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Simulation of an Automatic Commercial Ice Maker

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ABSTRACT

Automatic commercial ice making machines that produce a batch of cube ice at regular intervals are known as "cubers". Such machines are commonly used in food service, food preservation, hotel, and health service industries. The machines are typically rated for the weight of ice produced over a 24 hour period at ambient air temperatures of 90 °F and water inlet temperature of 70 °F. These cubers typically utilize an air-cooled, vapor-compression cycle to freeze circulating water flowing over an evaporator grid. Once a sufficient amount ice is formed, a valve switches to enable a harvest mode, where the compressor's discharge gas is routed into the evaporator, thereby releasing ice into a storage bin.

The U.S. Department of Energy has set a target of reducing energy usage by 10 - 15% by 2018. Engineering models are not publicly available to assist designers in achieving the new energy regulations. This paper presents an engineering simulation model that addresses this need. This model simulates the transient operation of a cuber ice machine based on fundamental principles and generalized correlations. The model calculates time-varying changes in the system properties and aggregates performance results as a function of machine capacity and environmental conditions. Rapid "what if" analyses can be readily completed, enabling engineers to quickly evaluate the impact of a variety of system design options, including the size of the air-cooled heat exchanger, finned surfaces, air / water flow rate, ambient air and inlet water temperature, compressor capacity and/or efficiency for freeze and harvest cycles, refrigerants, suction/liquid line heat exchanger and thermal expansion valve properties.

Simulation results from the model were compared with the experimental data of a fully instrumented, standard 500 lb capacity ice machine, operating under various ambient air and water inlet temperatures. Key aggregate measures of the ice machine's performance are: 1) cycle time (duration of freeze plus harvest cycles), 2) energy input per 100 lb of ice, and 3) energy usage during 24 hours. For these measures, the model's accuracy is within 5% for a variety of operating conditions.

1. INTRODUCTION

The U.S. Department of Energy (2015) has revised the energy efficiency standards for automatic commercial ice makers (ACIM) that produce 50 lbs to 4000 lbs per day. A major segment of the ACIM market are self-contained units that produce a batch of cube ice at regular intervals. These machines, known as "cubers", are primarily used in restaurants, hotels, convenience stores, and hospitals. Cube weight typically range from approximately 1/6 to 1/2 oz and each manufacturer usually produces a unique shape (cubic, rectangular, crescent, and pillow) to distinguish themselves from other manufacturers (Westphalen et al., 1996). To assist in the design of ACIM systems, Varone (1995) developed an empirically based simulation model. Since the design changes necessary to meet the new standards will likely exceed the bounds of an empirical model, a physics-based ACIM simulation model is desired to assess the performance implications.

Physics-based simulation models for the steady-state operation of a vapor-compression, refrigeration systems have been established several decades ago (Domanski and Didion, 1983, Fisher and Rice, 1983). These models continue to serve as the basis for more recent enhancements such as alternative refrigerants (Arora and Kashik, 2008) and complex system circuits (Bahel and Shivashankar, 2014). The steady-state models are sufficient for most refrigeration applications, where the system achieves a stable operating mode and continues to run in that mode for a majority of time.

Engineering models for the ice machine presents a particularly challenging application. The ice machine exhibits entirely transient behavior, as the operation continually cycles between the ice formation mode and ice harvest mode. Bendapudi et al. (2008) discuss various approaches for transient simulation models. Of particular interest for ACIM modeling include refrigeration systems during variable evaporator load (Chi and Didion, 1982, Macarthur, 1984), startup conditions (Li and Allleyne, 2010) and hot-gas bypass (Hoffenbecker et al., 2004) as used during ice harvest mode. Transient models for the heat exchange between refrigerant flowing through the evaporator tubing to water flowing over an ice forming grid do not exist in the literature and were developed by the authors.

Once established, the simulation model enables prediction of component conditions, loads under different operating environments, and assessment of system design changes. The remainder of the paper is organized as follows. The description of the ice machine components and operation is described in Section 2. Section 3 presents the model theory. An overview of the model is provided in Section 4. Section 5 presents the results and comparisons to an instrumented ice maker.

2. ICE MAKER DESCRIPTION

A schematic of a "cuber" is given in Fig. 1. This ACIM consists of two major subsystems: the vapor compression refrigeration system and water supply/circulation/purge system. The typical refrigeration system components include a compressor, air-cooled condenser, thermostatic expansion device, liquid line/suction line interchanger, and an evaporator that consists of copper tubing attached to copper or stainless steel grid forming the ice making surface. During harvest, a hot-gas solenoid valve switches and channels refrigerant directly from compressor to evaporator, which melts a boundary layer and releases the ice. The water system consists of a water sump, circulation pump, plastic tubing and an evaporator water distributor. A potable water supply connection supply control valve and purge drain controls water inflow and from the ice maker.

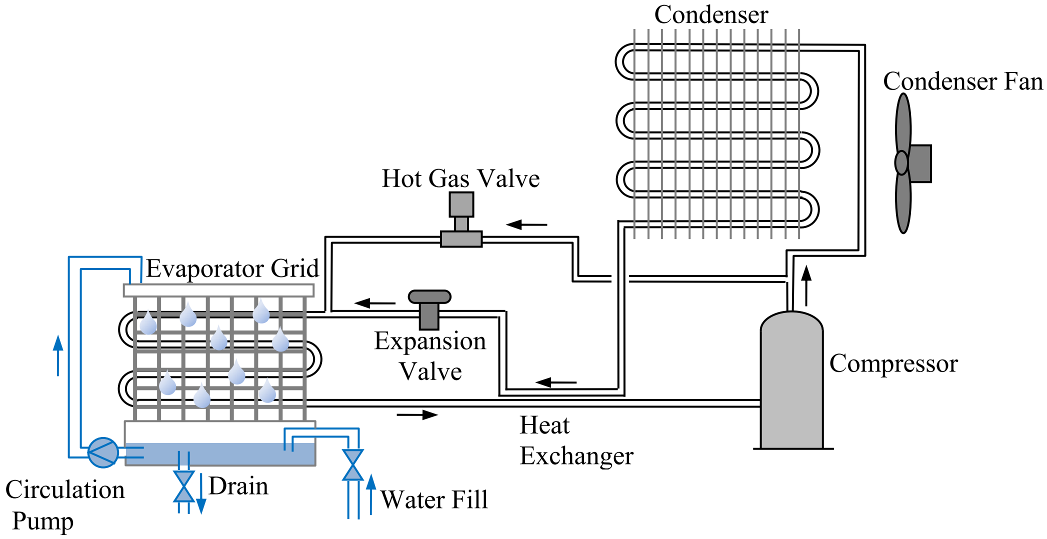

Figure 1: Schematic of an ACIM that produces batches of cubes.

Westphalen et al. (1996) described the conventional, batch, ice making process as follows:

a. Water fills the sump, which usually contains 10 – 40% more water than required to make a given batch of ice.
b. The refrigeration system is activated and sump water is circulated over the evaporator plate. During the freeze cycle, the compressor, condenser fan (for air-cooled machines) and the water circulating pump are activated.

c. The water is cooled down and gradually freezes on the evaporator grid plate.
d. Ice builds up on the plate to the proper ice batch weight as detected by some means: sump water level, compressor suction pressure, or thickness of ice on the grid plate.
e. Upon reaching the prescribed ice weight, the machine switches to the harvest mode.
f. Most machines use hot gas harvest, in which hot refrigerant vapor is directed directly from the compressor to the evaporator to warm the evaporator and melt enough ice to free the cube from the plate. Typically about 5 - 10 percent of the ice is melted during the harvest process. Once free, the ice falls by gravity into the storage bin below. During the harvest process the condenser fan for air-cooled machines is off and the water circulating pump may be operating, depending on the design. Some machines use a limited amount of hot gas for melting combined with mechanical means for removing the ice.
g. During the harvest process, water remaining in the sump is purged from the system and fresh, potable water is flushed through the system to remove impurities.
h. Water fills the sump and the system returns to the freeze mode as detected by evaporator temperature and/or time.

3. ACIM SIMULATION MODEL THEORY

The transient ice machine model incorporates a combination of algebraic and time-based differential equations for the main components, as in the vapor-compression system models created by Qiao et al. (2012). The specific operating conditions include the ambient air temperature T_a and the supply water temperature T_{W_S} .

- **Compressor:** The compressor model involves only algebraic equations. As detailed by Stroeker (1998), the amount of mass flow $\dot{m}_d$ delivered by the compressor to the components of the ice machine is dependent on compressor speed ω, compressor suction density ρ_{cs}, displacement V_d and volumetric efficiency η_v is

$$\dot{m}_d = \eta_v \omega \rho_{cs} V_d \tag{1}$$

A polytropic approach can be used to determine power consumption of the compressor, which is influenced by the evaporator pressure p_e, condenser pressure p_c, compressor efficiency η_d, and polytropic exponent κ,

$$\dot{W}k = \eta_d \left(\frac{\kappa - 1}{k}\right) \omega V_d p_e \left(1 - \frac{p_c}{p_e}\right)^{(\kappa-1)/\kappa} \tag{2}$$

Alternatively, compressor manufacturers conventionally provide rating information across an operating map in accordance with AHRI Standard 540 (2004). The compressor performance values are tabulated over a range of evaporator saturation temperatures T_e and condenser saturation temperatures T_c. The tabular data is fit to a ten-coefficient, third-order polynomial equation of the form

$$X = C_1 + C_2 T_e + C_3 T_c + C_4 T_e^2 + C_5 T_e T_c + C_6 T_c^2 + C_7 T_e^3 + C_8 T_e^2 T_c + C_9 T_e T_c^2 + C_{10} T_c^2 \tag{3}$$

where X can represent power consumption $\dot{W}k$ or mass flow $\dot{m}_d$. The appropriate rating coefficients C_i are determined by a regression and are provided by compressor manufacturers for engineers designing a system or components. Rice and Dabiri (1981) developed adjustments to Eq. (3) for the level of suction gas superheat.

An energy balance on the vapor in the compressor chamber is used to determine the temperature exiting the compressor T_d. Fisher and Rice (1983) established a compressor shell loss factor f_q to compensate for heat transfer through the compressor wall to the ambient air.

- **Expansion Valve**: The expansion valve operates during the freeeze portion of the cycle and also involves only algebraic equations. The valve restricts flow and creates a pressure differential between the low-side evaporator and the high-side condenser. Since refrigerant liquid at temperature T_l and density ρ_l is expected through the expansion valve, the one-dimensional, incompressible flow equation proposed by James and James (1987) is used to model the device,

$$\dot{m}_l = A_l \sqrt{2\rho_l (p_c - p_e)} \tag{4}$$

An effective valve flow area of A_l is fixed for an orifice or capillary tube expansion valves. Thermal expansion valve (TXV) or electronic expansion valve (EXV) provides a feedback system (mechanical or electronic) that alters A_l to maintain a certain level of evaporator superheat $\Delta T_{sh} = T_{ev} - T_e$, where T_{ev} is the temperature of the vapor exiting the evaporator. The feedback gain G_l and time constant τ_v serve as input into the expansion valve model,

$$A_l = A_{ss} + G_l\left[(T_b - T_{ev}) - \Delta T_{sh}\right] \tag{5}$$

where T_b is the sensing element (thermobulb) temperature and A_{ss} is a steady state flow area. Since the feedback for a TXV is purely mechanical, a time delay is associated with the temperature response of the sensing bulb. The response lag is modeled by

$$\mathrm{d}T_b / \mathrm{d}t = (T_b - T_{ev}) / \tau_v \tag{6}$$

- **Air-Cooled Condenser**: The condenser is modeled by dividing the total volume of the heat exchanger into N_c descrete elements along its legnth and using a finite-difference method as detailed by Bendapudi et al. (2008). As outlined in Ge and Cropper (2005), condenser heat rejection $\dot{Q}_c$ is computed using the effectiveness-NTU method,

$$\dot{Q}_c = \sum_{i=1}^{N_c} \varepsilon_c C_{p_c} (T_{c_i} - T_a) \tag{7}$$

where ε_c is the condenser effectiveness, C_{p_c} is a heat capacity, T_{c_i} is the refrigerant temperature in the *i*th element of the condenser, and T_a is the ambient temperarure. Wang and his collaborators developed appropirate models for the heat transfer correlations of fin and tube heat exchangers that depend on condenser fan flow $\dot{V}_a$, fin material and geometry, including smooth (2000), corrugated (1999), wavy (2001) and louvered (1999).

The refrigerant properties within the heat exhanger is governed by a conservation of refrigerent mass and energy along with pressure drop due to friction. These equations are integrated to remove the spatial dependence, resulting in a lumped-parameter, time-based, ordinary differential equation.

- **Evaporator**: Heat transfer from the water and into the refrigerant within the evaporator includes the interfaces through the water, ice, evaporator grid, plate, tubing and refrigerant. As with the condenser, the refrigerant within the evaporator tube is divided into into N_e descrete elements. A lumped resistance model is used to determine the evaporator heat flow,

$$\dot{Q}_e = \sum_{i=1}^{N_e} \frac{1}{R_{T_i}} (T_{e_i} - T_w) \tag{8}$$

where T_{e_i} is the refrigerant temperature in the *i*th element of the evaporator, T_W is the time varying circulation water temperature, and R_T is the effective resistance. The thermal resistance involves: 1) convection from the flowing water, 2) conduction through the ice being formed, 3) conduction through the evaporator tubes, 4) the convection to the refrigerant within the evaporator tubes. These individual interface components are

$$R_1 = 1/(\alpha_W A_W) \qquad R_2 = s_I / (k_I A_I)$$
$$R_3 = s_g / (k_g A_g) \qquad R_{4_i} = 1/(\alpha_{e_i} A_e) \tag{9}$$

with the cumulative resistance being

$$R_{T_i} = R_1 + R_2 + R_3 + R_{4_i} \tag{10}$$

For the different zones: 1) The convection coefficient for a flowing liquid over a plate is denoted α_W and A_W is the surface area of ice in contact with the flowing water. 2) The thermal conductivity of ice is k_I, A_I is the surface area of the grid, and s_I is the ice thickness. 3) The thermal conductivity of the evaporator grid and plate is k_g, A_g is the surface area of the plate, and s_g is the effective thickness of the evaporator plate. 4) The convection coefficient for the two-phase refrigerant in the ith element of the evaporator is α_{e_i} and A_e is the surface area of the evaporator tube.

Appropriate correlations were selected for the thermal conductivities and heat transfer coefficients (Incropera, 2006). The ice thickness s_I is zero at the start of the freeze cycle and increases in relation to the cumulative evaporator heat transfer $\sum \dot{Q}_e \Delta t$. The conduction through the ice is observed to be the dominant resistance.

- **Circulating Water**: At the start of the freeze cycle, the circulating water has a total mass M_{W_0} at a temperature T_{W_0} . At the end of the freeze cycle, an amount of water has been transformed into ice, having mass M_I. The remaining water in the sump has a mass $M_{W_F} = M_{W_0} - M_I$ and temperature T_{W_F} . Prior to the start of the subsequent cycle a mass M_{W_S} of water at a temperature T_{W_S} is supplied to the sump. Since the amount of water circulating is constant for each freeze cycle, $M_{W_S} = M_I$. As the supply water mixes with the remaining water in the sump, the resulting temperature of the circulating water at the start of the freeze cycle is

$$T_{W_0} = \frac{M_{W_S} T_{W_S} + M_{W_F} T_{W_F}}{M_{W_0}} \tag{11}$$

- **Liquid/Suction Line Heat Exchanger**: Suction line heat exchanger heat flow $\dot{Q}_s$ between the compressor suction line at T_{cs} and the condenser liquid line at temperature T_{cl}. The value $\dot{Q}_s$ is based on an effective contact width w_s of the tubing, the length of contact L_s and an appropriate heat transfer coefficient α_s,

$$\dot{Q}_s = \alpha_s L_s w_s (T_{cs} - T_{cl}) \tag{12}$$

- **Hot Gas Valve**: As the ice machine simulation switches to harvest mode, an alternate flow path permits refrigerant discharged from the compressor to flow through a bypass restriction defined by A_v and directly into the evaporator. With the bypass restriction, the compressor discharge pressure p_d, and mass flow through the hot gas valve $\dot{m}_v$ is governed by

$$\dot{m}_v = A_v \sqrt{2\rho_d (p_d - p_e)} \tag{13}$$

  During the freeze cycle, A_v = 0. As the hot gas valve is opened during harvest ($A_v \neq 0$), the condenser and expansion valve are bypassed. The governing equations for the other components in the system remain unchanged in the harvest mode.

The ***state postulate*** is an important principle of thermodynamics that is required to assemble the equations describing each component. The state postulate asserts that the state of a compressible substance is completely defined by two independent properties (Sontag, 2008). That is, two given properties of a superheated refrigerant are sufficient to determine any other thermodynamic property. For instance, with values of T_{cs} and p_e at the compressor inlet, the compressor suction density ρ_{cs} and enthalpy h_{cs} can be determined by using refrigerant databases such as RefProp (Lemmon et al., 2010). To reduce computation time, Laughman (2012) created look-up tables that store thermodynamic properties for selected refrigerants that are generated from a database. The look-up tables are used to quickly determine necessary state variables of the refrigerant as it flows through the components.

The theories and equations presented above are general and equally apply to the freeze cycle and harvest. During the harvest, the bypass valve is opened and heat is removed from the ice and into the evaporator.

The simulation will increment through time t until a specified number of freeze and harvest cycles are encountered. Implicit routines within the SimScape™ modeling environment (Mathworks, 2015) are used to solve set of overall algebraic and differential equations as needed such, that Kirchhoff's first and second laws are satisfied at the nodes where components are connected. That is, all through variables (mass flow rate and heat flow rate) need to sum to zero and all the across variables (pressure and enthalpy) should be equal.

4. MODEL OPERATION OVERVIEW

The model of the ACIM is executed as follows:

1. Simulation begins with a specified mass of ice to be formed within evaporator grid, M_I, and physical parameters such as condenser dimensions V_c, and evaporator dimensions V_e.
2. Water supply at a designated temperature T_{w_s} is mixed with water in the sump.
3. A startup system (evaporator and condenser) pressure $p_{e_0} = p_{c_0}$ is designated, refrigerant charge (mass) is calculated.
4. The transient simulation begins with the freeze stage. A schematic of the ACIM model operating in freeze mode is shown in Fig. 2.

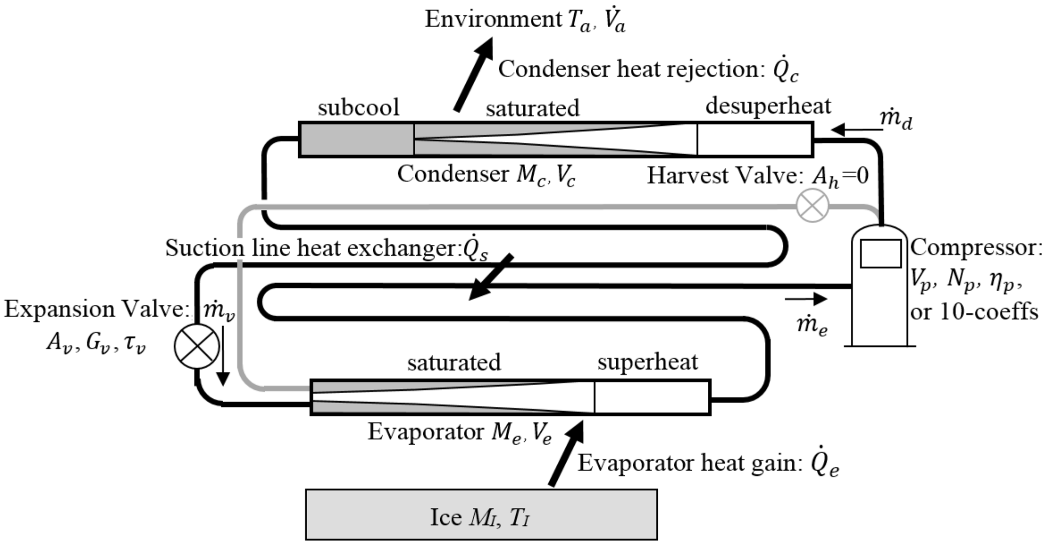

Figure 2: Ice machine model operating in freeze mode.

5. Evaporator heat flow $\dot{Q}_e$ is based on standard refrigerant-side heat exchanger models. Water-side equations involve custom developed equations for heat transfer from evaporator tube wall to flowing water through an increasing ice resistance.
6. Once the specified amount of ice has been formed (M_I) with corresponding thickness (s_I), the harvest mode is initiated. A schematic of the ACIM model operating in harvest mode is shown in Fig. 3.

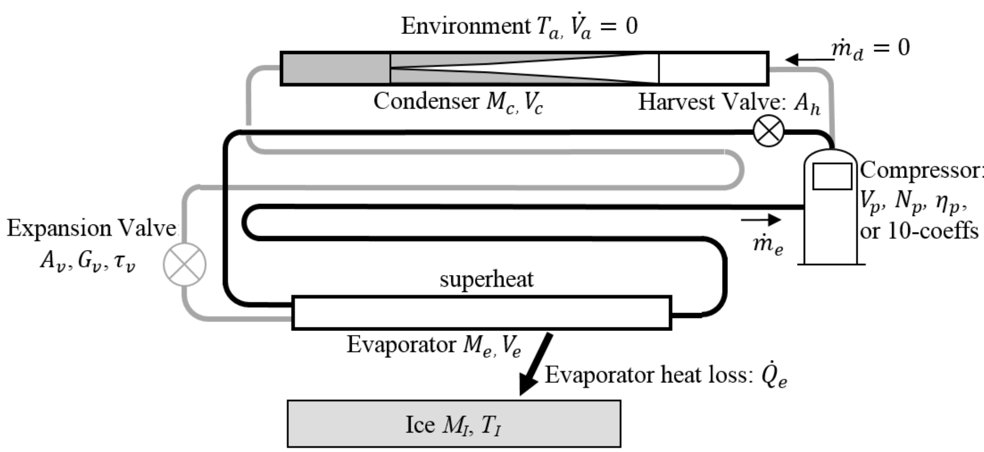

Figure 3: Ice machine model operating in harvest mode.

7. Hot-gas bypass valve is opened during the harvest cycle, routing the compressor discharge line directly into the evaporator. During harvest, a restriction area A_v is implemented within the bypass valve.
8. Harvest is complete when a specified percentage of the ice is melted.
9. Water inlet at a designated temperature T_{W_s} is used to replenish the mass of ice harvested ice, and mixed with existing water in the sump.
10. The simulation returns to the freeze stage (Step 6).

5. RESULTS

A 500lb, instrumented ACIM was equipped with sensors to measure the operational characteristics of the machine. The instrumented machine was run at various operating points defined by the ambient temperature and the water inlet temperature. A summary of the experimental values (*E*) and the predictions made by the simulation model (*S*) are given in Table 1. Also provided is the percent absolute value of error (Δ) between the experiment and simulation.

Table 1: Comparison of summary results between experimental results and simulation model.

	100/110 °F			90/70 °F			70/50 °F		
	E	*S*	Δ	*E*	*S*	Δ	*E*	*S*	Δ
Cycle time (min.)	26.11	25.42	2.6%	18.3	17.44	4.7%	14.5	15.21	4.9%
Ice per 24 hrs. (lbs.)	257.8	283.7	2.9%	393.4	412.9	4.9%	496.6	473.4	4.7%
Energy input per 100 lb. (kWh)	19.78	20.04	1.3%	18.28	17.54	4.0%	16.88	16.73	0.9%
Energy input per 24 hrs. (kWh)	23.75	24.04	1.2%	21.93	21.05	4.0%	20.25	20.08	0.9%

Figures 4-5 provides a comparison of the transient response of pressures, temperatures and compressor power at various locations on the ice machine.

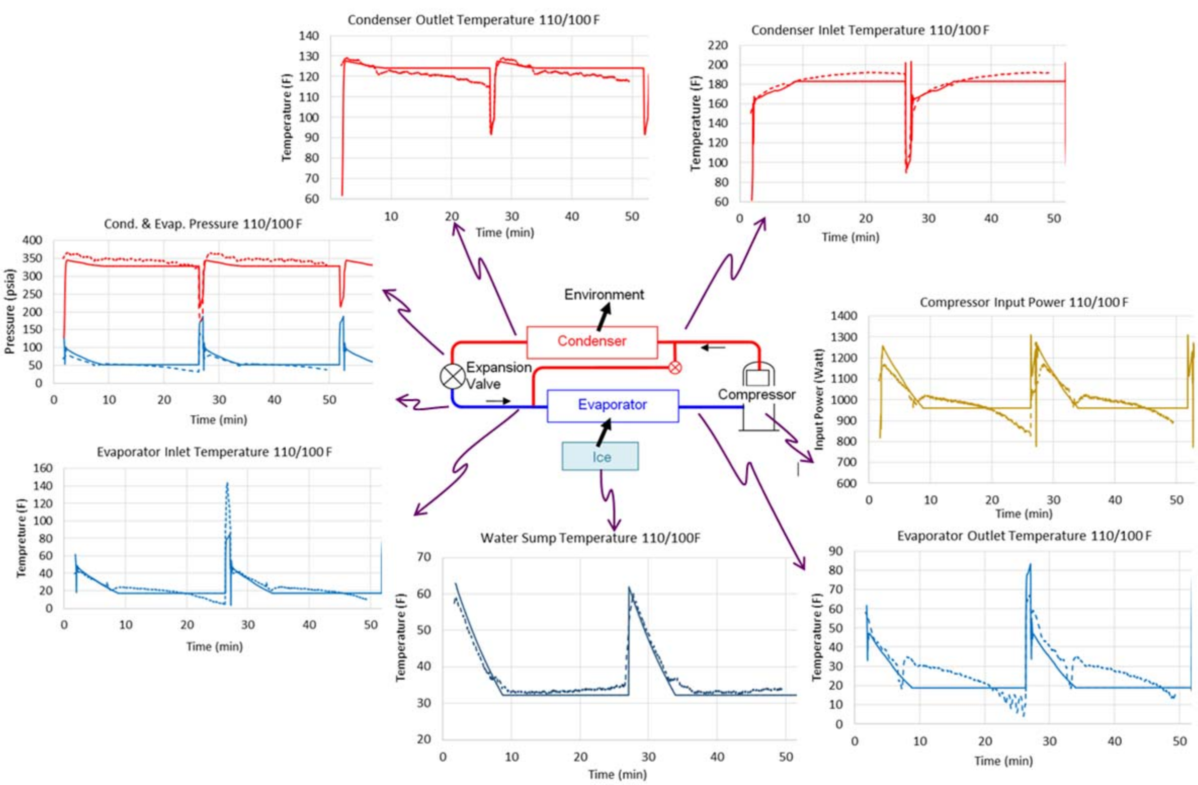

Figure 4: Graphical representation of the transient comparisons at 110/100 °F.

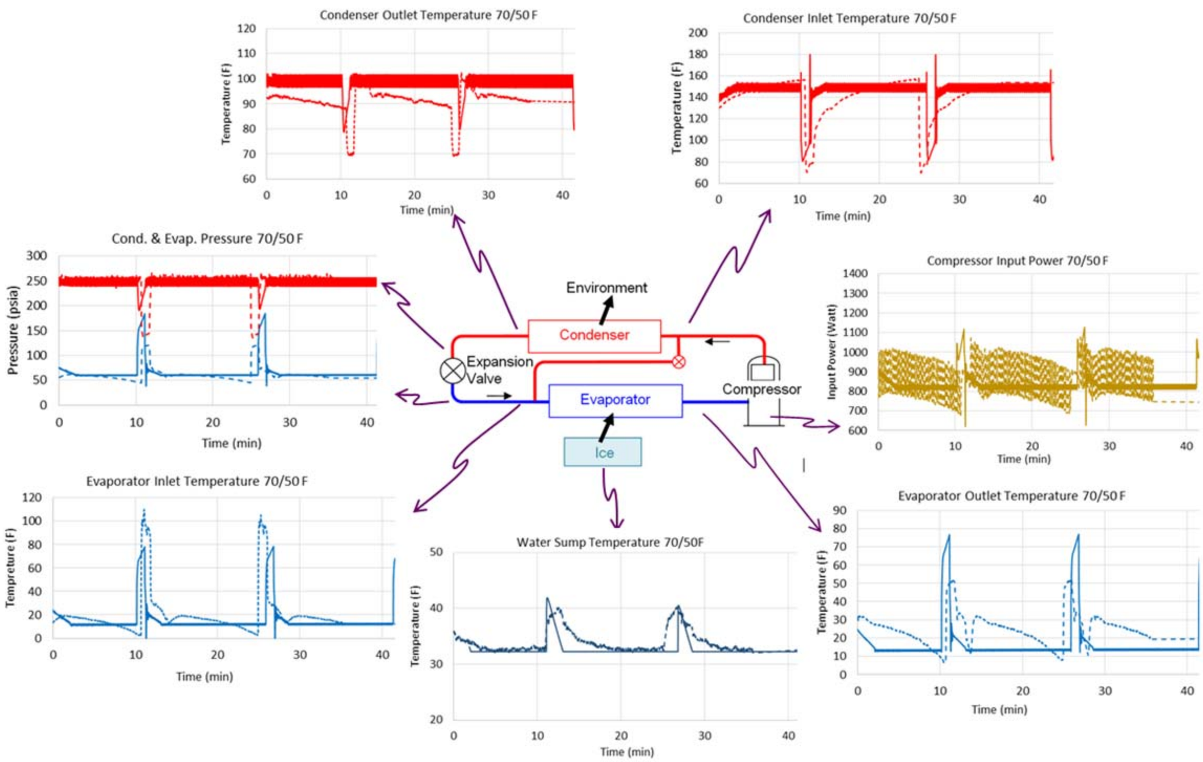

Figure 5: Graphical representation of the transient comparisons at 90/70 °F.

6. CONCLUSIONS

This paper outlined a transient simulation model of the operation of an automatic commercial ice maker. The model is based on fundamental, physics-based principles of individual system components. Governing equations for the compressor, condenser, expansion valve, and connecting tubing were adapted from prior research available in the literature. A custom evaporator model was developed to describe the heat transfer between the refrigerant and water flowing over an ice-formation grid. Simulation results from the model were compared with the experimental data of a fully instrumented, standard 500 lb capacity ice machine, operating under various ambient air and water inlet temperatures. Key aggregate measures of the ice machine's performance include the freeze and harvest cycle time, energy input per 100 lb of ice, and energy usage during 24 hours. For these measures, the model's accuracy is within 5% for a variety of operating conditions.

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