APPENDIX 5A. ENERGY MODELING

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APPENDIX 5A. ENERGY MODELING

5A.1 INTRODUCTION

This appendix describes the FREEZE model that the U.S. Department of Energy (DOE) used to conduct energy modeling for the engineering analysis for automatic commercial ice makers. FREEZE simulates the performance of both batch ice makers (*e.g.*, cubers) and continuous ice makers (*e.g.,* flakers and nugget type machines). FREEZE is a transient model that can calculate changes in operating conditions as a function of time. While this feature is not needed for continuous ice makers, it is important for batch machines in which refrigerant temperatures and pressures change significantly during the freeze cycle due to ice buildup on the evaporator.

5A.1.1 FREEZE Model Description

The original version of FREEZE was developed by Anthony Varone in 1995[1] to operate in an MSDOS environment. It has since been upgraded to operate in a Windows environment. The model can simulate systems with a variety of configurations, which include:

- batch or continuous ice making,
- air-cooled or water-cooled condensers,
- batch type or continuous water makeup,
- internal or external water pump,
- existence of a liquid-line/ suction-line heat exchanger,
- choice of compressor, and
- choice of common refrigerants.

FREEZE is a system-level model that calculates the complex interactions among individual components. With the exception of air-cooled condensers, component performance characteristics are specified by the user. An advantage to this approach is that the model can simulate machines of many different configurations as long as component performance data is consistent with the input parameters in the FREEZE model.

For batch type machines, output includes:

- overall ice production rate, energy consumption, and freeze cycle time
- evaporator, sump water, and refrigeration system temperatures
- refrigeration system pressures, power input, and ice growth versus time during the freeze cycle

The FREEZE process is modeled using energy balances on the evaporator mass, ice mass, sump water, and water system. The evaporator plate, ice mass, sump water, and water system are each treated as lumped bodies, each at its own uniform temperature to calculate transient heat transfer between components. The energy balances account for heat transfer into and out of each component, energy storage, latent heat conversion associated with freezing, and enthalpy fluxes associated with refrigerant flow and/or makeup water flow in and out of each component. External heat transfer into the system includes ambient heat leak and the electrical energy associated with the water pump.

A major challenge to modeling batch type machines is calculating the overall evaporator thermal conductance, which varies throughout the cycle due to ice buildup on the plate and decreasing refrigerant mass flow rate with time. This is accomplished using user-defined curves for heat transfer versus ice mass on the plate that account for conduction heat transfer from the water, through the ice, and through the evaporator plate. Refrigerant heat transfer coefficients are calculated within the program and are used along with the evaporator thermal conductance to calculate the overall thermal conductance from the water to the refrigerant. Refrigerant evaporator heat transfer coefficients and pressure drop are calculated within the program using models originally developed by Varone.[2]

The evaporator thermal conductance curves are generated by a separate conduction heat transfer analysis and depend on the specific evaporator design. The curves are described as sixth-order, best-fit polynomials for evaporator thermal conductance as a function of ice mass on the plate. The evaporator UA curves were developed using auxiliary spreadsheets for the two most common evaporator types, as discussed the following section.

The refrigerant cycle model is a descendent of the NIST CYCLE7[3] model, modified to incorporate the transient effects associated with ice buildup on the evaporator plate. There is a choice of six refrigerants, including R-404A, the most common refrigerant used in ice making applications. Refrigerant thermodynamic and thermophysical properties are calculated using Refprop 8.0.[4] Refrigerant thermal conductivity and viscosity are calculated using curve fits developed from refrigerant manufacturer literature.

Continuous ice makers operate at steady-state; thus, conditions do not change with time. Program output includes ice production rate, energy consumption, and refrigerant cycle conditions. As with the modeling of batch type machines, evaporator thermal performance is a user-defined parameter and is not calculated within the program. Additional inputs include ice quality, auger motor power and efficiency, and heat leak from the ambient air into the evaporator.

Compressor performance is determined from user-created data files based on compressor maps provided by compressor manufacturers. Compressor capacity and power are adjusted at each point in the calculation to reflect suction conditions different from rating conditions.

Machines with either air-cooled or water-cooled condensers can be modeled. For air-cooled condensers, refrigerant and air-side heat transfer coefficients and pressure drop are calculated within the program using models originally developed by Varone2 and enhanced to incorporate more recent correlations. Required heat exchanger input data include tube diameter and wall thickness, number of tubes, tube spacing, fin type, spacing, and material, airflow rate and inlet temperature, fan power, and refrigerant subcooling.

Water-cooled condenser operation is based on maintaining a constant condensing pressure and therefore temperature, and is thus modeled based on maintaining a constant condensing bubble-point temperature as defined by the user. User inputs include condensing temperature, refrigerant subcooling, and pressure drop.

The expansion device (*e.g.*, thermal expansion valve) is not modeled explicitly. Instead, it is simulated in the model as maintaining a constant evaporator superheat throughout the freeze cycle as specified by the user.

The program can model systems with a liquid-line/suction-line heat exchanger and accounts for suction line heat gain from the ambient air and refrigerant pressure drop. User inputs include heat exchanger effectiveness, suction line heat transfer, and pressure drop.

Each freeze cycle requires a flow of water necessary to flush impurities from the system and to provide water to make up for the water harvested as ice. Generally, one of two strategies for providing makeup water is used. The first, referred to as a batch fill strategy, involves filling the sump with the water needed to produce a batch of ice. The second strategy, referred to as a continuous fill strategy, involves supplying a flow of makeup water corresponding to the rate at which ice is frozen on the evaporator plate. Both strategies can be modeled.

Evaporator cold compartment configurations vary widely and thus are not amenable to general analytical treatment. Some designs enclose the evaporator in a fully insulated cabinet with a plastic liner, while others form the evaporator cold compartment using cabinet sheet metal with Styrofoam insulation. Some designs have parts of walls or even entire walls that are uninsulated. Some place the sump entirely inside the cold compartment, while others have the sump straddling the cold compartment and the compressor compartment. Because of the wide variety of designs, the model does not calculate the heat leak into the cold compartment from basic cabinet design parameters. Instead, it is modeled using a user-defined evaporator cold compartment thermal conductance parameter.

The harvest process involves bypassing the condenser to allow hot refrigerant to flow directly from the compressor to the evaporator to provide the heat input to free the ice from the plate. A number of manufacturers also assist the harvest process by mechanical means or by using the heat contained in the makeup water. Thus, the harvest process has complexities similar to those involved with the freeze process. In FREEZE, the effects of harvest on performance are determined based on input data for the harvest time, ice meltage, and harvest energy consumption.

Parasitic power consumption of components such as control circuit boards, solenoid valves, and harvest assist devices are modeled by entering the power consumption of each component.

A summary of the FREEZE modeling approach by component is summarized in Table 5A.1.1 and in the following section.

Table 5A.1.1 Review of FREEZE Component Modeling

Component	Model Approach
Compressor	Performance determined using manufacturer's performance data adjusted for conditions different from rating conditions.
Condenser	Air-cooled: Air-side and refrigerant-side heat transfer coefficients and pressure drop are calculated based on condenser design and airflow rate. Water-cooled: Modeled to maintain a constant specified condensing temperature.
Expansion device	Modeled as constant specified evaporator superheat.
Evaporator	Batch type: Thermal conductance calculated using user-defined data for thermal conductance as a function of ice mass on the evaporator combined with refrigerant heat transfer coefficients calculated based on instantaneous refrigerant mass flow rate and properties. The cycle is defined by the preharvest batch weight as defined by the user. Continuous: Performance calculated based on user-defined thermal conductance, auger motor power, ice quality, and ambient heat leak.
Suction line heat exchanger	Performance calculated based on user-defined thermal effectiveness.
Suction line	Heat leak from ambient and refrigerant pressure drop specified by user.
Cold compartment	Heat transfer calculated based on user-defined thermal conductance parameter.
Harvest	Requires user to input data for harvest time, duration, and power consumption.
Auxiliary electrical loads	Power consumption specified by user.
Refrigerant properties	References 2 and 4, and curve fits of manufacturer data.
Heat transfer coefficients and pressure drop	References 5, 6, 7, 8, 9, and 10.

5A.1.2 Supplemental Evaporator Thermal Conductance Spreadsheets for Batch Type Machines

For batch type machines, the FREEZE model requires the user to input the evaporator thermal characteristics in the form of a sixth-order polynomial curve fit for evaporator thermal conductance as a function of ice mass on the plate. In addition to the six coefficients, the curve is defined in terms of the evaporator thermal conductance at the start and end of the freeze process. This approach allows FREEZE to simulate the performance of any evaporator design since evaporator design specifics are not considered in the FREEZE model.

The majority of batch type machines incorporate one of two evaporator designs: (1) the copper plate-grid; or (2) the stainless steel sheet-copper tube. Each of these is described in the following.

- The copper plate-grid, also called a "waffle" evaporator, consists of copper gridwork bonded to a copper plate. Copper refrigerant tubing is bonded to the back of the copper plate. The components are bonded by brazing.

The evaporators are mounted vertically. Water is introduced at the top of the evaporator and cascades over the copper grid. Ice builds up inside each cell formed by the grid until each cell is filled with ice. Ice also forms on the edges of the copper grid, forming a "bridge" between cubes. The bridge serves two purposes. First, in many ice makers harvest is initiated when the bridge thickness reaches a predetermined thickness. Second, the bridge creates a monolithic slab of ice on the plate that ensures all cubes fall off the plate simultaneously. Ideally, the bridge should break when the ice falls into the bin to create individual cubes. Most machines of this type also assist the harvest process by using a mechanical means to push the ice slab off the plate.

The copper gridwork determines cube size. The most common cube sizes are (a) 3/8 inch wide by 7/8 inch high by 7/8 inch deep; and (b) 7/8 inch wide by 7/8 inch high by 7/8 inch deep. The cube face is not usually flat, but often has concavities or "divots" due to relatively poor heat transfer through the cube compared to the copper grid.

- The stainless steel sheet-copper tube evaporator consists of pleated stainless steel sheets soldered to both sides of a flattened copper serpentine tube. The pleats are vertical, forming channels that direct water down the sheet. Ice forms as water freezes where the sheet is bonded to the copper tube. The stainless steel sheet is relatively thin, resulting in relatively poor heat transfer along the sheet and pleats. Most of the heat transfer is directed through the ice directly toward the contact patch between the sheet and tube.

  The ice is harvested using a combination of hot-gas bypass and incoming makeup water introduced between the two stainless steel sheets. The resulting cube has an approximately semicircular cross section and is 1-1/2 inches long by 1-1/8 inches wide by 1/2 inch thick.

Spreadsheets have been created to help the user develop the thermal conductance curves of each evaporator design described above. Normally, the thermal analysis involves calculating the ice growth as a result of the heat removed from the ice and water. The analysis used in the spreadsheets is actually the reverse of that process, in that the thermal conductance is calculated based on an assumed pattern of ice growth. Note that this analysis does not include the effects of the refrigerant heat transfer coefficient, which is calculated in the FREEZE model during the simulation.

For the waffle design, input parameters include the cell dimensions, number of cubes, copper grid thickness, copper tube dimensions, bond thickness, and material thermal properties. Cube input parameters include ice thermal properties, bridge thickness, and divot dimensions. The water heat transfer coefficient is also an input parameter.

For the pleated stainless steel sheet-copper serpentine design, input parameters include dimensions of the cube, contact patch, and copper tube; the water heat transfer coefficient; and material thermal properties.

The output of the spreadsheets consists of the evaporator thermal conductance at the start and end of the freeze process along with the six polynomial coefficients of best-fit polynomial curve of thermal conductance versus ice mass on the plate. An example of the results of a thermal conductance analysis for a waffle evaporator from a 250 lb/24 hr capacity machine is shown in Figure 5A.1.1.

The results shown in Figure 5A.1.1 demonstrate the effect that the ice has on the evaporator thermal conductance. At the start of the freeze process, the conductance is roughly 620 Btu/hr -F. The conductance decreases rapidly as ice builds up on the plate until it is reaches a value of about 100 Btu/hr -F.

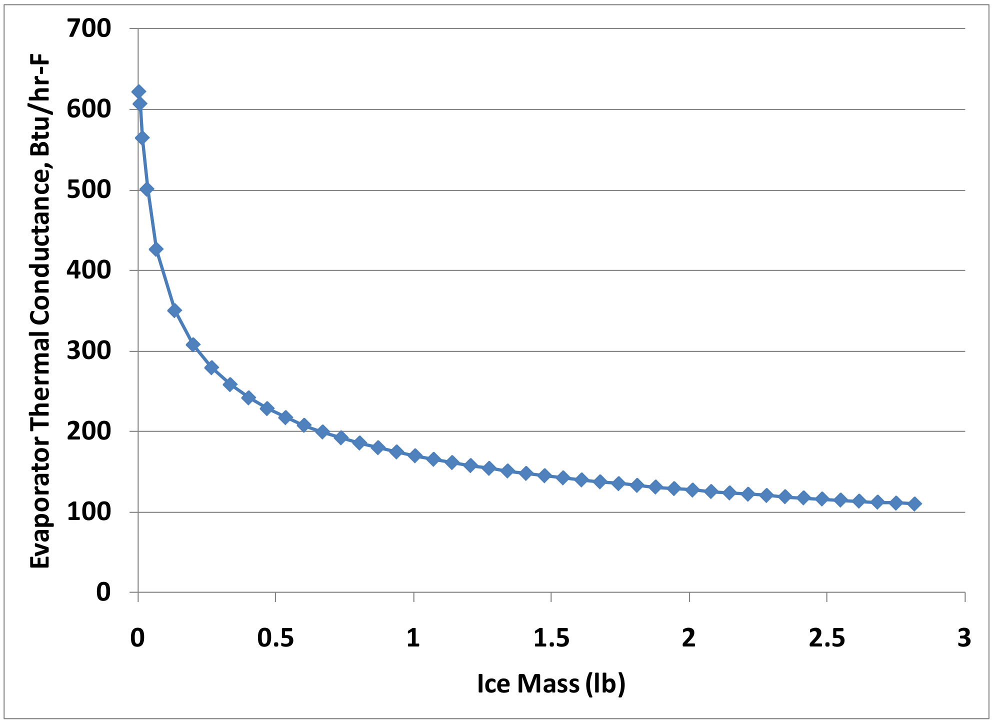

Figure 5A.1.1 Evaporator Thermal Conductance versus Ice Mass

The effect of the refrigerant heat transfer coefficient on the calculated overall thermal conductance as a function of time into freeze cycle is shown in Figure 5A.1.2. Also shown is the resulting effect on refrigerant evaporator temperature. At the start of the freeze cycle, the overall conductance is about 350 Btu/hr-F. The conductance declines throughout the cycle as a result of ice buildup on the plate, reaching a minimum of about 70 Btu/hr-F at the end of the freeze process.

The effect on the refrigerant evaporator temperature is dramatic. The refrigerant temperature is about 65 °F at the start of the freeze cycle then decreases continuously throughout the cycle until reaching a minimum value of about -2 °F at the end of the freeze process.

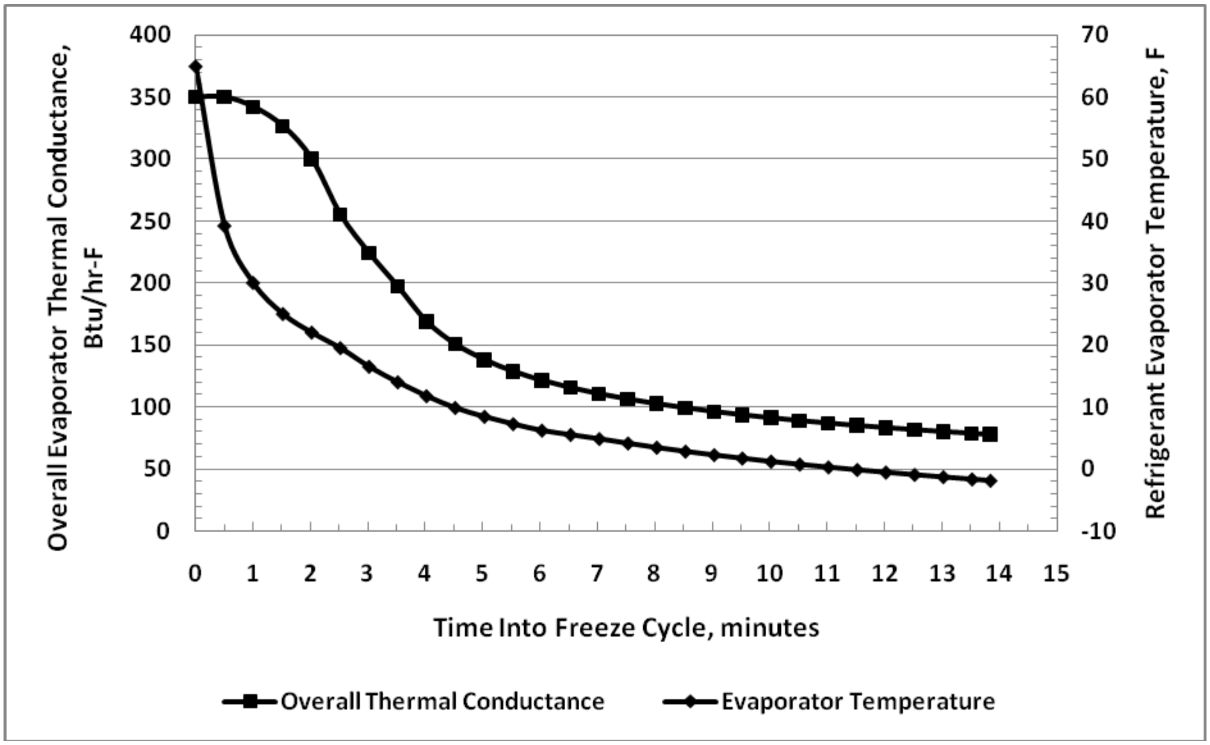

Figure 5A.1.2 Calculated Evaporator Overall Thermal Conductance and Refrigerant Temperature versus Time into Freeze Cycle

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