ENERGY USAGE PREDICTION MODEL 
COMPARING INDOOR VS OUTDOOR ICE RINKS

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Master of Science Thesis EGI-2012-010MSC
Division of Applied Thermodynamics and Refrigeration
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KTH School of Industrial Engineering and Management
Energy usage prediction model comparing indoor vs. outdoor ice rinks

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Abstract

Indoor ice rinks use 1091 MWh per annum for ice hockey based on statistics from over 100 Swedish ice rinks (Stoppsladd, 2011). The refrigeration system contributes 35 to 75% (Rogstam, 2010) of total energy usage in ice rinks with average value of 43% (Stoppsladd, 2010) for indoor to 75% for outdoor ice rinks.

The basic aim of project is to reduce energy consumption in Swedish ice rinks and scope is for indoor and outdoor ice rinks in cold and mild summer climatic conditions like Sweden. To achieve target of energy reduction in ice rinks actual heat loads on outdoor bandy ice rink are being estimated along with performance analysis of refrigeration machine. The refrigeration system, heat loads on ice surface and their correlation is studied and analyzed in detail for Norrtälje Outdoor bandy ice rink for four warm days and whole season 2010-2011. The tricky and significant task of validation of input climate data for accurate heat loads calculations is completed with Swedish Metrological & Hydrological (SMHI) climate model data, correlations and related web based geographical data.

The heat loads (conductive, convective and radiant) on outdoor bandy ice rink are calculated through thermodynamic relations with validated input climate data and measurements where as refrigeration system performance is monitored and analyzed with ClimaCheck(CC) instrumentation. The average cooling capacity is calculated for four warm days by CC internal method and actual cooling energy produced is obtained by practically assumed COP of system with aid of MYCOM compressor software. The cooling capacity and heat loads on ice surface are compared and analyzed considering energy usage affecting parameters and weather parameters like temperature, wind speed, relative humidity and solar load. The convection and condensation are contributing 75%, radiation 18%, ice resurfacing 4% and ground and header heat gain 3% to total heat loads on ice sheet for whole season. The deviation between total cooling energy produced by refrigeration machine and total heat load energy is found 19% and 27% for four warm days and whole season 2010-2011. The deviation is due to overestimation of heat losses from compressor’s body, compressor’s on and off operations, overestimated radiation heat load due to unmeasured negative radiation and lack of actual ice resurfacing heat load evaluation.

The developed model in MS Excel allows comparison of field climate data with SMHI model data, indoor and outdoor ice rinks in terms of predicted energy usage by refrigeration system and in total and acts as decision tool to choose for building an indoor/outdoor ice rink.

Key words: Ice rink, Refrigeration, Energy Efficiency, Heat Load, Cooling capacity, Energy Usage, Measurement, Climate Change, Model
Acknowledgement

This thesis is a part of the project Stoppsladd managed by Energi & Kylanalys together with the Swedish Ice hockey association and financed by the Swedish Energy Agency.

First of all I am sincerely grateful to my master thesis supervisor Jörgen Rogstam for his continuous support as without his technical guidance and time to time moral support I could not do so. He always helped me whenever I got stuck up during the work with his useful advices and directed me towards right track to achieve the target till end.

Secondly I acknowledge my local supervisor at KTH Samer Sawalha who was always on back up to guide me at odd times. I am grateful to him for his technical advices which saved my time and gave me right approach to think and his help in EES programming. Furthermore I should not forget Mazyar Karampour’s discussions and help during every phase of my project as he was already working on the same target but with indoor ice rinks. His sincere help in form of technical suggestions and data always facilitated me and he was throughout a helping hand for me.

Kenneth Weber from ETM Kylteknik always gave me really useful technical advices and always corrected and channelized me whenever things were going in wrong dimension as he is very closely linked with monitoring and installations of refrigeration system performance analyzer at Norrtälje Sports Centrum. I am really obliged to him. I am also very thankful to John Ekwall from Customer Services SMHI who really helped me in extracting climate data from model, analyzing and comparing it with field measurements radically. His immediate response and technical guidance made me possible to crack the hard nut of climate data validation. Klas Berglöf and Jakob Månberg’s technical help also facilitated me in resolving CC advanced software template and downloaded files processing issues.

And in last I am grateful to especially my program director Andrew Martin and Examiner Professor Björn Palm for administrative issues and giving me opportunity to do my thesis in Division of Applied Thermodynamics and Refrigeration and.
# Nomenclature

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<td>A</td>
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<td>Electric power (W)</td>
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<tr>
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1 Introduction

1.1 Background

The aim of the project is to gather information and knowledge on the energy usage in ice rinks and the ultimate goal of this project generally is to reduce energy usage in Swedish ice rinks.

The use of energy in ice rinks (electricity and heat) is widespread in Sweden due to factors like length of season, number of activity hours in the rink and building characteristics. Since the refrigeration system is a major energy user with an average of 43% (Stoppsladd, 2010) of total energy used, it is wise to decrease heat load on the ice and ultimately reduce the energy usage of refrigeration systems. The work has been done and still in progress to refine the estimate of energy usage for indoor ice rinks.

1.2 Challenge

The climate change and usage pattern of ice rinks promotes the development towards indoor ice rinks rather than classical outdoor arenas. The local clubs of many municipalities want to go indoor due to extended season. So it is hard to predict the cost of operation which depends on seasonal differences of weather and many parameters affecting the energy usage. Labour cost for building up the ice and extra maintenance due to weather conditions is also one of the parameters which affects.

1.3 Objectives

The aim is to develop models allowing comparing indoor and outdoor ice arenas depending on local specific conditions. To accomplish the aim of the study, the following objectives have been set:

- To ensure validation of input climate data locally with Swedish Meteorological & Hydrological Institute (SMHI) climate data for accurate measurement of heat loads on ice rink surfaces.
- To evaluate Norrtälje outdoor bandy ice rink, Sweden with ClimaCheck field instrumentation enabling to monitor cooling capacity, COP and heat of rejection by refrigeration system.
- To calculate, compare and analyze cooling capacity & heat loads on ice surface considering specific weather parameters like temperature, wind speed, relative humidity and solar load.
- To develop a model allowing comparison of indoor and outdoor ice rinks in terms of energy usage by refrigeration system and in total.
- The developed model by answering above mentioned question should support the decision to build an indoor or outdoor ice rink.

1.4 Scope and Limitations

The scope of the project is general and developed model can be used for any country by defining weather as well as technical parameters of ice rink. The climate input and technical parameters of Swedish ice rinks are used in the developed model. The cold winter and mild summer for indoor and winter climate conditions for outdoor ice rink are taken into account as season with SMHI model data as reference. The refrigeration system, heat loads and their correlation is studied in detail.

The limitations used for experimental results are further discussed in relevant chapters.

1.5 Methodology

The thesis work starts with literature review of ice rink design technology and then follows by weather data extraction and validation with SMHI model for Norrtälje outdoor ice rink. The heat load on ice surface is calculated after necessary corrections in input climate data for few warm days and on daily
averaged basis for season 2010-2011 started from 13\textsuperscript{th} October to end of February, 2011. The refrigeration system cooling capacity is calculated with aid of compressor’s manufacturer software through internal method by ClimaCheck for performance of cooling system for same season period. The cooling capacity provided by evaporator is compared and analyzed with heat loads to ice. The model is developed through energy usage data in total and by refrigeration system specifically for indoor and outdoor Swedish ice rinks to predict energy usage considering energy affecting parameters and local standard geographical weather parameters.
2 Ice Rink Technology & Energy System

2.1 Ice Rink-Norrtälje Bandy

The outdoor ice rink studied in research is located at Norrtälje which is one hour north of Stockholm. Norrtälje Sports Centrum is having two ice rinks; indoor ice hall is for hockey and figure skating and Bandy\(^1\) Ice rink (outdoor ice rink) as shown in Figure 2-1.

![Figure 2-1 Norrtälje Sports Centrum a) Ice hockey hall b) Bandy Ice rink](image)

It is part of Norrtälje Sports Centrum having area of 6000m\(^2\) (60×100m) which is lit in the evenings. It is approved for international matches and works from mid October to mid April. The normal activities on it are training activity for players, leisure skating by school girls and boys on week days and everybody on weekends.

2.2 Ice rink refrigeration system

The indirect system for ice rink refrigeration system is most conventionally used. The reason of it is compact design of refrigeration system with small evaporator and extremely small refrigerant charge for large ice rink system. In the direct system refrigerant is pumped below the ice pad and then whole refrigerant distribution pipes serve as a large evaporator due to which method is rarely used for huge amount of refrigerant charge required. The most used refrigerants for direct systems like R-22 is banned due to its global warming potential in many countries and ammonia cannot be used in such large systems like ice rinks due to charge limit relevant to its hazards.

In this indirect system layout shown in Figure 2-2 a primary refrigerant cools secondary refrigerant and then distribution system circulates this secondary brine below the ice pad and returns it back to evaporator. (Karampour, 2011)

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\(^1\) Bandy is a ball sport, team sports and winter sports. It is played on ice between two eleven player teams with a massive ball and clubs. It has greatest popularity in Sweden, Finland, Russia and Norway. It is played in two halves of 45 minutes each in organized level.
2.3 Ice rink Energy system-Norrträije Outdoor Bandy

The understanding of energy flows across the boundaries of ice rink energy system is significant from system effectiveness analysis as whole system can become efficient when residual waste energy flows from different components are dealt for its needs. (Makhnatach, 2010)

The energy system of ice rink consists of many systems like refrigeration, heating, ventilation, dehumidification and lighting etc. The first three mentioned require distribution systems powered by pumps and fans for energy/mass transfer. (Karampour, 2011)

For Norrträije outdoor bandy rink where season length is normally 3-5 months there is need for cooling and heating to provide temperature ranging from -4°C (ice surface) and +60°C (DHW) for toilets and shower rooms for players and +25°C for ice resurfacing. The Norrträije outdoor bandy ice rink energy system is discussed below:

Refrigeration system produces ice for large ice sheet surface with nominal cooling capacity around 1200kW at -14°C evaporation and 30°C condensation temperature. It is electricity powered vapour compression indirect system and details of Norrträije refrigeration system for outdoor bandy ice rink are explained in details further.

Heating system is provided domestic hot water (DMW) by district heating for bandy ice rink under consideration but the energy efficient, cost effective and environment friendly method is to utilize heat rejected by refrigeration system (condenser and desuperheater). Actually Norrträije’s refrigeration machine for outdoor bandy ice rink produces cooling on warm days of whole season (3-5 months) when heating is not required for ventilation, space heating, ice resurfacing (warm water with temperature of 25°C used) and floor heating except for DMW for toilets, showers and locker rooms for players. Due to unavailability of figures of energy used for heating it is not considered in Outdoor bandy ice rink energy system of Norrträije.

Ventilation system is not needed for outdoor ice rink (large ice sheet: bandy ice rink).

Dehumidification system is not required for outdoor ice rinks due to absence of metallic and wooden structures having risk of getting corroded and rotten by humidity and difficulty to remove humidity from huge volume of air creating fog on ice surface and adding heat loads as condensation. (Karampour, 2011)

Lighting is required for few hours in the evening for various activities on outdoor bandy ice rink. The efficiency of lighting system depends on input wattage, life time and efficiency of the ratio lumens to input wattage of the fixtures installed. (Karampour, 2011).

The ice rink energy system with heat recovery for various applications like floor and ventilation heating and hot water storage for resurfacing water is shown in Appendix 11.10.
3 Weather Data Validation

The input weather data validation is the prerequisite for calculating accurate heat loads on ice surface which was also one of tricky task in this thesis work. The field measurements of climate parameters are obtained by ClimaCheck which are being compared with SMHI measurements at Svanberga. Svanberga is SMHI closest station at Norrtälje which is at 11.8km by car to north of Norrtälje Sports Complex.

3.1 ClimaCheck Measurements

The measurements for weather parameters are taken by downloading raw data files from ClimaCheck online (ClimaCheck online). It is monitoring of refrigeration and heat pumps in real time over internet which uses REFPROP library from NIST program (ClimaCheck online). The measurements for all below mentioned parameters are recorded at Swedish local Time (SLT).

3.1.1 Wind

Wind is measured at Norrtälje outdoor bandy by Wind Speed Detector (Produal Oy, 2004) by Produal Oy as shown in Figure 3. It is installed at height of 4 meters approx. from ground. The detector shown by Figure.3-1 can measure wind speed as well as outside temperature and used for heating and ventilation systems where temperature gets affected by wind.

The wind speed measurement range is 0-20 m/s and temperature from -50°C to 50°C. The strange values of wind for few minutes are omitted for wind data and average values are used instead. The deviation for wind speed is less than 20% of the measurement and for temperature is less than 0.5°C at 25°C.

3.1.2 Temperature & Relative Humidity (RH)

The outdoor temperature and RH are measured by Outdoor Humidity Transmitter KLU 100 (Produal Oy, 2007). It is installed at 3 meters above ground and measures both relative humidity (RH) and temperature shown in Figure 3-2. The range for RH is from 0-100% and for temperature is from -50°C to 50°C. The accuracy of transmitter for RH is ±2%(0…90%RH/25°C) and temperature is ±0.5°C /0°C.
3.1.3 Solar Radiation

Global Irradiance (G) is measured in W/m² by Silicon Pyranometer Smart Sensor S-LIB-M003 (Onset Computer Corporation, 2001-2010) shown in Figure 3-3. It is installed on site at height of 4 meters approximately from ground. Its measurement range is 0-1280 W/m² within spectral range of 300-1100 nm. It has accuracy within ±10 W/m² or ±5% whichever is greater in sunlight. The azimuth error is less than ±2% at 45 degrees from vertical, 360 degrees rotation.

The sensor is calibrated for use under natural sunlight and measurements errors are small as compared to using them under artificial sunlight like within plant canopies, in greenhouses which may have significant errors in results.

3.2 SMHI Measurements

SMHI is Swedish Metrological and Hydrological Institute. It is government agency which supply forecasting and decision support for people and business depending on weather, water and climate. It is a body of experts in meteorology, hydrology, oceanography and climatology.

The climate data for validation of field measurements of weather parameters is extracted by SMHI database model. The closest weather station for measuring outdoor temperature, wind speed and relative humidity is at Svanberga, Norrtälje, Sweden which has been compared to field measurements by ClimaCheck weather station.

3.2.1 Wind

Wind speed is measured at sea-level which is 10 meters above ground level. The maximum wind to Norrtälje comes from Southwest and West direction which is also shown in Figure 3-4 attached for percentage wind from different direction from 1961-2004.
The wind speed measured at Svanberga station (SMHI) is usually higher due to buildings and forest present in between station and Norrtälje outdoor bandy field as local topography plays vital role in affecting wind speed.

### 3.2.2 Temperature & Relative Humidity

Temperature and relative humidity are measured at sea level which is 2 meters above ground at Svanberga station. These are extracted as UTC as time reference but then recorded as UTC+1h (Swedish local time) during winter for bandy ice rink season (Mid October to February) with correction.

### 3.2.3 Solar Radiation

Global Irradiance (G) is extracted from STRÅNG-a mesoscale model for solar radiation on hourly time series data basis (SMHI). The STRÅNG data used from SMHI was produced with support from the Swedish Radiation Protection Authority (SSM) and the Swedish Environmental Agency (SEA).

It is obtained by specifying longitude as 18° 38' 45" E and latitude as 59° 50' 08" N for Norrtälje, Stockholm. The average hourly values of G are being obtained with specification of UTC+1h as Swedish local time for season from 13th October-28th February, 2011 for outdoor bandy ice rink from model. The quality of output on global radiation by STRÅNG model is compared through measurements from SMHI (metrological) radiation network (SMHI).

### 3.3 Corrections for Parameters

The field measurements for weather parameters are used to calculate heat loads on ice with some corrections in wind speed and global radiation measurements.

#### 3.3.1 Wind Speed

To calculate convection and condensation heat load on ice, the wind speed of air very close to ice surface is required so CC wind speed measurement is corrected which measures the wind speed of air at approximately 4 meters above ice surface. So wind speed \((v_w(h))\) at \(h=10cm\) above ice is obtained by taking CC value as gradient wind speed \((v_{ref})\) at gradient height \((z_{ref})\) of 4 meters. The value of \(\alpha\) is taken as 0.10 for neutral air above open water surface. The formula used is shown in equation below. (Heir, 2005)

\[
v_w(h) = v_{ref}\left(\frac{h}{h_{ref}}\right)^{\alpha}
\]  

(3.1)

Where,

- \(v_w(h)\) = Velocity of wind at height \(h\)
- \(v_{ref}\) = Velocity of wind at some reference height \(h_{ref}\)
- \(\alpha\) = Hellman exponent. Its value for neutral air above water surface is 0.10

#### 3.3.2 Temperature

The ice surface temperature measured as field measurement is lower than actual ice surface temperature. The reason behind it is that probe which is frozen with ice to measure ice surface temperature is at 3cm below ice surface. So the formula used to find \(t_{surf}\) is discussed here,

\[
q = \frac{k}{\alpha}(t_{surf} - t_{sensor})
\]  

(3.2)

Where

- \(q\) = heat load on ice surface
\( k = \text{thermal conductivity of ice} \)

\( x = \text{thickness of ice} \)

\( t_{surf} = \text{ice surface temperature} \)

\( t_{sensor} = \text{ice temperature at sensor location} \)

For Norrtälje Bandy \( \Delta t \) calculated for correct \( t_{surf} \) where

\( x = 0.03 \text{m}, \)

\( k = 2.25 \text{W/m-K}, \) Value of water at \(-5^\circ\text{C}\) (Source: The Engineering Toolbox)

\( q = \text{sum of convective, diffusion, radiation and ice resurfacing heat loads} \)

The value of \( \Delta t \) is found for four warm days of October, 2010 and results are shown in Table 3.1.

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</tbody>
</table>

It is obvious from these \( \Delta t \) values that it is 2 \(^\circ\text{C}\) on average but for whole season calculations there are many factors which influence the ice surface temperature like negative radiation in nights with clear sky which ultimately reduce the daily radiation heat load and outdoor temperature etc. So finally +1\(^\circ\text{C}\) added to sensor temperature \( t_{sensor} \) of ice for correction of ice surface temperature \( t_{surf} \) in all heat load calculations for 4 warm days as well as whole season heat load calculations.

### 3.3.3 Global Radiation

The negative values by control system error for global radiation are taken as zero as negative radiations (ice to sky) are not measurement on field.

### 3.4 Comparison of Measurements

The validity of weather data is really significant to calculate accurate heat loads to ice surface. The comparison of field by ClimaCheck and SMHI measurements of weather parameters is presented below as sample for warm days in season.

![Wind Speed-November 1st, 2010](image.png)
The difference in wind speeds recorded for 1st Nov, 2010 in Figure 3-5 attached as an example between CC and SMHI trend line (red and blue lines) is significant. SMHI speed measurement is more as it is close to Lake Erken as compared to CC one which is less due to the buildings, hills, forests and local topography in between a distance of 11.8 km by car from Svanberga (SMHI station) to Norrtälje sports complex.

Secondly wind meter installed by CC on site only measures wind from only two directions South and West as we expect most of wind from these so may be due to unavailability of speed recorded from other directions it is less than SMHI value. The final corrected wind speed of air for which method has already been explained taken at 10cm above ice surface for load calculation is purple line shown in Figure 3-5.

For RH same like for wind speed we can expect more relative humidity near Lake Erken at Svanberga rather than at Norrtälje sport complex which is far from lake. The difference is not significant which can be verified by plot shown in Fig. 3-6.
The RMSE error when comparing SMHI hourly model data with point observations is approximately 30% for the global radiation which is approximately the same difference between CC and SMHI measured radiation shown by Fig. 3-7.

The average difference in outdoor temperature is almost 0.6-6 K for four warm days of October in season 2010-2011 as local topography and buildings affect considerably like for wind speed and RH. The average temperature difference for 1st Nov, 2010 is 0.6K shown in Fig. 3-8 attached as an example so local measured value of temperature by CC is used.
4 Load Calculations

4.1 Convective Loads

4.1.1 Load due to Heat transfer

The weather affecting parameters for convective heat load are air velocity, relative humidity and air temperature near and far from ice surface. The convective heat load is calculated with help of estimated convective heat transfer coefficient given by (ASHRAE, 2010) as,

\[ \alpha_c = 3.41 + 3.55V \]  \hspace{1cm} (4.1)

Where

\[ \alpha_c = \text{convective heat transfer coefficient, between the surface and air, W/ (m}^2\text{-K)} \]

\[ V = \text{air velocity over the ice, m/s} \]

The effective heat load is calculated as,

\[ Q_c = \alpha_c A (t_a - t_i) \]  \hspace{1cm} (4.2)

Where

\[ t_a = \text{temperature of surrounding air, } ^\circ\text{C} \]

\[ t_i = \text{temperature of air close to ice surface, } ^\circ\text{C} \]

\[ A = \text{area of ice sheet surface, m}^2 \]

The effective convective heat load is found on average as 45% of total heat load on ice.

4.1.2 Load due to mass transfer

The dehumidification process (important for locations with high ambient wet-bulb temperatures) lowers the load on ice-making plant as it reduces condensation and fog formation. The heat transfer due to mass transfer which is due to latent heat of condensation of water vapour (convective mass transfer) associated with a heat transfer to or from the surface where it occurs is calculated as

\[ Q_d = \alpha_d (t_a - t_i) \]  \hspace{1cm} (4.3)

Where

\[ \alpha_d = \text{diffusion/condensation heat transfer coefficient, W/ (m}^2\text{-K)} \]

\[ \alpha_d (\text{frost}) = 1740, \Delta P_{H2O} \frac{\Delta t}{\Delta t}, \alpha_c \]  \hspace{1cm} (4.4)

\[ \Delta P_{H2O} = \text{difference between the partial pressure of the water vapour in the air and air in the boundary layer very close to ice surface} \]

\[ \Delta t = \text{difference between temperature of air and air very close to ice surface} \]

For the ice surface temperature below 0°C the deposits on the surface are in the form of ice. So value of constant is taken as 1740 which has included latent heat of condensation from water vapour to water (frost) as well as latent heat of fusion of water from liquid to solid ice (Eric Granryd, 2005)

4.2 Radiation Load

Solar radiation is taken from field measurements at Norrtälje Outdoor bandy by ClimaCheck. It is
measured by Silicon Pyranometer smart sensor (S-LIB-M003) which is installed at height of 4 meters approximately on site. It measures global radiation in $W/m^2$ within $\pm 10W/m^2$ or $\pm 5\%$ whichever is greater in sunlight.

### 4.3 Conductive Loads

#### 4.3.1 Heat gain from ground & header

The heat gain from ground heat and header is taken as 3% of total heat load on ice (ASHRAE, 2010). In Norrtälje outdoor ice rink we have insulation below the rink above sand and gravel bed on ground of polyurethane (PU). As it is outdoor ice rink and season length is maximum 5-6 months so heating pipe is not needed for avoiding frost heaving for maintaining usable ice surface, building structural integrity and safety of users.

#### 4.3.2 Heat gain from coolant circulating /brine pumps

The nominal capacity for brine pump in Norrtälje Outdoor bandy refrigeration system is 18.5kW. The speed is controlled by frequency converters. The compressors work on temperature of brine out from ice pad (brine in to evaporator) with set point of $-7 \pm 1.5\degree$C. It works in three steps. When no cooling then works at 10Hz, less need of cooling then it works at 30Hz consuming 12.5kW when one compressor works and during peak load it works at 50Hz with electrical consumption of 18.5kW (nominal capacity) with two compressors work in parallel. So the heat gain is 10.9kW ($0.97 \times 0.90 \times 12.5$) and 16 kW ($0.97 \times 0.90 \times 18.5$) when one and two work compressors work considering 3% losses from frequency converter (VACON) and 10% from electric motor mentioned in manual.

#### 4.3.3 Ice resurfacing

Ice resurfacing is flooding of water onto ice surface to restore the ice surface condition by melting and removing rough top ice layer and creating flat and smooth surface again. The volume and temperature of flood water is dependent on water quality, load and time to freeze flood water. The resurfacing water temperature is between 55-60°C (ASHRAE, 2010) normally but for Norrtälje outdoor bandy ice rink 20-25 °C temperature is used. The flood water volume ($V_f$) typically is 0.4 to 0.7m$^3$ for 30×60m rink (ASHRAE, 2010) so 1 m$^3$ for 60×100m for outdoor bandy ice rink surface is used. The heat load resulting from flood water application is calculated as,

$$Q_f = 1000V_f [4.2(t_f - 0) + 334 + 2.0(0 - t_i)]$$

(4.5)

Where

- $Q_f$ = Heat load per flood, kJ
- $V_f$ = Flood water volume, m$^3$
- $t_f$ = Flood water temperature, °C
- $t_i$ = Ice temperature, °C

The heat load due calculation due to ice resurfacing is done by taking 3 resurfacings on daily average basis for whole season due to lack of clear examination of it by rise in ice surface temperature and unavailability of heating details of warm water for resurfacing.
5 Measurements & Calculations

5.1 Refrigeration System

The refrigeration system for Outdoor bandy ice rink is premade container solution. It is located inside the machine room adjacent to Bandy Ice rink shown in the Figure 5.1. It is big ice surface of 6000m² so it is divided into two sections(FC1 & FC2) from refrigeration and control point of view. FC1 is west and FC2 is east section both having separate circuits each having twin compressors. The two compressors in each circuit are 132kW MYCOM open type compressors of L-series having 8 cylinders with nominal capacity of 300kW each. They are capacity controlled by regulating operation of suction valve using oil pressure-driven unloaded piston for actuation. The evaporator is direct expansion manufactured in Finland and condenser is VAHTERUS Plate & Shell Heat Exchanger heat exchanger with technical specifications by same manufacturer. There is one frequency controlled 18.5 kW brine pump and coolant pump with nominal power as 11kW. The desuperheater, condenser, evaporator, water and coolant pumps are shown in Figures 5-3 & 5-4.

![Figure 5-2: Outside view of Machine Room](image1)

![Figure 5-1: Inside Machine Room View](image2)

![Figure 5-4: a) Desuperheater b) Condenser (Plate & Shell Heat exchanger)](image3)

![Figure 5-3: a) Water pump b) Coolant pump c) Evaporator](image4)

The heat rejected by desuperheater and condenser is not being used and condensation temperature is floating temperature.
5.2 Calculation Method—Refrigeration Performance Analyzer

5.2.1 ClimaCheck Method

ClimaCheck (CC) Sweden AB is the company which has developed complete product with hardware and software for troubleshooting and optimizing of refrigeration system, heat pumps and air conditioning (ClimaCheck Sweden AB).

ClimaCheck Performance Analyzer optimizes heat pumps and refrigeration, freezing and air conditioning equipment with increase in efficiency with decrease in wear which results in lowered costs and reduced carbon dioxide emissions (ClimaCheck Sweden AB).

ClimaCheck-method is an “internal method” used for analyzing performance of ice rink refrigeration system. The internal method used is for performance analyses of refrigeration, air-conditioning and heat pump applications (Berglöf, 2010). In it the compressor is used as mass flow meter and no need to install an external mass flow meter. With help of energy balance over compressor and measurements of pressure and temperature before and after the compressor, the refrigerant mass flow rate can be calculated by given equation below (Berglöf, 2005),

\[
\dot{m} = \frac{\eta_{el} P_{el} - Q_{out}}{h_{comp.out} - h_{comp.in}} \tag{5.1}
\]

Where

*\dot{m}*: Refrigerant mass flow rate  
*\eta_{el}*: Electric motor efficiency  
*P_{el}*: Electric power to the compressor motors  
*Q_{out}*: Heat loss from compressor body and/ or compressor cooling by oil/water  
*h_{comp.out}*: Enthalpy after compressor  
*h_{comp.in}*: Enthalpy before compressor

For accurate calculation of mass flow rate, the electrical motor efficiency and heat rejection from compressor body are two very crucial parameters. The type of motor, age and load on motor affect the electrical efficiency of it. With amount of compressor cooling by oil and water through compressor manufacturer software (MYCOM), heat losses of 7% from compressor body (Berglöf, 2005), mass flow, refrigerant state of entering expansion valve, etc. the cooling capacity and COP of system is calculated.

In addition to above mentioned parameters, temperatures of fluids heated by de super heater and condenser and brine in and out temperature are measured. To monitor heat loads on ice sheet, ice temperature for two halves of bandy, weather parameters, other necessary parameters for calculating refrigeration capacity are extracted through ClimaCheck online. The Figure 5-5 shows flowchart of measurement configuration for one half of Norrtälje bandy ice rink in which cooling capacity is 632kW with COP_{cool}= 3.3 and two parallel compressors are working with total power consumption of 190kW for 18th Nov, 2011 (warm day in season) at 05:30 pm. The temperature and pressures at each and every critical point of refrigeration system are shown in flowchart. The clear and magnified picture is attached at end as Appendix 11.7.
5.2.2 Method modification

To calculate accurate cooling capacity, mass of refrigerant should be calculated accurately. For actual figures oil and jacket heat rejected are extracted from MYCOM compressors design software and then relative heat rejected as percentage of absorber power at relevant condenser temperature is found compressor normal speed of 1400 RPM shown in Figure 5-6. So heat rejected by oil and jacket is found every minute at relevant condensation temperature with help of equation developed by figure shown. The absorbed power is calculated with consideration of 10% losses in motor and no losses through mechanical power transmission. The heat losses to ambient are taken as 7% along with relevant heat rejected by oil and jacket as percentage of absorbed power to calculate correct mass of refrigerant. The relative heat rejection by oil and jacket (y-axis) vs. condensation temperature(x-axis) at averaged sub cooling and superheating and different speeds is shown in Figure 5-6 below and magnified view is attached as Appendix 11.8.

Relative Heat Rejection Vs RPM & tcond
tevap=13°C, superheat=7K & subcooling=5K

Figure 5-6 Relative Heat Rejection Calculation by Compressor
5.3 Measurements

ClimaCheck (CC) system measures weather data and input and output parameters at critical points of refrigeration system and ice sheet condition every minute. The view of sample downloaded file from ClimaCheck online after running it in CC advanced software for extracting these parameters is attached as Appendix 11.9.

The Figure 5-7 shows temperatures of brine in and out of evaporator, evaporation and corrected ice surface temperature for väst (west) section of outdoor bandy for 1st November, 2010. To monitor the stability of system these temperature profiles help a lot. (Karampour, 2011) The profiles show that evaporation temperature remains stable at approximately at -13.8ºC when one compressor works and at -16ºC when two compressors work whereas the peak due to rise in evaporation temperature is showing that no compressor is in operation. The other thing that it shows brine pump governing no of compressors working on temperature of brine in to evaporator with set point of -7 ± 0.5 ºC here. This brine pump works in 3 steps at 10hz when no need of cooling, 30Hz when one compressor works (from midnight till 10am in figure) and fulfils need of cooling and at 50Hz when 3 ºC when two work in parallel at peak load time(10am to 5pm) and then again at 30Hz one compressor works from 5 to 12pm in night.

![Figure 5-7 Temperature measurements: November 1st, 2010](image)

The pressure of refrigerant before and after compressor is shown in Figure 5-8 for 1st November, 2010. It is shown in figure by 4 hours once after midnight and other 3 after 4pm in evening when no need of cooling and compressors are out of operation so evaporation temperature is increased and condensation temperature is reduced during these hours. So $t_{\text{evap}}$ and $t_{\text{cond}}$ remains at -13°C and 22°C when one compressor runs and go to -16°C and 30°C when another is kicked in during peak load period.

![Figure 5-8 Condensation & Evaporation Temperatures](image)
Similar trends are shown by refrigerant pressures on low and high side shown by Figure 5-9.

Figure 5-9 Pressure Measurements: November 1st, 2010
6 Experimental Results

6.1 Bandy Ice Rink Energy System

The ice rink energy system is discussed in chapter 2 already. The energy consumed by refrigeration and lighting in season 2010-2011 is presented in Table 6-1. The dehumidification and ventilation systems are not needed for outdoor ice rink and heating system is not considered in my calculations due to unavailability of energy consumption figures and heating details. The percentage energy share is shown by pie-chart in Figure 6-1 below.

Table 6-1 Outdoor Bandy Ice rink Energy Figures

<table>
<thead>
<tr>
<th>Systems</th>
<th>Energy Consumption(kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigeration</td>
<td>272471</td>
</tr>
<tr>
<td>Lighting</td>
<td>20160</td>
</tr>
<tr>
<td>Total</td>
<td>292631</td>
</tr>
</tbody>
</table>

![Figure 6-1 Energy System: Outdoor Bandy Ice rink](image)

6.1.1 Lighting

For Norrtälje bandy ice rink we have requirement of lighting for 4 hours in evening from 3-7pm. It works on half capacity with consumption of 40kW from 3-5pm and on full capacity with consumption of 80kW from 5-7pm. So in total then it accounts for 7% of total energy consumed which is verified by pie chart attached for energy systems shares based on my calculations for Season 2010-2011.

6.1.2 Refrigeration System

The energy consumed by refrigeration system for season by Norrtälje bandy ice rink is found out by CC instrumentation. The total energy figures were taken by counter and for compressors taken by EP Pro energy meter. The electrical energy consumed by brine, coolant pumps and dry cooler fans is found by subtracting energy consumed by compressors from total energy consumed by refrigeration system. The energy figures are shown in Table 6-2.
Table 6-2 Energy Consumption: Refrigeration system

<table>
<thead>
<tr>
<th>Load</th>
<th>Energy Consumption(kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>228876</td>
</tr>
<tr>
<td>Brine &amp; Coolant Pumps, Dry Cooler fans</td>
<td>43595</td>
</tr>
<tr>
<td>Total</td>
<td>272471</td>
</tr>
</tbody>
</table>

So on basis of energy figures for season 2010-11 by refrigeration system, Norrtälje bandy ice rink it is deduced the compressors consume 84% of total energy (electrical energy) and rest 16% is consumed by brine, coolant pumps and dry cooler fans shown by pie-chart in Figure 6-2.

6.2 Cooling Capacity & Compressor Power

The Figure 6-3 shows compressor power and cooling capacity by evaporator considering brine pump heat gain as loss to produce cooling for 31st October, 2010(Sunday). The averaged outdoor temperature is 7.27°C (warm day in season) with relative humidity(RH) of 82%. The Figure 6.3 shows that from midnight till 2am one compressor is running and after that another kicked in for two hours till 4am, after 5am both compressors start running in parallel and evaporator is producing cooling till 20:00 pm in night due to different activities like leisure skating, matches between 1 or 2pm and trainings after that till midnight cooling load is reduced where one compressor is enough to produce required cooling through evaporator.

Figure 6-2 Energy Share: Refrigeration system

Figure 6-3 Cooling Capacity & Compressor Power trend
6.3 Heat Loads

The heat load variation by different source categories on ice surface though out the day on 1st November, 2010 is shown in Figure 6.4. The averaged whole day values of outdoor temperature, wind speed and RH are 5.4°C, 0.7 m/s and 81%. The convection and condensation heat loads are maximum from 02:00 to 07:00 hrs and then from 14:00 to 19:00 hrs due to high wind speed, relative humidity and air temperature and low ice temperature during these hours. The radiation heat load is maximum from 10:00 to 14:00 hrs because of maximum global radiation received during these 4 hours. The gain due to ground heat and header heat is on average 3% of total heat load which is verified by Figure 6-4.

![Figure 6-4 Heat Loads: 1st Nov, 2010](image)

6.4 Total Heat Load Energy Share: Warm Day as an Example

The total heat load energy share from all source categories for whole day for Norrtälje bandy ice rink is shown in Figure 6-4 for warm day (1st November, 2010) in season. On this day averaged outdoor temperature is 5.4 °C, ice surface temperature is -6.23 °C and average wind speed of 0.7 m/s which contribute to most of convective and then diffusion heat transfer also due to averaged RH of 81%. The resurfacing heat transfer is due to 3 ice resurfacings on average taken for whole day. The heat gain due to ground heat and header is due to 3 percent of total heat load and brine pump heat taken into account when cooling is produced by evaporator. The heat load energy figures added on ice surface by different load categories including heat gain due to brine pump on heat load side are shown in Table 6-3 and percentage is shown in pie chart in Figure 6-5.

<table>
<thead>
<tr>
<th>Heat Load Source</th>
<th>Energy (kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Convection</td>
<td>10403</td>
</tr>
<tr>
<td>Condensation</td>
<td>5706</td>
</tr>
<tr>
<td>Radiation</td>
<td>2532</td>
</tr>
<tr>
<td>Ground &amp; Header heat</td>
<td>530</td>
</tr>
<tr>
<td>Resurfacing</td>
<td>754</td>
</tr>
<tr>
<td>Brine Pump</td>
<td>569</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>20495</strong></td>
</tr>
</tbody>
</table>

Table 6-3 Heat Load Energy Figures: 1st November, 2010
6.5 Cooling Capacity Vs Heat Load

The Figure 6-6 shows total cooling capacity eliminating brine pump heat gain added as loss to produced cooling of bandy in comparison to total heat load for whole day of 31st November, 2010. It is shown that from midnight till 09:00 hour convection and condensation are the driving factors with ground conduction and header heat gain of total 3 percent of total heat load. After 09:00 hour heat load rises sharply again due to radiation along with convection and condensation heat transfer on ice surface and remain more than cooling produced till 15:00 hour and then decreased for 3 hours as wind speed and RH reduces convective loads on ice due to radiation. After 19:00 hour wind speed and outdoor temperature increase again increase total heat load than the cooling produced.
For same warm day total cooling capacity and total heat load energy for 24 hours is shown by bars in Figure 6-7 and deviation of total cooling energy produced to heat load energy transferred to ice sheet is 11.8% shown in Table 6-4. This deviation is found due to factors like overestimation of heat losses to ambient air by compressor’s body for calculating mass of refrigerant, neglecting negative radiation (ice to sky) during night times with clear sky and lack of proper evaluation of ice resurfacings and heat load due to it during 24 hours.

<table>
<thead>
<tr>
<th>Date</th>
<th>Wind Speed V(m/s) Avg</th>
<th>Global Radiation G(W/m²) Avg</th>
<th>Relative Humidity RH (%) Avg</th>
<th>Outdoor Temperature (°C) Avg</th>
<th>Cooling Energy Qcool(kWh)</th>
<th>Heat load Energy Qload(kWh)</th>
<th>Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>31st Oct, 2010</td>
<td>0.84</td>
<td>12.24</td>
<td>81.79</td>
<td>7.26</td>
<td>20197</td>
<td>22899</td>
<td>12</td>
</tr>
<tr>
<td>1st Nov, 2010</td>
<td>0.70</td>
<td>17.58</td>
<td>80.81</td>
<td>5.43</td>
<td>16084</td>
<td>19747</td>
<td>18</td>
</tr>
<tr>
<td>17th Oct, 2010</td>
<td>0.69</td>
<td>22.68</td>
<td>76.50</td>
<td>7.80</td>
<td>17937</td>
<td>22946</td>
<td>22</td>
</tr>
<tr>
<td>20th Oct, 2010</td>
<td>0.74</td>
<td>20.21</td>
<td>76.54</td>
<td>8.83</td>
<td>17760</td>
<td>23982</td>
<td>26</td>
</tr>
</tbody>
</table>
6.7 Season Calculations

The calculations for season 2010-2011 are done by taking inputs (weather parameters and measurements for heat load and refrigeration system capacity) on daily average basis. The heat loads from different source categories for whole season (136 days) are calculated in same fashion as of four warm day’s calculations method and heat transfer load due to ice resurfacing taken 3 on daily average basis are added to total heat load of day. The comparison of total cooling energy produced for whole season is done with total heat load energy for filtered warm days (cooling produced days: 48 out of 136 days when (Pcomp>20kW)

The heat loads from different source categories except from ice resurfacing is shown in Appendix I. The final total heat load energy figures for season 2010-2011 are shown in Table 6-6 and the percentages are shown in Figure 6-7 as pie chart.

Table 6-6 Total Heat Load Energy Figures: Season 2010-2011

<table>
<thead>
<tr>
<th>Heat Load Source</th>
<th>Energy(kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Convection</td>
<td>671922</td>
</tr>
<tr>
<td>Condensation</td>
<td>217826</td>
</tr>
<tr>
<td>Radiation</td>
<td>213574</td>
</tr>
<tr>
<td>Ground &amp; Header heat</td>
<td>33100</td>
</tr>
<tr>
<td>Resurfacing</td>
<td>42054</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>1178475</strong></td>
</tr>
</tbody>
</table>

The final figures of electrical energy consumed by compressors and in total and cooling energy produced by refrigeration systems of two section of bandy for whole 2010-2011 season are shown in Table 6.7. The energy consumed by compressors (Ecomp) is found 84% and by auxiliary equipment like brine and water pumps and dry cooler fans is 16% of total energy consumed (Etotal) by refrigeration system. The COPcomp is assumed as 4.5 at tevap -13°C and tcond 23°C by MYCOM software so COPtotal of whole refrigeration system for each section of bandy is found as 3.78 which is used for calculating cooling energy produced by machine shown in Table 6-8.
Table 6-7 Bandy Ice rink Cooling Capacity: Season 2020-2011

<table>
<thead>
<tr>
<th>Season length (days)</th>
<th>FC2</th>
<th></th>
<th></th>
<th>FC1</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>E_{comp} (kWh)</td>
<td>E_{tot} (kWh)</td>
<td>Total Cooling Energy (kWh)</td>
<td>E_{comp} (kWh)</td>
<td>E_{tot} (kWh)</td>
<td>Total Cooling Energy (kWh)</td>
</tr>
<tr>
<td>136</td>
<td>95418</td>
<td>113593</td>
<td>360680</td>
<td>133458</td>
<td>158878</td>
<td>504470</td>
</tr>
</tbody>
</table>

So during the evaluation season total cooling energy is found as 865150 kWh and corresponding heat load energy considering surface ice sheet conditions is calculated to 1136421 kWh as shown in Table 6-8. It comes up with deviation of 27% which ensures measure of modelling accuracy to some extent for such a large ice surface.

Table 6-8 Season Final Energy Figures

<table>
<thead>
<tr>
<th>Season</th>
<th>E_{total}(kWh)</th>
<th>COP_{total}</th>
<th>Total Cooling Energy Q_{cool}(kWh)</th>
<th>Total Heat load Energy Q_{load}(kWh)</th>
<th>Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2010-2011</td>
<td>228876</td>
<td>3.78</td>
<td>865150</td>
<td>1136421</td>
<td>27</td>
</tr>
</tbody>
</table>
7 Model Development

7.1 Considerations

- Norrtälje Outdoor bandy data is used as outdoor ice rink data input.
  - Specific Energy for outdoor ice rink is calculated by taking refrigeration and lighting into consideration as heating for ice resurfacing and locker rooms and showers is not covered
  - Length of Season is 4.5 months which is actual length of season 2010-11

- Indoor ice rink calculations are not being done so
  - Etotal is taken by Stoppsladd Report 2011 (Stoppsladd, 2011)
  - Ecomp is taken as 43% by Stoppsladd Report 2010 (Stoppsladd, 2010) of Etotal
  - Air is calculated through Etotal and Specific Energy taken by 9 Indoor Bandy fields (Stoppsladd, 2011)

7.2 Salient Features: Developed Model

- Field Weather parameters measurements can be compared with SMHI model measurements
- Prediction of energy usage by different systems and in total by indoor and outdoor ice rinks
- Decision tool to choose between alternatives on basis of predicted energy usage

7.3 Model Overview

Model overview is shown in Figure 7-1

<table>
<thead>
<tr>
<th>Climate Data</th>
<th>Indoor Ice rink Data</th>
<th>Outdoor Ice rink data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input</td>
<td>SMHI</td>
<td>FIELD</td>
</tr>
<tr>
<td>Wind (m/s)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Relative Humidity (%)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solar Radiation (W/m²)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Input</td>
<td>Start date</td>
<td>End date</td>
</tr>
<tr>
<td>length of season (days)</td>
<td>191</td>
<td>7000</td>
</tr>
<tr>
<td>COP_comp</td>
<td>-4</td>
<td>1.8</td>
</tr>
<tr>
<td>Specific Energy (kWh/day·m²)</td>
<td>1.8</td>
<td>0.35</td>
</tr>
<tr>
<td>Output</td>
<td>Refrigeration</td>
<td>COPtot</td>
</tr>
<tr>
<td>COPtot</td>
<td>3.78</td>
<td>E_comp(kWh)</td>
</tr>
<tr>
<td>E_comp(kWh)</td>
<td>1634838</td>
<td>E_comp(kWh)</td>
</tr>
<tr>
<td>E_tot(kWh)</td>
<td>43595</td>
<td>E_tot(kWh)</td>
</tr>
<tr>
<td>Lightning</td>
<td>272471</td>
<td>Lightning</td>
</tr>
<tr>
<td>Total Energy(kWh)</td>
<td>2406600</td>
<td>Total Energy(kWh)</td>
</tr>
</tbody>
</table>

Figure 7-1 Developed Model View
8 Discussion

The on and off and unstable operation of compressors when no/less and more cooling produced can be seen in plots for input power to compressor ($P_{comp}$) and calculated cooling capacity ($Q_{cool, \text{calc}}$) in Appendices 11.1, 11.2, 11.3 and 11.4 for sections (FC1&FC2). Due to these on and off operation of compressors and unstable operation on 17th October, 1st November, 2010 and 20th October, 2010 less cooling produced compared to almost constant more heat loads till 3pm and little more cooling after 3pm in comparison to decreased heat loads for some hours in evening. These on and off operations of compressors affect the calculated average cooling capacity on hourly basis seriously in my excel model and reduces it considerably by putting zero when no cooling produced and compressors are out of operation. So calculation of average cooling capacity on hourly average basis is drawback in the developed excel model for these unstable operations of compressors.

The heat loss by compressor body to ambient is taken as 7% (Berglöf, 2005) of absorbed power along with 16% on average by heat rejection from oil and jacket and in total 25% of absorbed power is a total heat loss by compressor. The 7% losses taken to ambient by body are too high for open type compressors as compared to hermetic and semi hermetic compressor as it is cooled by oil and jacket by water. So overestimation of 5-7% heat losses by compressor body is considered for calculation of total heat losses and then mass of refrigerant for cooling capacity. Due to it cooling capacity calculated for 4 warm days in the calculations is somewhat less then produced but for season calculations it is correct which is calculated through actual energy consumption by compressors ($E_{comp}$) and assumed COP$_{comp}$ on basis of data by MYCOM software and cooling produced days consideration.

The ice resurfacing heat load is calculated by taking 3 resurfacings daily on average. The actual heat load calculations due to ice resurfacings is not being done due to lack of its examination through ice surface temperature and heating details of warm water for resurfacing.

The negative radiation (ice to sky) is not measured and evaluated by CC instrumentation. During nights with clear sky having no clouds when ambient temperature is too low we have radiations from ice to sky when ice surface is warmer than ambient air. So actual measurements of these radiations will then reduce solar radiation load on ice sheet for 24 hours evaluation as well as total radiation heat load for whole season.

The deviation in calculated total heat load energy and measured total cooling energy produced is 19% on average for 4 warm days which is due to above all discussed reasons. The final deviation of 27% between total cooling energy and heat load energy for whole season 2010-2011 is clue for validity of model to some extent which depends on many factors. The reasons figured out are overestimation of heat losses from compressors body, assumed daily ice resurfacings instead of actual evaluation with relevant actual heat load calculation and lack of measurement and evaluation of negative radiation.
9 Conclusions

The largest energy user of energy systems in most of the ice rinks is the refrigeration system. The first target was to reduce this energy which was possible through cooling load reduction or refrigeration system improvements. The accurate measurement of weather parameters was pre requisite for having reduced heat loads to the ice surface and was really significant in case of outdoor bandy as these affect the ice surface directly. The task of weather data validation for accurate heat load calculations is accomplished successfully with aid of SMHI climate model data, correlations and related web based geographical data.

The heat loads especially convection and condensation which account for almost 75% of total heat load on average during whole season were found really critical depending on correct wind speed, relative humidity and temperatures of air far and close to ice surface. The radiation from ice rink to sky during night time are not taken into account in measurements as well as calculations which has increased radiation loads to ice surface to some extent. The heat loads due to ground heat gain and header heat gain are taken as average on 3% of total heat load and due to ice resurfacing is found 4% by taking 3 resurfacings on average daily basis.

Another challenge was to monitor and analyze the performance of refrigeration system under these heat loads. The method adopted in study analyzed the refrigeration with performance analyzer system which established provided cooling capacity and COP etc. (Jörgen Rogstam, 2011). The conditions of ice sheet were also monitored to analyze the affect of heat loads at different times on various days of season.

The average cooling capacity considering brine pump heat gain is being compared with heat loads based on ice sheet conditions for warm days. The same comparison and analysis is being done for whole season neglecting loss of cooling due to brine pumps by calculating cooling capacity with actual energy consumed and practical assumed COP value of MYCOM compressor.

The predicted energy usage by developed model refrigeration system considering local geographical weather conditions and required technical specifications of ice rink supports the decision to build an indoor or outdoor ice rink in future.
10 Bibliography


Information. *International Ice Hockey Federation (IIHF)*. 3.


SMHI. FlYg. avdDaqfa. [Online] [Cited: June 12, 2010.] https://www.supersavertravel.se/ditt-kvitto.


11 Appendices

11.1 Results: 31st October, 2010

- **Q_{heatload}:** convection, radiation, condensation, ground & header heat gain
- **Q_{total cooling capacity-brine pump heat gain eliminated}:**

**FC1**

**FC2**
11.2 Results: 1st November, 2010
11.3 Results: 17th October, 2010

- $Q_{\text{total cooling capacity}}$
- $Q_{\text{heatload\-convection, radiation, condensation, ground & header heat gain}}$

**FC1**

**FC2**
11.4 Results: 20TH October, 2010
11.5 Power Loss in Frequency Converter: Brine Pump

The frequency converter for brine pump is VACON NXS-22kW. The power loss for it is found from curve (power loss as function of switching frequency) of NX-50038...0061 having same trend with 3.6 kHz as default switching frequency value for it. So at 3.6 kHz switching frequency, power losses are found as 700kW which is 3% of 22kW which can be verified by red dotted lines plotted on third line (0045NX5400V) from bottom in Figure 11-1 attached here.

![Figure 11-1 Power loss as function of switching frequency: NX-50038...0061](image.png)
11.6 Compressor Heat Rejection

MYCOM software has been used for heat rejection from compressor. As an sample for heat rejection calculation from software the Figure 11-1 attached below shows view of interface when \( t_{\text{evap}} = -13 \, ^\circ\text{C} \) and \( t_{\text{cond}} = 25 \, ^\circ\text{C} \), superheat = 7 \, ^\circ\text{C} \) and subcooling = 5 \, ^\circ\text{C} \). The compressors under consideration for Norrtälje Outdoor bandy refrigeration machine are L-series with 8 cylinders.

![MYCOM software interface view](image)

Figure 11-2 MYCOM software interface view
11.7 Relative Heat Rejection Study: MYCOM

The heat rejection from jacket and oil are studied at various condensation temperatures RPMs is shown as discussed earlier in section 5.2.3 Method Modification. Table 11-1 shows results of relative heat rejection at specific RPM and condensation temperature.

Table 11-1 Relative Heat Rejection finding results: MYCOM software

<table>
<thead>
<tr>
<th>tcond (⁰C)</th>
<th>RPM=1000</th>
<th>RPM=1100</th>
<th>RPM=1200</th>
<th>RPM=1300</th>
<th>RPM=1400</th>
<th>RPM=1500</th>
<th>RPM=1600</th>
<th>RPM=1700</th>
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<td>10</td>
<td>264.5</td>
<td>35.8</td>
<td>0.7</td>
<td>5</td>
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<td>15.92</td>
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<td>15</td>
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<td>20</td>
<td>240.9</td>
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<td>11.3</td>
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<td>69.8</td>
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</tbody>
</table>

Relative Heat Rejection study: MYCOM

The relative heat rejection from jacket and oil are studied at various condensation temperatures RPMs is shown as discussed earlier in section 5.2.3 Method Modification. Table 11-1 shows results of relative heat rejection at specific RPM and condensation temperature.
11.8 ClimaCheck Measurement Flow Chart

Sub Cool = 4.7 K

Condensing = 32.3 °C

Power tot = 190.3 kW

Evaporation = -14.4 °C

Super Heat = 3.9 K

Ambient temp = 9.4 °C

Sun rad = 2.0 W/m²

Ice temp V = -5.5 °C

Ice Temp Ō = -5.4 °C

COP Cool = 3.3

Cap Cool = 632.7 kW

Cap Heat = 736.9 kW

COP Heat = 4.1
11.9 Relative Heat Rejection Calculation

Relative Heat Rejection Vs RPM & tcond
teavap=-13°C, superheat=7K & subcooling=5K

\[ y = 0.2846x + 8.9374 \]

- RPM=1000
- RPM=1100
- RPM=1200
- RPM=1300
- RPM=1400
- RPM=1500
- RPM=1600
- RPM=1700

Linear (RPM=1400)
### Energy Usage Prediction Model Comparing Indoor Vs. Outdoor Ice Rinks

#### 11.10 Measurements: ClimaCheck Advanced Software

| Test Condition | AMMONIA | ClimaCheck Prestanda Analysis | Term. eff. Compl. | El. eff. | Stab COP | Stab Accept.
<table>
<thead>
<tr>
<th></th>
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<tbody>
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<td>0.0</td>
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<td>75</td>
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#### Table

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#### Conditions

- **Exposure:** 7-9
- **Press. Ref.:** 31
- **Cond. Sec.:** 31
- **High Pressure Ref.:** 31
- **Compressor:** 31
- **Oilcooler type 3 = Secflow:** 31
- **Oilcooler type 4 = dtiv. data:** 31

#### Values

- **Energy:** 7-9
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- **Time:** 7-9
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- **Temp.:** 7-9
- **Vol.:** 7-9
- **Perf.:** 7-9
- **Eff.:** 7-9
11.11 Ice Rink Energy System
11.12 Heat Loads: Season 2010-2011