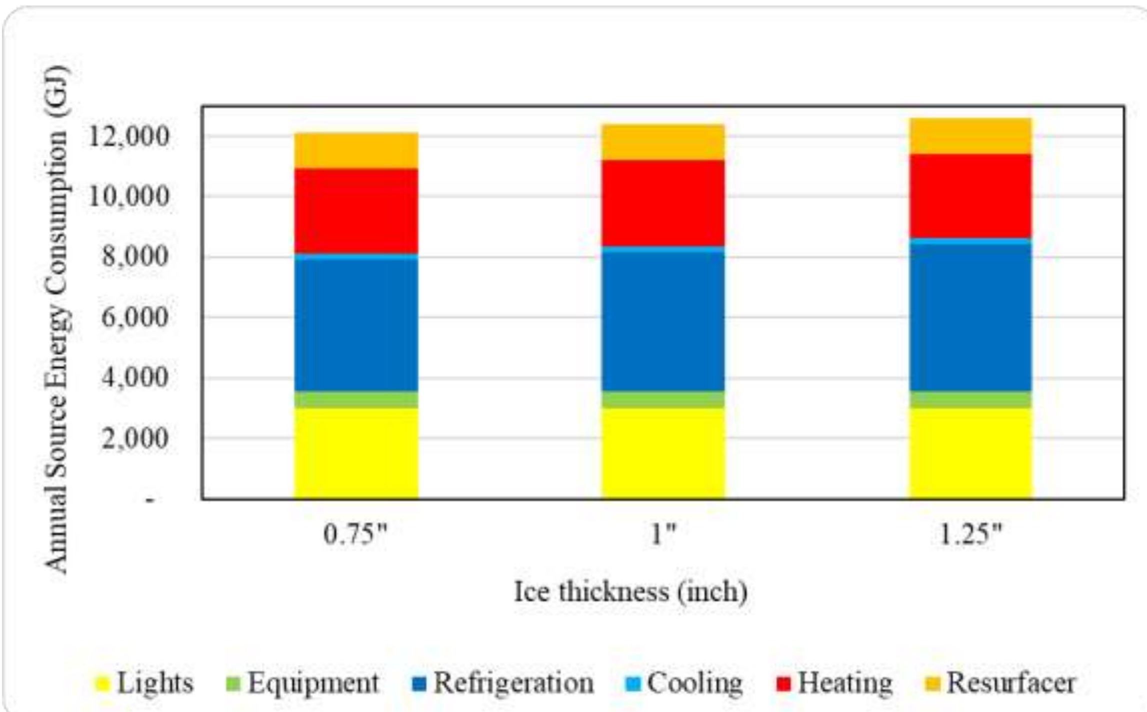


Figure 4. 15: Impact of occupancy on source energy consumption of the ice rink facility located in Denver, CO

4.13 Impact of Annual Operation Schedule

In this section, the impact of annual operation schedule on the energy performance of the ice rink facility is assessed for various US climates. For this analysis, the size of the concession area has been increased to better understand the impact of HVAC loads on the overall facility's electricity consumption due to variations in the climate. The basic layout of the ice rink facility described in

Chapter 3 remains the same with the total floor area of the ice rink zone area is 3000 m^2 (32,292 ft^2). However, the floor area of the concession zone is set at 3600 m^2 (38,750 ft^2). The ice rink is scheduled to operate throughout the year with no breaks. The lighting and equipment power densities remain as described in Chapter 3 with operation schedules shown in Figure 4.16. Occupancy schedule also remains the same as described in Chapter 3 and follows the annual schedule for the ice rink facility. In addition, the resurfacer's schedule remains unchanged.

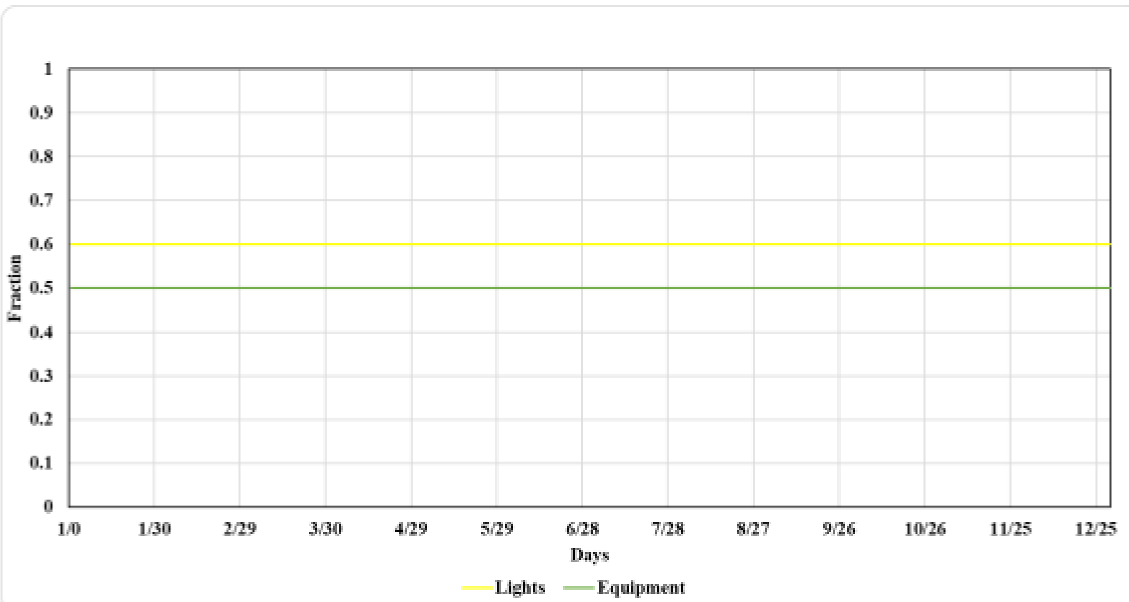


Figure 4. 16: Ice rink lighting and equipment schedules used for the sensitivity analysis specific to the annual operation schedule

Figure 4.17 shows the variation of monthly electricity use of the ice rink facility located in four US cities. In general, ice rink facilities located in hot climates consume more electricity than those in colder climates due to higher space cooling demands. The largest difference in electricity consumption between cities is 5 MWh and occurs in August between Phoenix, AZ, where the ice rink facility consumes the highest energy use and Denver, CO, where the facility has the least electricity demand. Figure 4.18 shows the energy end-use breakdown of the ice rink facility for the four US cities. The annual energy consumption is 24,123 GJ, 24,169 GJ, 24,230 GJ and 24,248

GJ for Newark, NJ, CO, Miami, FL, and Phoenix, AZ, respectively. As mentioned in Chapter 4, the energy end-uses specific to lighting, equipment and resurfacer are constant among all climates since they are based defined operation schedules and not on the climatic conditions. Climate impacts primarily HVAC energy end-uses including those associated with heating, cooling, and refrigeration. The l HVAC energy end-use is 4,736 GJ, 4,787 GJ, 4,835 GJ and 4,831 GJ for Newark, NJ, Denver, CO, Miami, FL, and Phoenix, AZ, respectively.

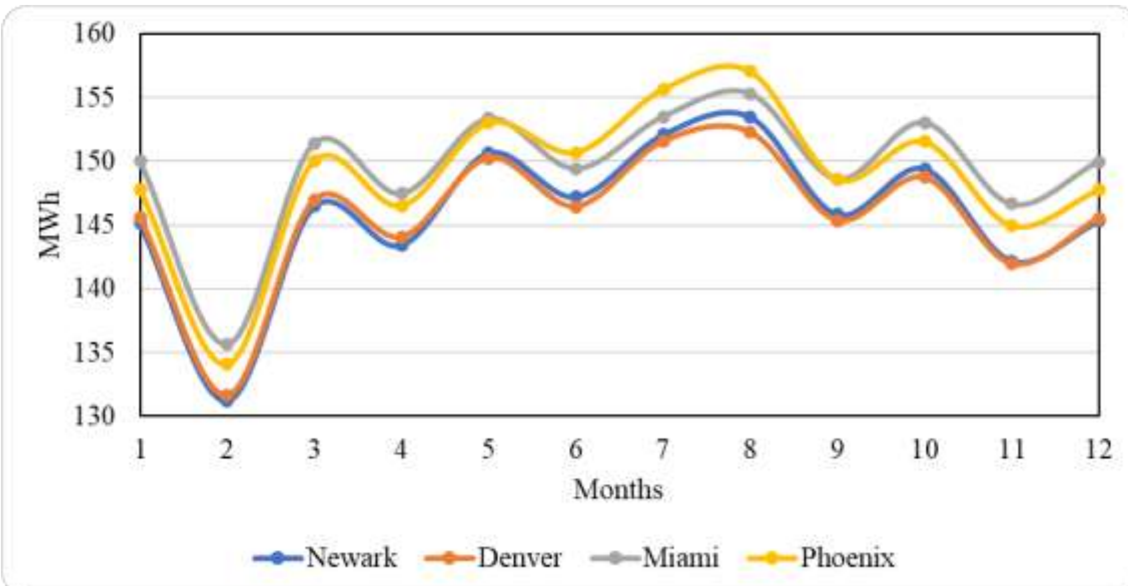


Figure 4. 17: Monthly electricity use in the ice rink facility located in different cities

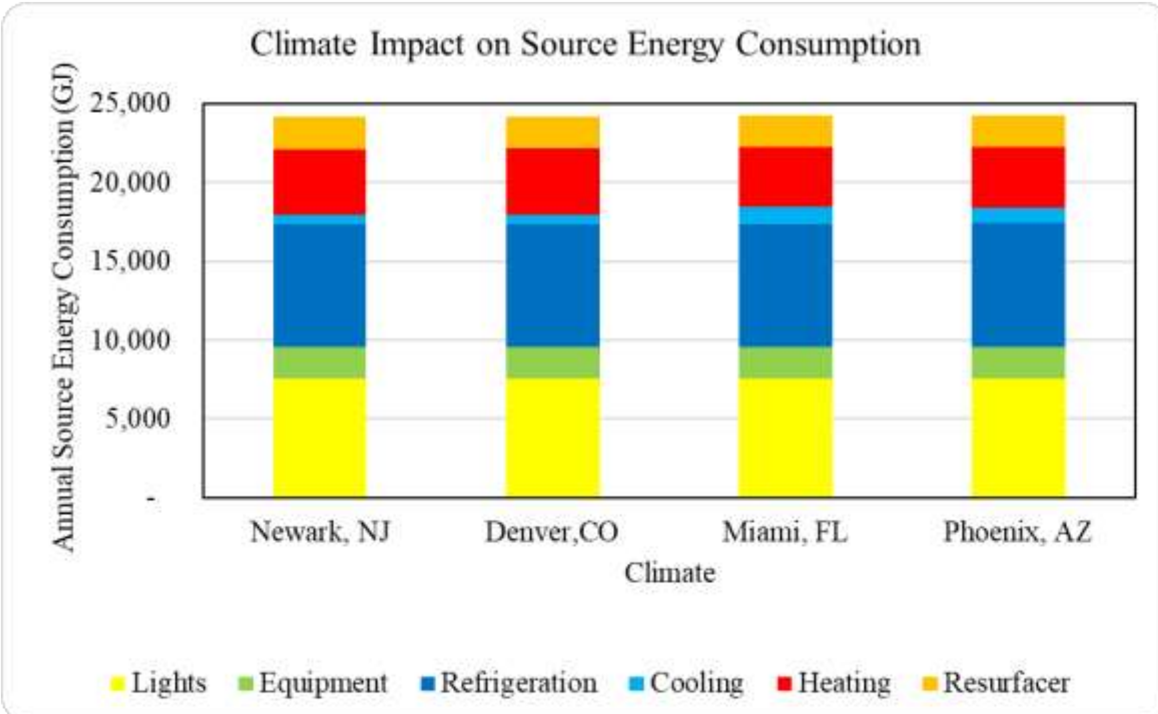


Figure 4. 18: Effect of climate on annual source energy consumption for the ice rink facility using an annual operation schedule

Figures 4.19 and 4.20 show the monthly profiles for energy end-uses specific to cooling and heating demands, respectively. When the ice rink facility is located in Phoenix, AZ, it consumes the most energy during the summer due to the high cooling loads. However, when the facility is located in Miami, FL, it requires the highest annual cooling demand and the lowest heating demand estimated at 1,064 GJ and 3,771 GJ, respectively.

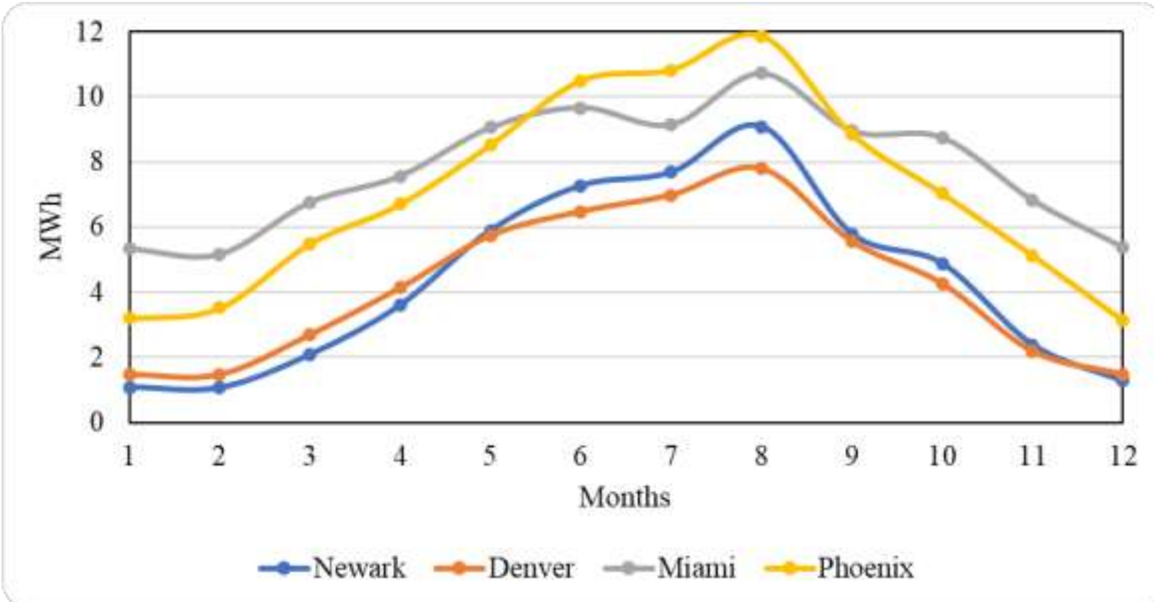


Figure 4. 19: Monthly cooling electricity end-uses for ice rink facilities located in different cities

On the other hand, when thte ice rink facility is located in Denver, CO, it experiences the highest annual heating and the lowest cooling demands with 573 GJ and 4,213 GJ, respectively. Figure 4.20 shows that ice rink facilities require heating for all cities even during the summer due to the ice rink cooling effects. Facilities located in Phoenix, AZ, and Denver, CO, have the lowest and highest heating demands respectively, during summer months. Figure 4.21 represents the monthly ice rink refrigeration energy end-uses for facilities located in four US cities considered in the analysis. The refrigeration energy end-use is almost constant for all cities since the same indoor zone conditions are maintained throughout the year. The slight variation in refrigeration energy end-uses between locations is due to heat gains and losses from windows as well as walls and roof surfaces. In addition, the number of days of each month significantly affects the refrigeration energy end-use as depicted in Figure 4.21. Months with 31 days have higher energy end-use than those with 30 days. February has the lowest consumption because it has 28 days.

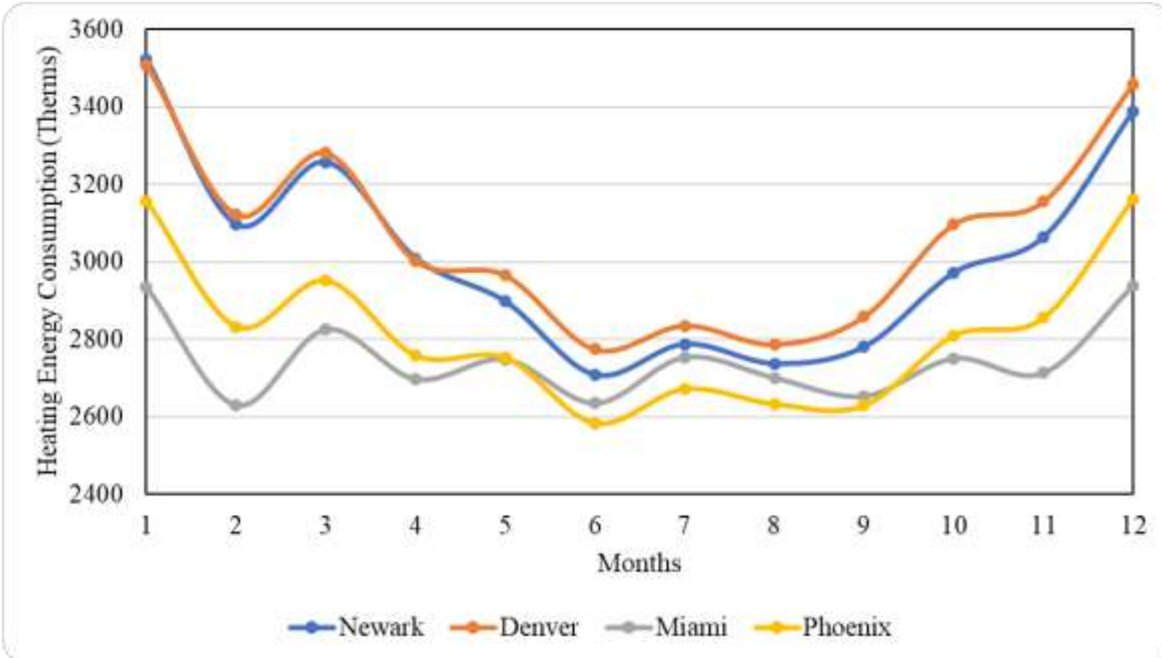


Figure 4. 20: Monthly heating energy end-uses for ice rink facilities located in different US cities

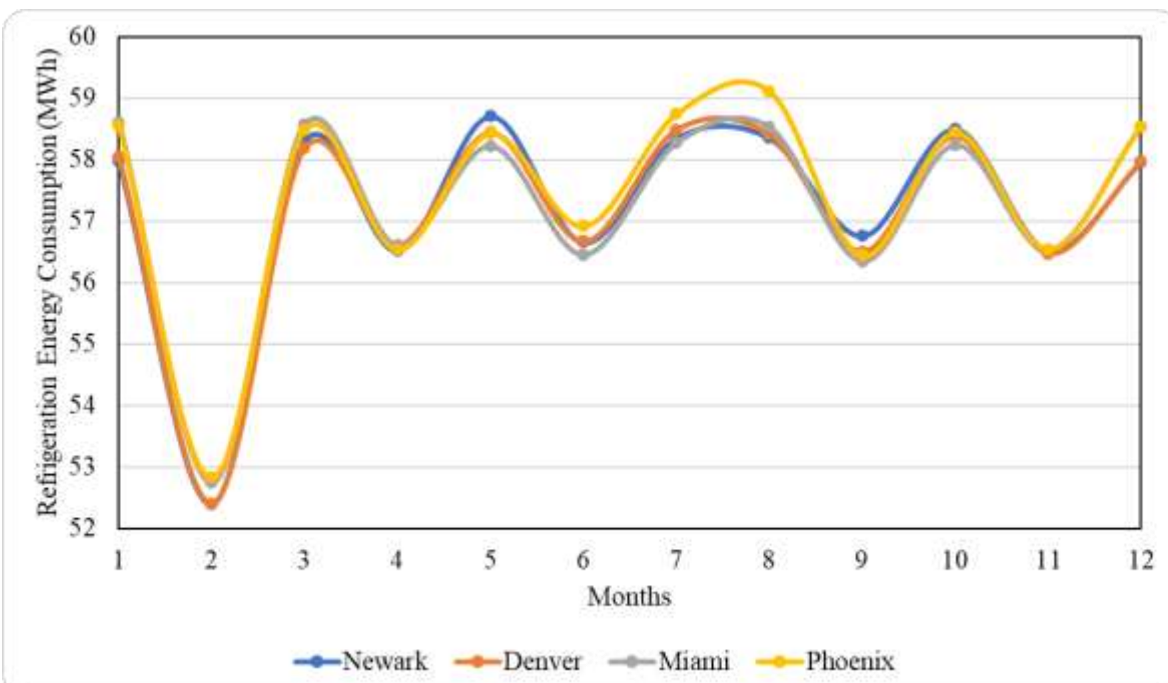


Figure 4. 21: Monthly refrigeration energy end-uses for the ice rink facilities located different cities

4.14 Energy Efficiency Analysis

In this section, the interactive effects of a set of energy efficiency measures (EEMs) are considered for the baseline ice rink facility located in Newark, NJ to estimate the potential energy savings that the facility can achieve. The selected EEMs used in this analysis are based on the most influential parameters determined from the series of sensitivity presented in the previous sections. Therefore, the location and size of the ice rink arena are kept the same as the baseline case. Moreover, the lighting and equipment power densities are not changed in this analysis. Table 4.1 lists all the measures that are considered in this analysis. The measure related to the resurfacer is based on the minimum acceptable specifications for a 30-mx60-m ice rink arena [13]. By implementing the listed EEMs, the annual source energy consumption for the ice rink facility is decreased from 12,258 GJ to 8,462 GJ leading to 31% savings in annual source energy. Figure 4.22 shows the impact of EEMs on the energy performance of the baseline ice rink facility.

Table 4.2: Energy Efficiency measures used in the analysis

	EEMs
Schedule	Seasonal
COP	3.5
CeilingThermal Absorptance	0.3
Indoor Temperature	16°C
Ice surface Temperature	-3°C
Resurfacer	0.4m^3 (105.7 gallons) Water temperature: 55°C (131°F)

The saving in energy consumption related to the ice rink refrigeration system is due mainly to the higher COP, lower indoor temperature, higher ceiling reflectivity and a resurfacer machine that uses lower temperature water. In addition to the reduction in refrigeration energy end-use, the lower indoor temperature setting led to a lower space heating thermal load. The refrigeration consumption is decreased to 47%. Space heating and resurfacer energy end-uses are lowered by 44% and 36%, respectively. Space cooling energy end-use has not changed significantly because of the seasonal operation schedule considered in this analysis The site and source EUI values for the ice rink facility change from 1.64 3.65 GJ/m^2 and 3.65 GJ/m^2 for the baseline to 1.07 GJ/m^2 and 2.52 GJ/m^2 for the energy efficient case. Since most of the considered EEMs are easy to implement, it can be concluded that proper design and operation of ice rink facilities can result in significant energy savings.

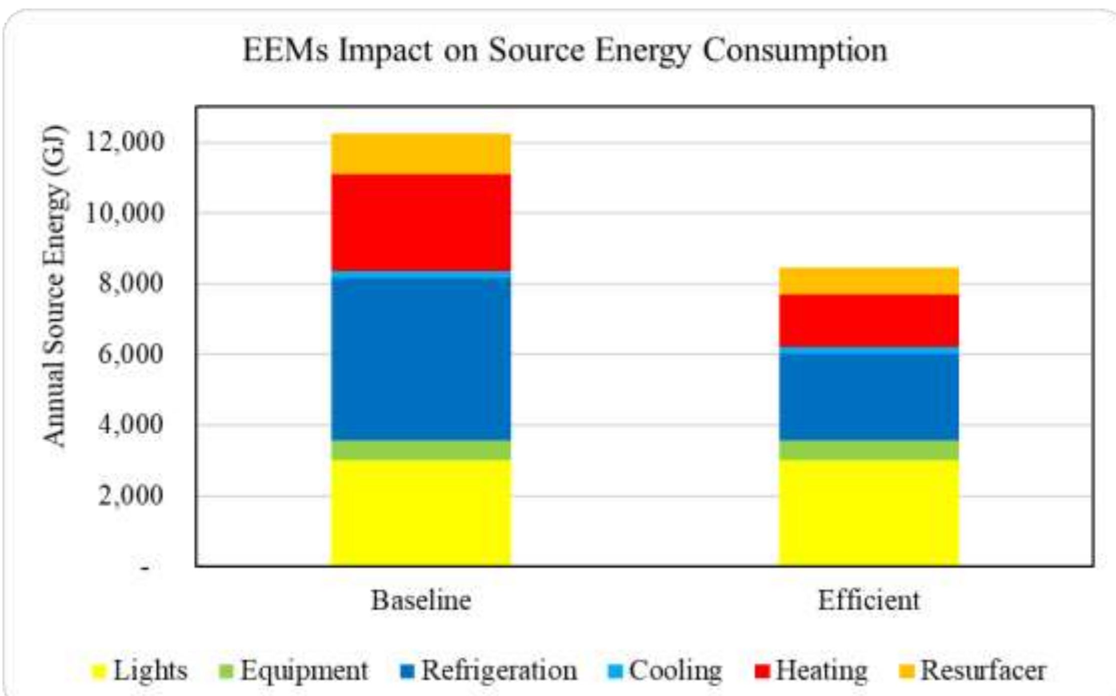


Figure 4. 22: Impact of selected set of EEMs listed in Table 4.1 on annual source energy consumption of the ice rink facility

4.15 Modeling a Refrigeration Chiller

In previous sections, an EnergyPlus ideal plant component "PlantCompoenent: TemperatureSource" was used to model the refrigeration system of the ice rink. The ideal plant component assumes 100% efficiency of different refrigeration system components including the chiller and the condenser. In this section, the impact of modeling specific components of the refrigeration system on energy performance of the ice rink facility is investigated. First, the simulation results using an air cooled chiller system to meet the refrigeration needs of the ice rink are analyzed and compared to those obtained using an ideal refrigeration system as discussed in previous sections. Then, the impact of the chiller sizing on energy consumption is evaluated. In this initial analysis, a 100-ton (351685 W) air cooled electric chiller is considered. Figure 4.23 shows the values of various input parameters used to model the refrigeration chiller system for the ice rink facility described and modeled in EnergyPlus in Chapter III

Field	Units	Obj1
Name		Main Chiller
Reference Capacity	W	351685
Reference COP	W/W	3.2
Reference Leaving Chilled Water Temperature	C	-10
Reference Entering Condenser Fluid Temperature	C	29.4
Reference Chilled Water Flow Rate	m3/s	0.4
Reference Condenser Fluid Flow Rate	m3/s	0.2
Cooling Capacity Function of Temperature Curve Name		Main Chiller RecipC.
Electric Input to Cooling Output Ratio Function of Temp		Main Chiller RecipEl
Electric Input to Cooling Output Ratio Function of Part L		Main Chiller RecipEl
Minimum Part Load Ratio		0
Maximum Part Load Ratio		1
Optimum Part Load Ratio		1
Minimum Unloading Ratio		0.25
Chilled Water Inlet Node Name		CondenserIn
Chilled Water Outlet Node Name		CondenserOut
Condenser Inlet Node Name		
Condenser Outlet Node Name		
Condenser Type		AirCooled
Condenser Fan Power Ratio	W/W	
Fraction of Compressor Electric Consumption Rejected I		1
Leaving Chilled Water Lower Temperature Limit	C	-12
Chiller Flow Mode		ConstantFlow
Design Heat Recovery Water Flow Rate	m3/s	0
Heat Recovery Inlet Node Name		
Heat Recovery Outlet Node Name		
Sizing Factor		

Figure 4. 23: EnergyPlus input parameters for the ice rink refrigeration chiller

Figure 4.24 compares the annual energy end-uses for the ice rink facility using the chiller system and the ideal system. The refrigeration system uses more energy than the ideal system due mostly to fans and pumps. Indeed, energy end uses for refrigeration, HVAC, lighting, and plug loads are similar for both models as noted in Figure 4.24. However, the annual energy demands for fans and pumps associated the refrigeration chiller system are estimated at 389 GJ and 1023 GJ, respectively. Both the fans and pumps are not accounted for when using an ideal refrigeration system. The ice rink facility consumed about 12,258 GJ and 16,546 GJ annually for respectively, the ideal system and the chiller system. Thus, the refrigeration chiller system increased the energy use intensity (EUI) of the ice rink facility to 4.9 GJ/m^2, that is, 32% increase from the baseline EUI of 3.69 GJ/m^2.

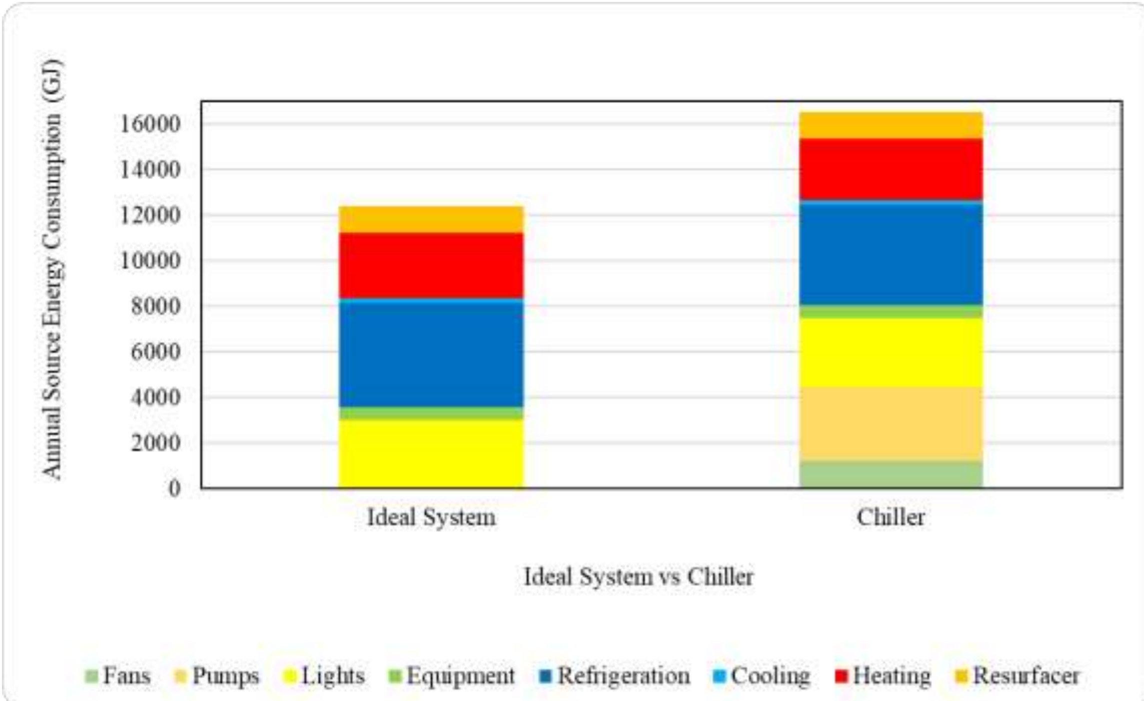


Figure 4. 24: Annual energy end-use distribution of the ice rink facility when modeled using both ideal and chiller refrigeration systems

Figure 4.25 indicates that both the ideal and the chiller refrigeration systems maintain the ice rink surface temperature close to the set-point of -3ºC. However, the use of the ideal system, with unlimited capacity, results in smaller fluctuations in surface temperature compared to the case of using the chiller system with a limited refrigeration capacity of 100-ton.

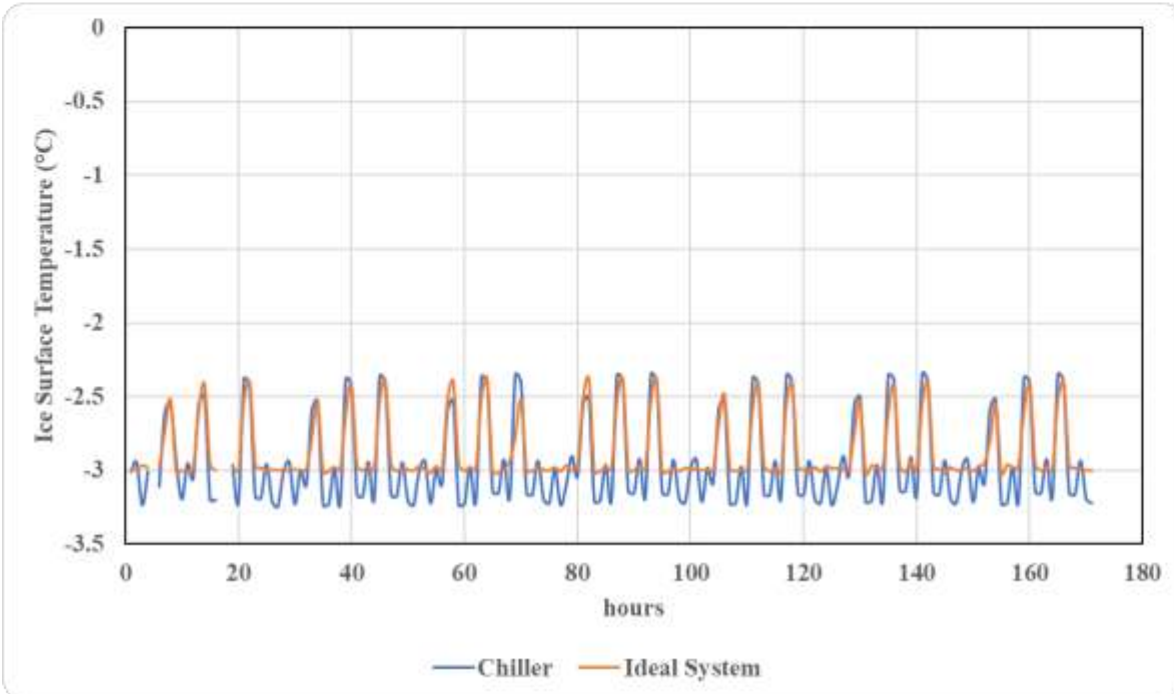


Figure 4. 25: Ice rink surface temperature during one week when using refrigeration systems with a chiller and an ideal system

4.16 Impact of Chiller Capacity

In this section, the impact of refrigeration chiller capacity is determined on the energy performance of ice rink facility. Specifically, different chiller capacities and refrigerant design volume flowrates are considered as listed in Table 4.3.

Table 4. 3: Chiller capacities and refrigerant volume flowrates used in the sensitivity analysis

Chiller Capacity	Refrigerant mass flowrate
50 tons (175843 W)	0.2 m^3/s (7 ft^3/s)
100 tons (351685 W)	0.4 m^3/s (14.1 ft^3/s)
150 tons (527528 W)	0.6 m^3/s (21.2 ft^3/s)

200 tons (703371 W)	0.8 m^3/s (28.25 ft^3/s)

As shown in Figure 4.26, increasing the chiller's capacity leads to higher energy consumption by the ice rink facility driven mostly by the fans and pumps. Indeed, the annual fan energy demand is estimated to be 633 GJ, 1,233 GJ, 1,698 GJ, and 2,205 GJ for 50-ton, 100-ton, 150-ton and 200-ton chillers, respectively. In addition, the annual energy consumption attributed to the pumps is determined to be 1,667 GJ, 3,240 GJ, 4,456 GJ and 5,786 GJ when the chiller capacity is set to 50-ton, 100-ton, 150-ton, and 200-ton, respectively. Therefore, the annual energy consumption of the ice rink facility has increased from 14,330 GJ to 20,016 GJ, that is 40% when the chiller capacity is selected to be 4 times larger from 50-ton to 200-ton.

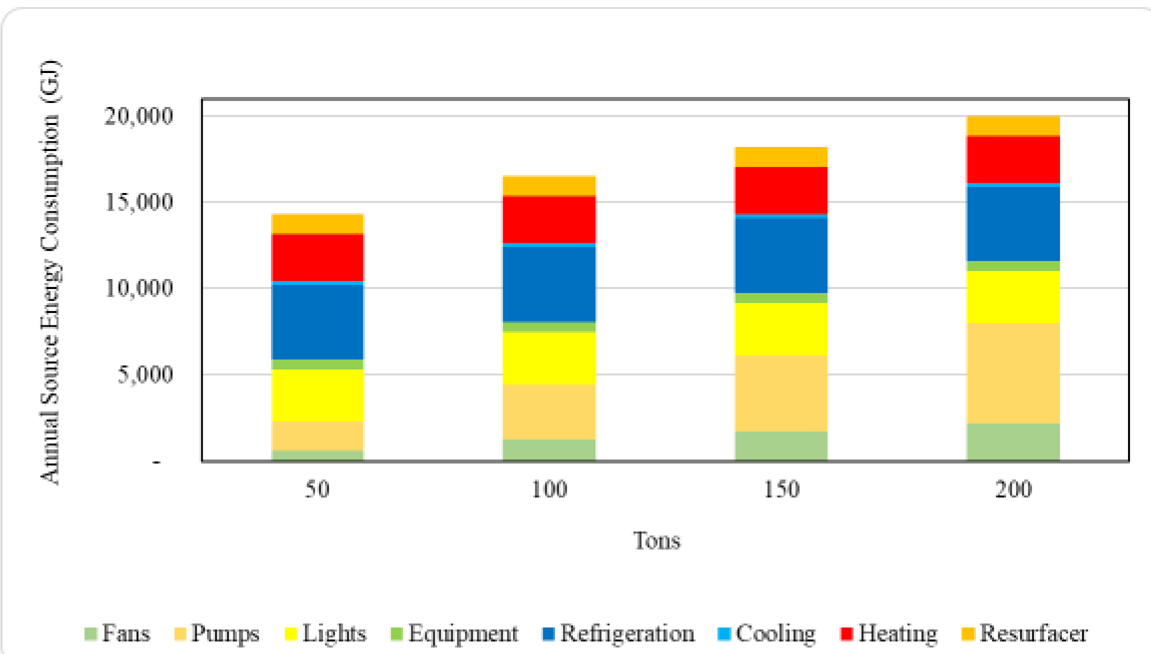


Figure 4. 26: The impact of increasing chiller's capacity on the annual energy end-uses for the ice rink facility

Figure 4.27 illustrates the time-variation of ice rink surface temperature for two chiller capacities during one week. The results of Figure 4.27 indicate that when the chiller capacity is 50-ton, the refrigeration system struggles to maintain the ice surface at -3 °C temperature setpoint compared to the case of the 200-ton chiller which has sufficient capacity to handle any sudden increases in refrigeration loads.

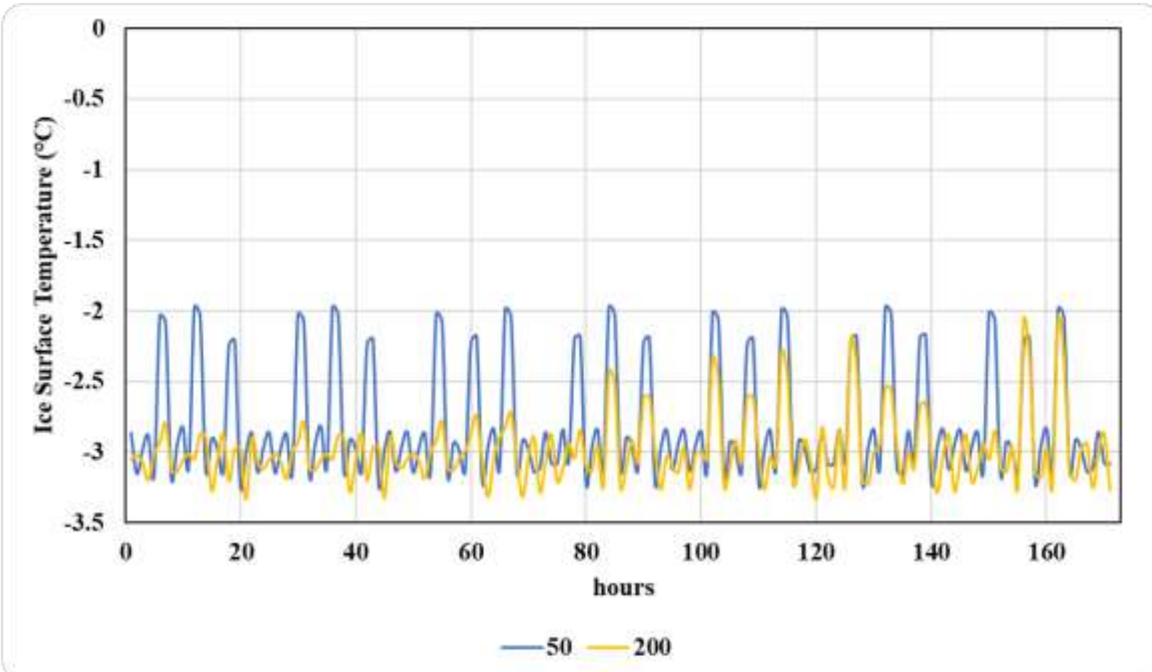


Figure 4. 27: Time-variation of the ice surface temperature attributed to 50-ton and 200-ton refrigeration system.

Chapter V: Summary and Conclusions

Facilities that house ice rinks represent an energy challenge due to the high consumption. A need for an accurate modeling of ice rinks is needed to achieve the optimum design. Therefore, a new ice rink module has been developed and integrated in EnergyPlus, a whole building simulation tool. The new module successfully interacts with different building parameters such as zone air temperature, HVAC and electricity, etc. The baseline annual consumption is 1533 MWh/year and the source EUI is 3.65 GJ/m^2. The baseline model results are realistic and within the energy consumption range mentioned in previous studies [5,9]. A sensitivity analysis was conducted to help test the impact of 13 parameters on the ice rink feature. The results of the sensitivity analysis were reasonable. The following list summarizes sensitivity analysis findings:

- Using an annual operation schedule instead of a seasonal one leads to 50% increase in energy consumption.
- Replacing the baseline's refrigeration system of a COP 3.5 with a 2.5 one leads to 28.6% energy savings.
- Based on the EnergyPlus simulation results, lowering the ceiling reflectivity provides only less 2% savings in total energy consumption.
- The zone air temperature plays an important role in HVAC and refrigeration loads. Lowering the indoor air temperature from 22 °C to 16 °C leads to 17% in energy savings.
- The impact of different climate on the total energy consumption was not significant because of the ice rink cooling effect is high in all climates.
- Lowering the ice rink surface temperature to -5°C from -3°C results in 4.1% increase in energy consumption due to the higher refrigeration load.

- The sensitivity analysis shows that a 26-mx56-m ice rink instead of the baseline's size of 30-m x 60-m provides 10.3% savings in energy consumption.
- An airtight ice rink facility helps avoiding higher energy consumption. Increasing the baseline's infiltration to 0.5 ACH leads to an 18% increase of energy savings in cold climates such as Newark, NJ and Denver, CO.
- The simulation results show that a well-insulated building can save energy but over-insulating it does not provide significant savings.
- Higher occupancy leads to high energy consumption. Increasing the occupancy to twice the baseline's value can lead to 2.3% increase in total energy.
- Using a 0.01905m (0.75") instead of 0.0254m (1") ice layer results in a 5% energy savings.
- Simulation results show that an ice rink with an annual operation schedule in cold climates consumes less electricity than ones in hotter climates.
- Applying the best performing parameters to the baseline model led to 31% savings in energy consumption.

A limitation encountered while modeling the ice rink facility in EnergyPlus consists of the need to create one floor with two functions (i.e., building's floor, and ice rink's floor). Although the developed ice rink module yielded acceptable total energy consumption results that matches the literature, more sensitivity analysis is needed to assess the additional modeling needs for EnergyPlus specific to ice rink facilities.

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