Description and validation of a computer based refrigeration system simulator

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Abstract

This paper describes and evaluates the validation of a novel software package which simulates the transient and steady-state operation of whole refrigeration systems of the type used for the storage and processing of food. This software allows practitioners to study the implications of design choices in terms of energy usage and carbon generation in the storage and processing of food stuff by refrigeration.. The software can also provide refrigeration system owners and designers with information on how a system's energy consumption might be reduced.

Highlights

- A useful general purpose computer based simulation model of a whole refrigeration system, which includes the refrigerator, refrigerated space, outdoor and indoor ambient conditions and food product, is described and evaluated.
- The software is provided free of charge to users and can be down-loaded through the internet.
- The model is currently limited to simulation air to air refrigeration systems of the type our research showed to be most commonly used in the food processing and storage industry.

Keywords

Refrigeration, computer simulation, modelling, validation

1. Introduction

Food refrigeration accounts for significant carbon emissions on a worldwide basis [1]. In the UK it accounts for around 11% of the total electricity consumption [2]. Few food manufacturers can now ignore energy use, from a commercial or environmental point of view. Refrigeration makes up a large proportion of the energy used in food manufacturing and there is a potential to optimise refrigeration systems that in the past have been purchased on the economics of initial cost rather than long term energy savings.

A large proportion of the energy use in the food cold chain and the associated carbon emissions are attributable to commercial food processing. In many cases food processing plant are bespoke in their design. Their capacity, size and function also vary depending on specific manufacturing needs. Energy savings in industrial and commercial food processing maybe possible and can be achieved by modifying the food process itself and/or the design of the refrigeration system.

Users and manufacturers of refrigeration systems can investigate the potential of design changes to save energy either experimentally or by simulation. Experimental investigation can be prohibitive in terms of time and available resources, whereas simulation is reliant on the models available. To fully understand the integration between the dynamic operation of the refrigeration system and the food being cooled a fully integrated model is required. The need to fully integrate the refrigeration system process and the heat

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transfer from the food was first highlighted by Cleleand in 1990 [3]. He stated that an accurate model of a complete food refrigeration system is impossible unless both the refrigeration process and mechanical plant are considered simultaneously. Since that time only a limited number of integrated models have been developed with most models being either steady-state, design-point models, not fully integrated system models or far too complex for a practitioner to operate.

In 2005 the Research Priorities Group at defra identified the need to develop more energy efficient technologies for use throughout the food chain whilst not compromising food safety and quality. They therefore funded a collaborative project between 4 UK universities [4] which identified the need for a simple to operate integrated modelling tool.

This paper describes and evaluates the validation of the model developed. The Vapour Compression Refrigerator Simulator (VCRS) model allows practitioners to investigate the effect of equipment design and process changes, such as loading schedules, on system performance and energy consumption. The VCRS package is free of charge and can be downloaded from the project website (http:/www.grimsby.ac.uk/what-we-offer/DEFRA-Energy), or from the corresponding author by writing to ianweames@aol.com.

2 A description of the VCRS

Figure 1 shows a schematic view of the VCRS. Mathematical models were written for each part of the system illustrated in Figure 1 and these are described, in outline along with some of the associated software functions and model programming options, in the following:

INSERT FIGURE 1 HERE

2.1 Evaporator

The VCRS model assumes the evaporator to be an air-cooling finned-coil DX type. Our study showed that at the time of writing the software this type of evaporator was the most commonly found in refrigerated food processing and storage equipment, such as blast coolers and freezers and food storage units. The evaporator model in the VCRS is a non-steady state type that assumes two heat transfer zones (latent and sensible), based on the lumped parameter method, described by Ding [5] and Bendapudi [6], combined.with the NTU-effectiveness method of analysis [7]. The model also allows for pressure losses in both on the air and refrigerant flows and partial flow blockage of the air-side due to frost formation and ice build-up. At present the defrost simulation assumes an electrically heated element embedded in the coil. A user can choose to trigger defrost either by time, at fixed intervals, or by the air-on to the coil approach temperature difference.

At the simulation design stage, when a user enters the data for the system they wish to model, they can select to enter theoretical values for an evaporator design in to the model or to input manufacturers' catalogue data. Default data is supplied by the software to help users get started.

2.2 Condenser

The VCRS model assumes a conventional finned-tube dry-air cooled condenser. The condenser model is analogous to the evaporator model except it used three heat transfer zones: Sensible cooling for the de-superheating section, latent cooling during condensation and a sub-cooling zone. The model used the lumped parameter method combined with the NTU-effectiveness method of analysis. Variations in pressure losses on both air-side and refrigerant side-side due to flow changes are accounted for by the model. Cooling air-on temperature and relative humidity track those of the outdoor ambient air, described later in Section 2.9. An option is provided by the VCRS to input either theoretical condenser data or to use manufacturer's catalogue data.

2.3 Compressor(s)

The performance of a refrigerator's compressors is of fundamental importance to the energy efficiency of the system as a whole. However, it is strongly affected by the performance of the heat exchangers and expansion device, [8, 9, 10]. The VCRS provides an option to simulate a system with up to six reciprocating compressors operating in stages. Drive motors are assumed to be compressor suction gas cooled three-phase induction types [10], of either hermetic or semi-hermetic design. The compressor model takes account of changes in volumetric efficiency, isentropic efficiency, (based on indicated work), mechanical efficiency and electro-mechanical efficiency of the drive motor. In addition to staged compressors, other compressor control options provided by the VCRS include variable speed and fixed speed. Transient response to motor speed changes is also simulated. At the design-point condition, when the user is setting up the simulation model, the VCRS allows for either theoretical data of manufacturer's catalogue data to be inputted. The compressor model was validated at steady-state against catalogue data supplied by a manufacture.

2.4 Thermostatic expansion valve

The operation of thermostatic expansion valve is described by Dossat [11]. The VCRS assumes a liquid charged valve which has the option to be either internally and external equalised. The liquid in the sensing bulb is assumed to be the same as that in the refrigerator. The model calculates both steady-state and transient variations in refrigerant flow. The steady-state model has been validated itself against valve manufactures' data.

2.5 Fans

The evaporator fan and fan motor also form part of the VCRS model. Once the design-point evaporator air flow, air-side pressure loss and fan speed are inputted into the model, the fan laws [12, 13] are used to simulate air flow through the evaporator and condenser fans in response to variations in pressure loss, fan speed and changes in aerodynamic efficiency. Three fan speed control options are provided by the VCRS. These are: variable, fractionally staged or fixed speed. VCRS also simulates the transient response of fans to speed changes.

2.6 Electric motors

The VCRS models the compressor and fan motor torque/speed curves by scaling an example characteristic for a three-phase squirrel cage induction motor given by Eastop and Croft [14]. A motor efficiency characteristics' is automatically generated from input design-point data, which the user may assume or take from NEMA values [15] provided in a 'HELP' page.

2.7 Pipe lines

The diameters and lengths or the suction, hot-gas and liquid pipelines are required by the VCRS. Assistance is provided on choosing approximate pipe sizes in a 'HELP' page. VCRS calculates pressure losses due to pipe-fittings and friction and also estimates heat loss or gain by the fluid, using standard correlations, [13, 16]. VCRS also provides options for pipe insulation thickness and type.

2.8 Refrigerant fluid

Thermodynamic and transport properties of six refrigerant fluids commonly found in commercial systems are provided for the user to select from.

2.9 Out door air conditions

The VCRS model takes account of changes in outdoor ambient air conditions by using an ambient temperature and humidity correlation based on a 29 year averages for 7 UK cities provided by the UK's meteorological office, [17]. The user has the option of choosing fixed conditions or variable conditions. For the variable weather data option the users needs select both the location of the plant (from a list of seven major UK mainland cities) and the month of the year for which the simulation is required. With the variable outdoor weather option set, the model calculates the diurnal variation in ambient temperature and humidity ratio. Variations in relative humidity, dry and wet-bulb temperature, air density and enthalpy are also calculated using psychrometric methods described by Jones [18] and based on sinusoidal variation in dry-bulb diurnal temperature between maximum and minimum values by assuming these occur at 03.00hrs and 15:00hrs daily and by assuming the air moisture content is constant throughout the subject day and gives a relative humidity of 50% at 15:00hrs. Deviations from the maximum and minimum temperature data provided by the meteorological office can be selected in order to simulate extremes of ambient air temperature.

2.10 Cooler or freezer room or cabinet

Simplified lumped parameter models are used for the enclosure shown in Figure 1. VCRS calculates time-wise variations in air humidity and temperature. The calculation takes account of door opening/closing, continuous and intermittent electrical loads, such as lighting, people load (both sensible and latent), racking type, the food product loading schedule and the heating effect of the food product itself. These conditions are set by the user before running the VCRS model. Also, the structure of the enclosure and its physical size must be inputted before running the model. Because the outside air temperature can be different for each wall of the enclosure, three options are provided to set temperature values outside for each wall as follows: (i) Tracking outdoor ambient temperature, (ii) User determined maximum and minimum temperature values, or (ii) Fixed outside wall temperature selected by the user.

2.11 Food product

A finite difference transient heat transfer model was written for food products, similar to that described by Evans, Russell and James (1996) [19]. The thermo-physical properties of food were determined from its contents of fat, carbohydrate, water, mineral ash, and protein. The model determines the specific heat and conductivity as functions of temperature and phase, as freezing is also taken into account. The model allows for 2 food shapes; cylindrical and slab. ASHRAE Handbooks [20] provides a source for the necessary data required to run the food model within the VCRS

2.12 System control

VCRS offers the following control system options:

- Six system thermostat control setting options
- Three evaporator fan speed control options (manual, automatic variable and automatic fractional)
- Three compressor control selections, (manual speed control, automatic speed control based on temperature and staged, with up to six compressor stages available)
- Two automatic electric defrost control options, set either on by time and on by Approach Temperature Difference (DT1).
- High and Low Temperature cut-out controls
- Compressor and fan motor start-up delay controls.

In addition users can alter the time-constant and cooling capacity of the thermostatic expansion value to simulate the effects of both a badly positioned sensing bulb and an over or undersized valve.

2.13 Other VCRS functions

VCRS requires a significant amount of input data before it is ready to run. Unfortunately, this is unavoidable due to the complexity of ‘real’ refrigeration systems. However, the VCRS provides design screens for the various components, a *WIZARD* function and *HELP* pages to assist users set up the model. In addition complete sets of optional default data are provided and traps are included to help reduce errors of logic in input data. Input data can be saved to file to be available later if the user wishes.

Once all the input data needed to run the simulator have been set and the RUN button has been clicked, a Run-time screen, shown in Figure 2, acts as both control and instrument panels. It displays in ‘model’ time the instantaneous values of output data, such as evaporator cooling effect, ambient temperature, energy consumption and power values as well as performance data such as instantaneous COP and COSP values. This data can be saved to a spreadsheet for later analysis. Continuously updated graphical output of selected operating parameters is also provided.

INSERT FIGURE 2 HERE

3. Validation case study

3.1 Description of the test facility

A study was undertaken to validate the VCRS's ability to simulate a whole refrigeration system including the food being cooled. The authors were given access to a typical industrial batch pie manufacturing process. This consisted of the preparation and cooking of a pie mixture in tureens followed by a 'rapid' cooling process using an air-blast chiller room before being added to prepared pastry cases. The authors measured various component data or took it from manufacturer's specifications or, where these were absent, estimated or calculated from measured data. The remainder of this section describes the data which was collected and it acquisition.

3.2 The refrigeration unit

The refrigeration unit, which formed the subject of this study, was an air-to-air DX system powered by a single semi-hermetic compressor with conventional dry-air-cooled condenser and air-cooling evaporator. The evaporator had two vertical evaporator banks each with three axial electric motor/fan units. The measured data for the evaporator, including its overall dimensions, are listed in Table 1. Total electrical power absorbed by the evaporator fans was estimated to be 1.5 kW. The evaporator also had an electrical resistance defrost coil.

INSERT TABLE 1 HERE

3.3 Compressor and condenser installations

The system used a semi-hermetic Presscold, compressor unit, (Model R1500/0138 S/D , Serial No 9483 (380/420VAC, 3-phase, 32 FLA, 40 LRA). Table 2 lists in the relevant manufacturer's specification needed to determine input data for the VCRS. The system included a high-pressure liquid receiver (approximately 100cm x 30cm diameter), filter/drier in the liquid line and a hot-gas line oil separator/muffler. Table 3 lists the measured data for the condenser.

INSERT TABLE 2 HERE

INSERT TABLE 3 HERE

3.4 Refrigerant pipelines

The suction line was insulated with a standard type of foam insulation. Both the liquid return and suction pipelines were about 35 metre in length and located outside the factory building and hung for most of their length on a 'sunny' wall. The input data for these is listed in Table 4.

INSERT TABLE 4 HERE

3.5 Air-blast-chiller installation

The air-blast chiller was a push-through rack chilling tunnel type. Figure 3 shows schematically the plan view layout of the unit and the direction of the air-flow through the evaporator, fans and across the food product. The relevant dimensions of the unit are listed in Table 5. Figure 4 shows a photograph of the chiller cabinet.

INSERT FIGURE 3 HERE

INSERT TABLE 5 HERE

INSERT FIGURE 4 HERE

3.6 Operation and control strategies

At the time of recording the data reported in this paper the blast-chiller operated daily between approximately 08:00 and 23:00, Monday to Friday. A thermostat was used to control the air-on temperature to the evaporator coil by switching the compressor on or off. The upper set-point temperature of the thermostat was -10°C and its lower set-point was -12°C. Evaporator coil defrost was by means of electric resistance heater coils, which was embedded between the evaporator tube rows. The defrost cycle was timed to come on every 4 hours and to remain on for 30 minutes. Evaporator fans were switched off during defrost cycles but left running when the compressor was switched-off by the evaporator air-on thermostat. The target cooling temperature for the pie filling was 10°C. According to the company, this cooling process could take up to 12 hours, depending on the air gap left between layers of tureens which controlled the air flow.

4. Experimental method

The blast-cooler was switched on, empty of food-product, at 09:08 on 13th Aug and left to pull down to its normal working temperature with both loading doors closed.

The pie filling was cooked in tureens in batches of approximately 60 kg. Three to four batches were required to fill one complete trolley of 28 tureens (7 trays per batch). The first batch was placed on the test trolley at 10:42 hrs and left to cool in still air until the last batch was placed on the test trolley. At which point the troy was pushed into the cabinet.

The mass of an empty stainless steel tureen was measured and found to be approximately 1.25 kg. A typical full tureen (tray + pie filling mixture) weighed 9.3 kg before cooling and 9.27 kg after cooling, so that the amount of water evaporation was negligible. This was achieved by covering all tureens containing pie filling mixture with 'cling-film' throughout the test. Six tureens out of the 28, containing pie filling, in the test trolley, were fitted with multipoint thermocouple probes to determine the slowest cooling point near the geometric centre of each tray. Two test-point tureens were positioned on the top shelf, centre shelf and bottom shelf of the test trolley. All thermocouple sensors were recorded at one-minute intervals using Comark 2014 loggers. The power consumption of the refrigeration system was also recorded at one-minute intervals using a Sinergy e-Tracker energy logger throughout the trial.

Two other trolleys were filled with empty tureens, each holding 2 columns of 14 empty tureens were prepared for the test. At 12:46:11 the entry door to the cabinet was opened and the two "empty" trolleys were wheeled in, followed by the "full" test-trolley (trolley 3) and the door closed at 12:47:53. Chilling was halted at 18:41 when the trolleys were removed from the exit door and the blast-cooler was switched off.

The recorded experimental measurements were entered into spreadsheets which were later analysed. The parameters measured are listed in Table 6

INSERT TABLE 6 HERE

5 Validation results and discussion

The data required from the validation test to run the VCRS software model are listed in Table 7 and comparative results between the theoretical data produced by the VCRS software and the measured data are shown in Figures 5 to .

INSERT TABLE 7 HERE

The results plotted in Figure 5 compare the VCRS prediction and experimentally measured variation in cabinet dry-bulb air temperature over 300 minutes of the cooling process, which included the loading of the food product and the two trolleys filled with empty tureens, and a complete defrost cycle. The results show on-off cycling of the compressor with a period of approximately 10 minutes. In the first 100 minutes the VCRS model predicted 13 cycles whilst 12 cycles were measured. This close correlation is more evident in Figure 6, which shows the close correlation between the measured and predicted cabinet temperature during this first 100 minutes. These experimental data also show that the system had cyclic variation in the maximum and the minimum air temperatures was present. The VCRS data showed similar, but more regular, oscillations.

INSERT FIGURE 5 HERE

INSERT FIGURE 6 HERE

INSERT FIGURE 7 HERE

The results in Figure 5 shows a significant difference between the measured and predicted air temperature data when the food was loaded and an even larger difference at the end of the defrost cycle, when it was measured at 18 ^{o}C, whereas the VCRS model predicted that it would be 10 ^{o}C. This difference can be explained by the additional heating effect introduced with the 56 empty tureens and the two additional trolleys. These trolleys and empty tureens were introduced into the experiment in order to replicate the air flow conditions over the food product that would exist when the cabinet was being used at its maximum capacity. However, at the time of writing the software this scenario, of loading a cooling cabinet with empty tureens, was not envisaged and so the VCRS model at present does not allow for the condition.

Although the empty tureens have a relatively small thermal mass compared with the filled tureens they have potentially a high rate of heat emission because of their large surface area to mass ratio. When the empty tureens and their trolleys where pushed into the cabinet their temperature would have been between 28 ^{o}C and 36 ^{o}C greater than the cabinet area temperature. The high rate of heat transfer that would have existed from the empty tureens tended to slow the fall in cabinet temperature whilst the compressor was running and when it was switched off during the defrost cycle the additional heat rate of heating probably caused the larger than anticipated difference between the measured and predicted air temperature results. When the results in Figure 6 and Figure 7 are compared it is clear that something is damping the measured air temperature oscillation compared with the predicted values. This too may be due to the presence of the empty tureens.

The results shown in Figure 8 compare the measured and predicted evaporator refrigerant saturation temperatures. Like those results shown in Figure 5 and 6, there is a close correlation between the data sets up until the defrost cycle begins. It should be noted that the defrost cycle for the experiment began at time = 237 minutes whilst the VCRS model input was set at 240 minutes.

INSERT FIGURE 8 HERE

Figure 9 shows comparative results for measured and predicted condenser saturation temperature throughout the experiment. The difference between the measured and predicted maximum appears to be caused by the error in the outdoor dry-bulb air temperature model when compared with the measured data.

INSERT FIGURE 9 HERE

Figure 10 shows the variation in food product temperature, taken at the centre of the centres of 6 tureens tray and then averaged, compared with the VCRS prediction for the food temperature at the geometric centre of a slab with the same thermo-fluid properties as the food. During the initial hours of chilling, the measured temperatures fell slightly more quickly than the model predicted, with an average difference of 2.3°C at a particular time. The most likely reasons for this are inaccuracy in one or all of the following: composition data, measured air velocities, conversion from air velocities to heat transfer coefficients. Also shown in Figure 10 are measured and predicted evaporator air-off temperatures. These matched closely during the empty period. The main difference during loaded run-time was ‘wider’ cycling which may have been for the reasons discussed previously and/or related to thermostat damping. However, the close correlation of these results is encouraging.

INSERT FIGURE 10 HERE

Figure 11 compared the measured and predicted energy consumption of the air-blast cooling cabinet over the period of the experiment. Values shown are averaged over 15 minute intervals. The total measured energy consumption for the 10 hour period was 64kWh. The VCRS model predicted an energy consumption of 58.9 kWh for the same period, which is an error of -8%.

INSERT FIGURE 11 HERE

6. CONCLUSIONS

In order to write a ‘general purpose’ computer simulation model of a system that can be easily programmed it is necessary that the amount of detailed input data should be as small and readily available as possible, without loosing out too much to accuracy. The quantity of input data needed to run the VCRS model has been kept to a size that makes its programming practical. The type of data needed, as listed in Table 7, can be collected by simple physical measurement or from manufacturers catalogues. From this study the following conclusions are made:

- The VCR software offers designers and users a means of simulating the transient and steady state energy performance of existing and proposed refrigeration systems, particularly those used for refrigerated food storage.

- Real component data for compressor, evaporators and condensers can be taken form manufacturer’s catalogues to run a simulation. Where such data is not available then scientifically based models are provided.

- The VCR software allows users to simulate the effects on energy consumption due to variations in outdoor and store temperatures, food properties and cooling air velocity, amongst other things.

- The software also allows users to simulate the effects of control system choices, particularly defrost power and timer settings, evaporator and condenser fan speed control, TEV settings, compressor stages and speed control.

- The validation case study described in the paper showed that the model predictions compared favourably with measured data both in terms of absolute values of results and general trends in terms of the test system’s temperatures and energy consumption over the test day.

- It is hoped the results of this work will provide a useful addition to the tools available to the designer.

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Table1: Measured evaporator data

Width	66 cm
Height	162.5 cm
Depth	30 cm
Coil pipe dia	6.4 mm
Coil length	60 rows x 4 deep
Fin spacing	8 fpi
Mean air-on velocity	2.1 m/s

Table 2: Compressor data,(courtesy of Copeland Ltd)

Compressor model:	PR1500/0138
Copeland compressor model input data:	
Motor speed:	50Hz , 4-pole
Compressor speed	1450 rpm, (24.167 hz)
Required refrigeration capacity: 12kW	
Refrigerant:	R22
Suction return temperature:	20.0^{o}C
Liquid sub-cooling:	0.0K
Condenser temperature:	30.0^{o}Csat
Evaporator temperature:	-25^{o}Csat
Performance at the above conditions:	
Refrigeration capacity:	13.19 kW
Electrical power consumption:	5.83 kW
Current at 400VAC:	13.32A
Mechanical data:	
Displacement at 50Hz:	49.9162 m^3/h
Optional capacity steps:	33.3%
Number of cylinders:	3
Bore:	61.925mm
Stroke:	63.5mm
Gross weight:	176kg
Max high pressure:	25bar
Suction inlet diameter:	1 5/8 inch
Discharge diameter:	1 1/8 inch

Table 3: Measured condenser data

Width	66 cm
Height	77 cm
Depth	36 cm
Coil pipe dia.	9.7 mm
Coil	30 rows x 6 deep
Fin spacing	8 fpi
Mean air-on velocity	3.4 m/s

Table 4: Refrigerant pipeline data

Evaporator to compressor = 38.5 m x 28.6 mm O/D Compressor to condenser = 3 m x 28.6 Condenser to receiver = 3 m x 19.1 mm O/D Receiver to evaporator = 38.5 m x 19.1 mm O/D Insulation = Armaflex type, 15 mm thickness

Table 5: Air-blast cooling cabinet dimensions

Internal length = 195.0 cm Internal width (between inner wall and evaporator face) = 86.5 cm Internal height = 195.5 cm External length (excluding doors) = 221.0 cm External width = 163.0 cm External height = 226.5 cm

Table 6: Data recorded during experimental testing

Parameter	**Unit**	**Parameter**	Unit
Air-off evaporator coil	°C	Compressor discharge temperature	°C
Air-on evaporator coil	°C	Compressor crankcase temperature	°C
Evaporator liquid temperature	°C	Air-off condenser	°C
Evaporator saturation temperature	°C	Air-on condenser	°C
Evaporator suction temperature	°C	Condenser refrigerant temperature in	°C
Air relative humidity inside chiller	%RH	Condensing temperature	°C
Air temperature outside chiller	°C	Condenser refrigerant temperature out	°C
Air relative humidity outside chiller	%RH	External air relative humidity	%RH
Compressor suction temperature	°C	External air temperature	°C
Total electrical input	kW	Compressor and condenser power	kW

Table 7 VCR input data

Air-blast cooler temperature at start-up = 12.8 ^oC
Evaporator saturation temperature at start-up = -9.6 ^oC
Air temperature outside room = 19.3^oC(max) to 18^oC(min)
Outdoor ambient dry-bulb temperature set at Bristol for August with a maximum variation from standard day of -1.7 ^oC and a minimum variation of -5.5 ^oC
Defrost heat = 7 kW
Defrost period = 20 minutes (on) every 4 hours
Defrost heat control: Heat switch OFF when evaporator saturation temperature > 0^oC
Fan setting during defrost = OFF
Compressor motor minimum shutdown timer setting = 6 mins
Store thermostat upper set-point = -6^oC, lower set-point = -14oC set on evaporator coil air-on temperature.
Design-point condenser fan power = 2.7 kW (estimated)
Design-point evaporator fan power = 1.5 kW
Compressor settings Design-point power = 5.544 kW Compressor speed = 24.1 hz Motor synchronous speed = 25 hz Polytropic efficiency = 83% (estimated)
Thermostat control: measured variable = Te(air-on) upper set-point = -10 ^oC lower set-point = -12 ^oC Re-start delayed response time = 3 minutes for compressor starting and stopping Evaporator fan remains on when compressor shut-down by thermostat Evaporator fan switches off during defrost

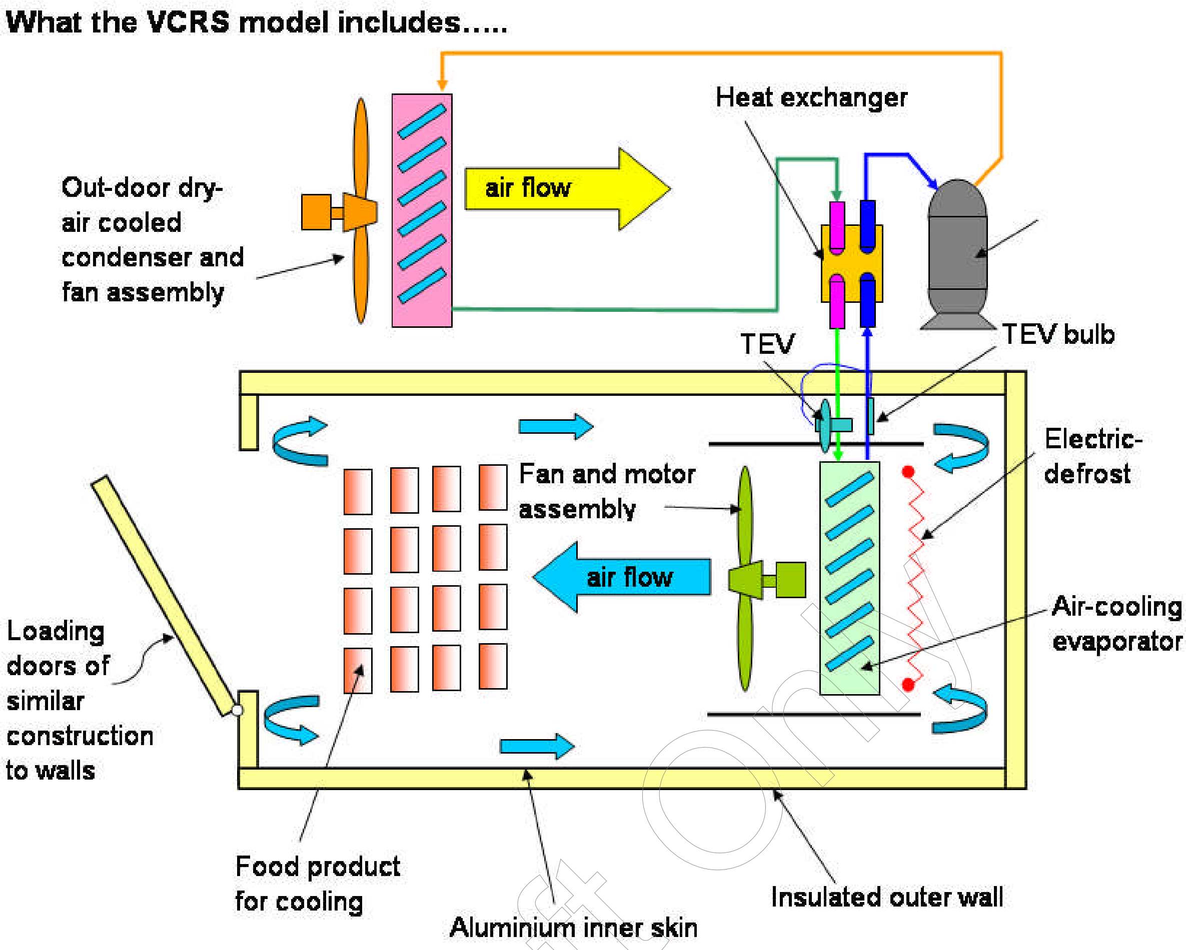

Figure 1: Schematic of refrigerated model cold space, from VCRS-HELP

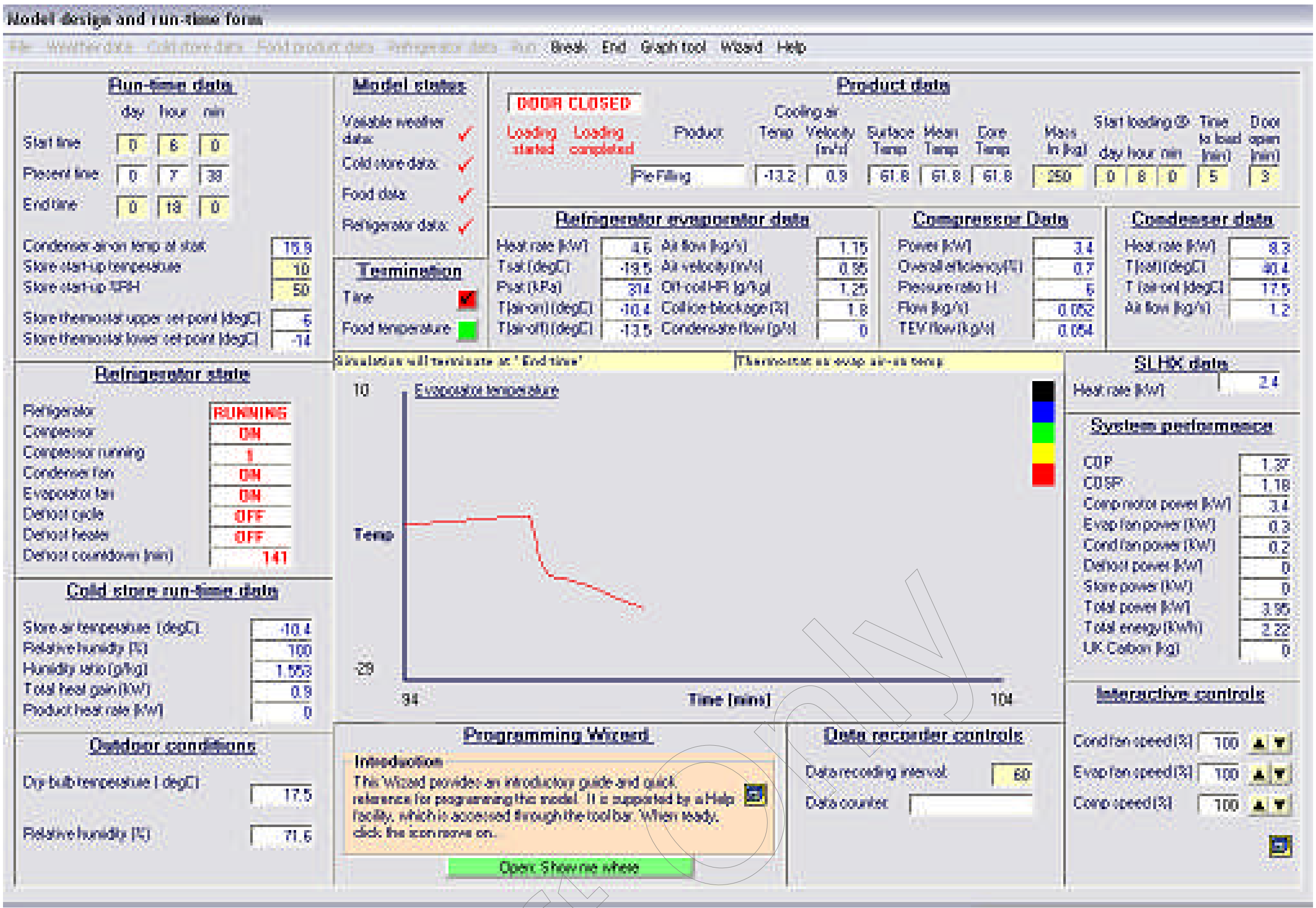

Figure 2: Run-time screen during run-time

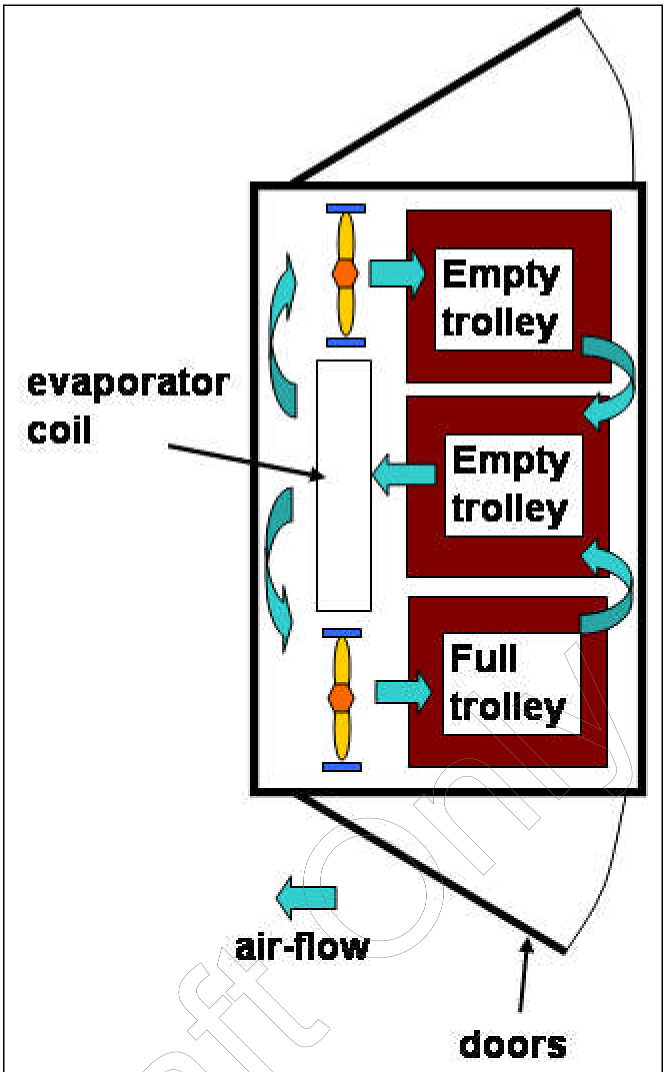

Figure 3: Schematic plan view of the air-blast cooler cabinet

Figure 4: Pie-filling air-blast cooling cabinet (door width = 71 cm, height = 174 cm).

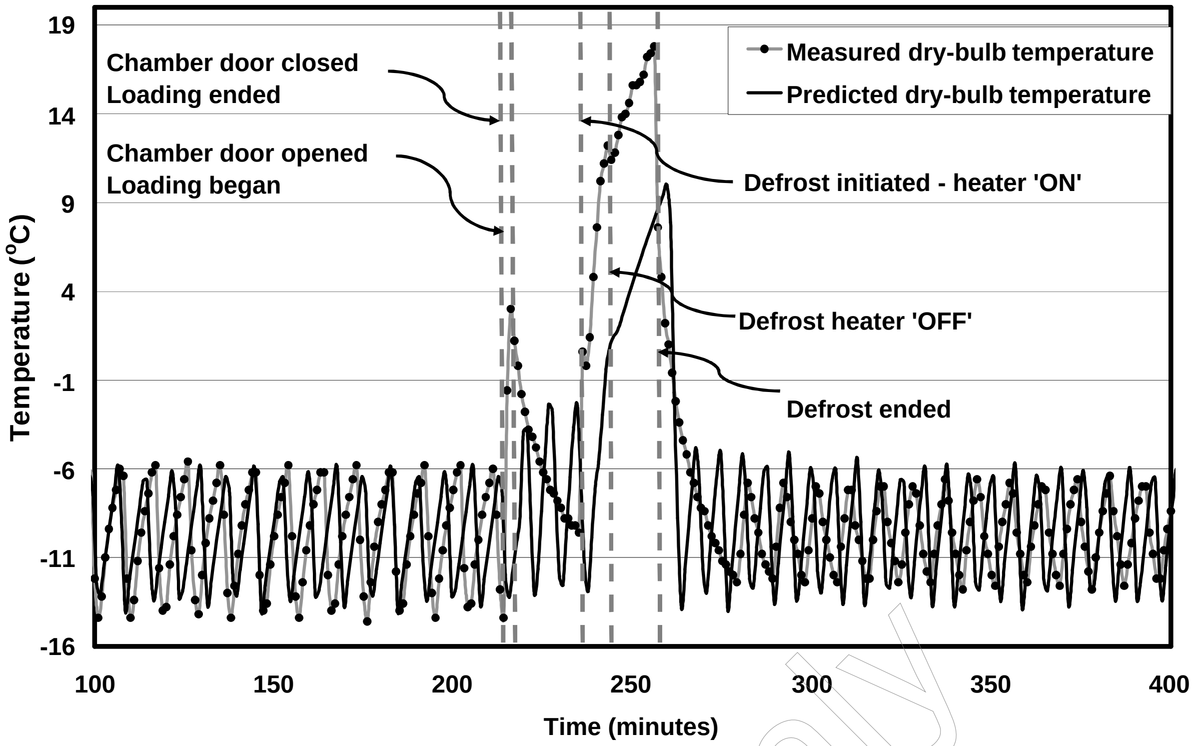

Figure 5: Comparison between measured and predicted air-blast cooling cabinet dry-bulb air temperatures.

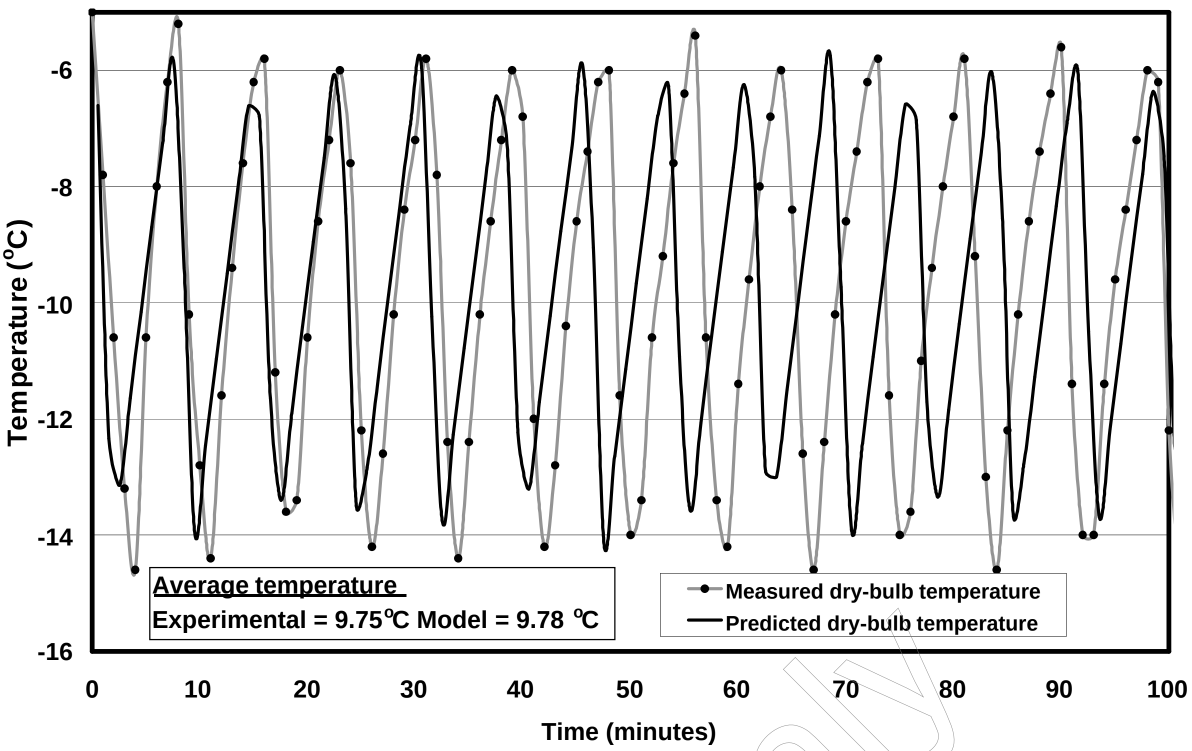

Figure 6: Comparison between predicted and measure dry-bulb air temperature before product loading (run time = 0 min to 100 min)

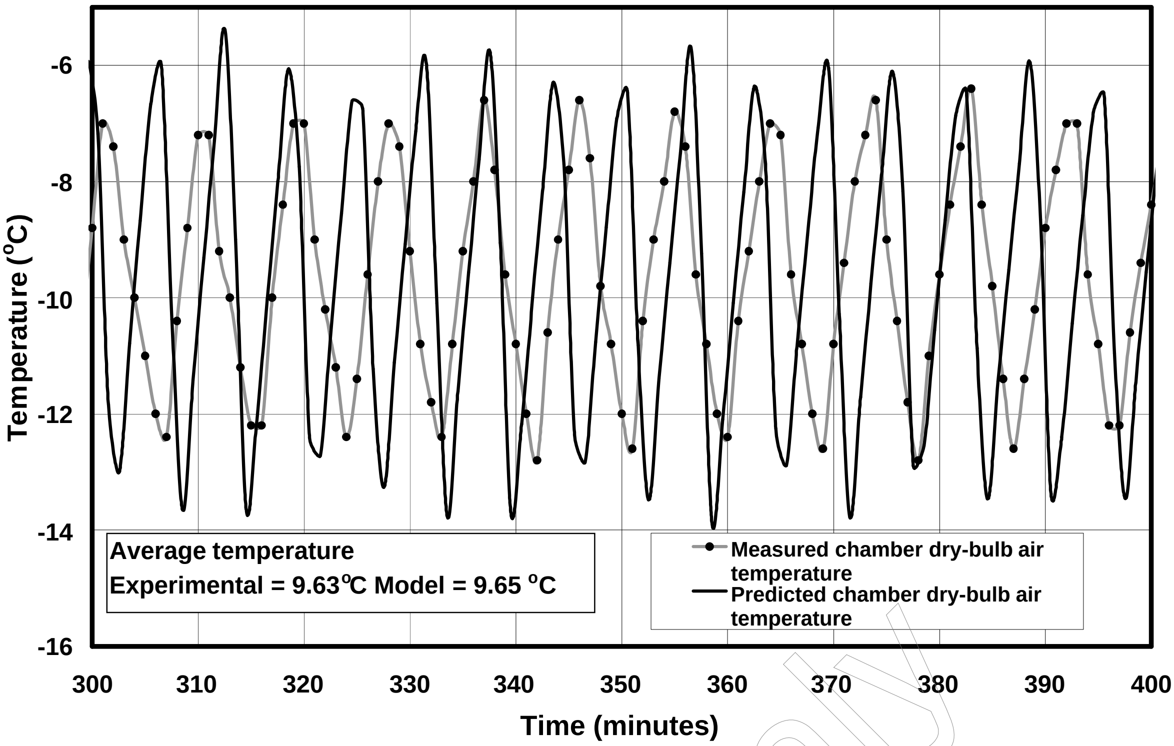

Figure 7: Comparison between predicted and measure chamber dry-bulb air temperature after product loading (run time = 300 min to 400 min)

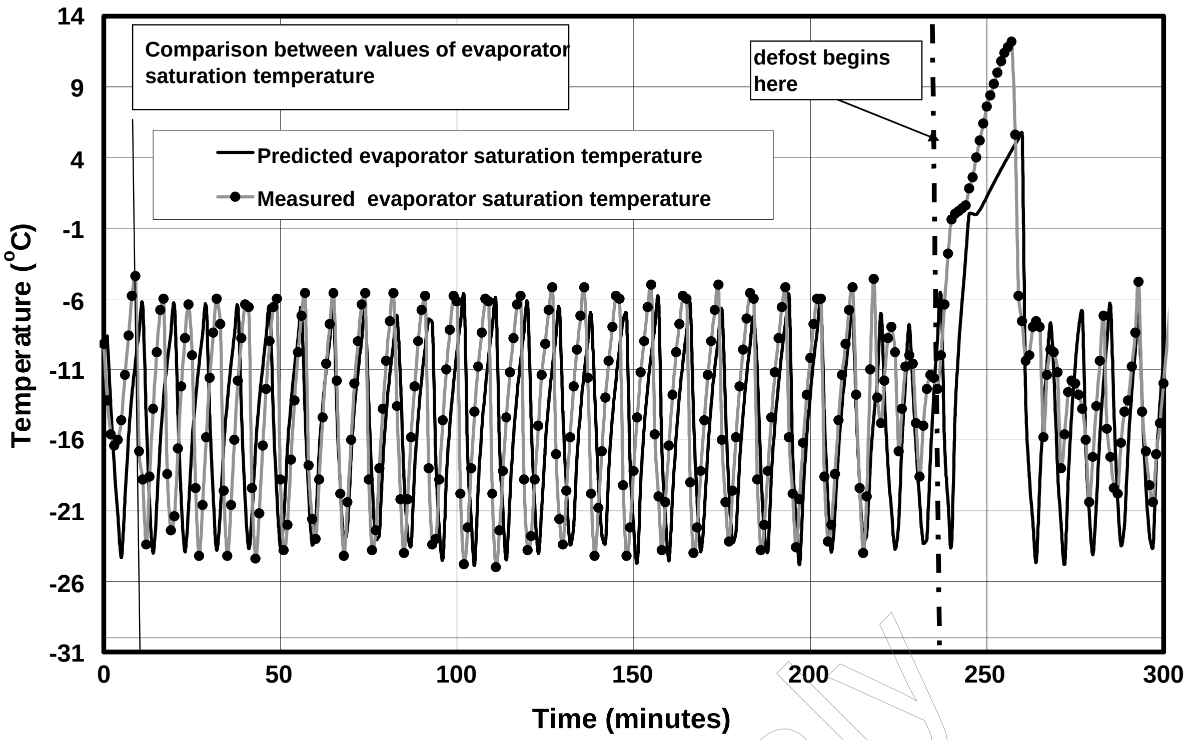

Figure 8: Comparison between measured and predicted evaporator saturation temperatures.

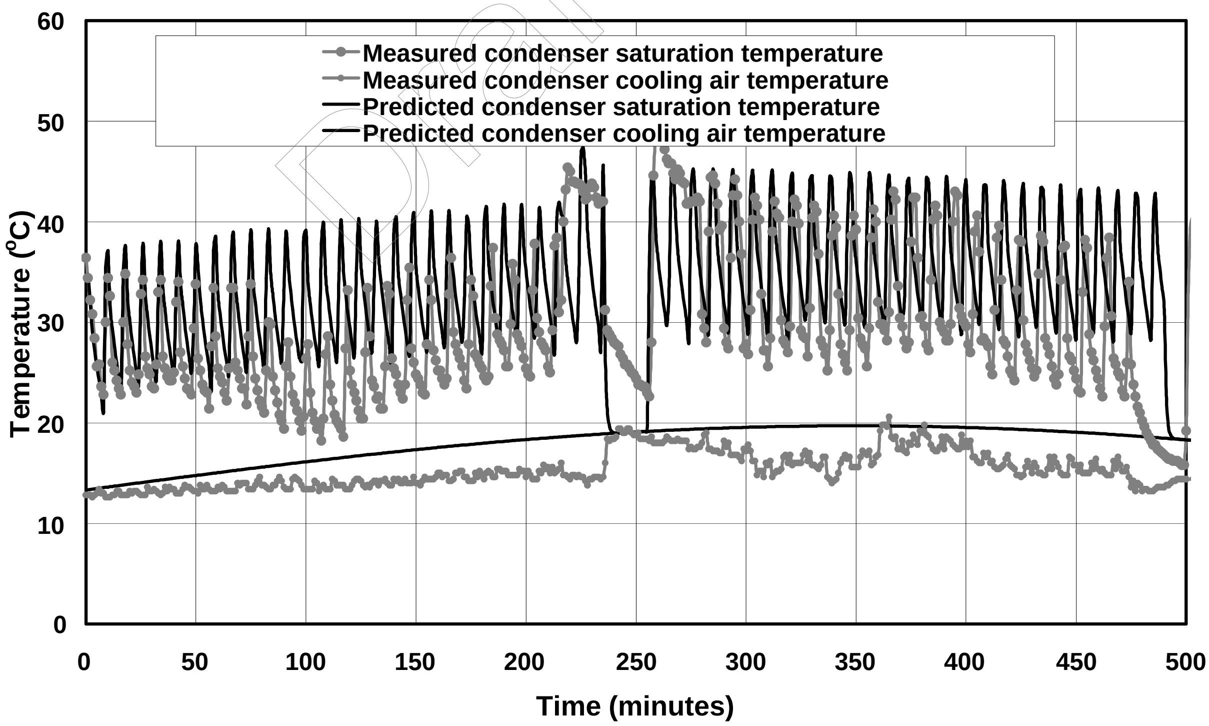

Figure 9: Comparing measured and predicted data for condenser saturation temperature and outdoor dry-bulb air temperature.

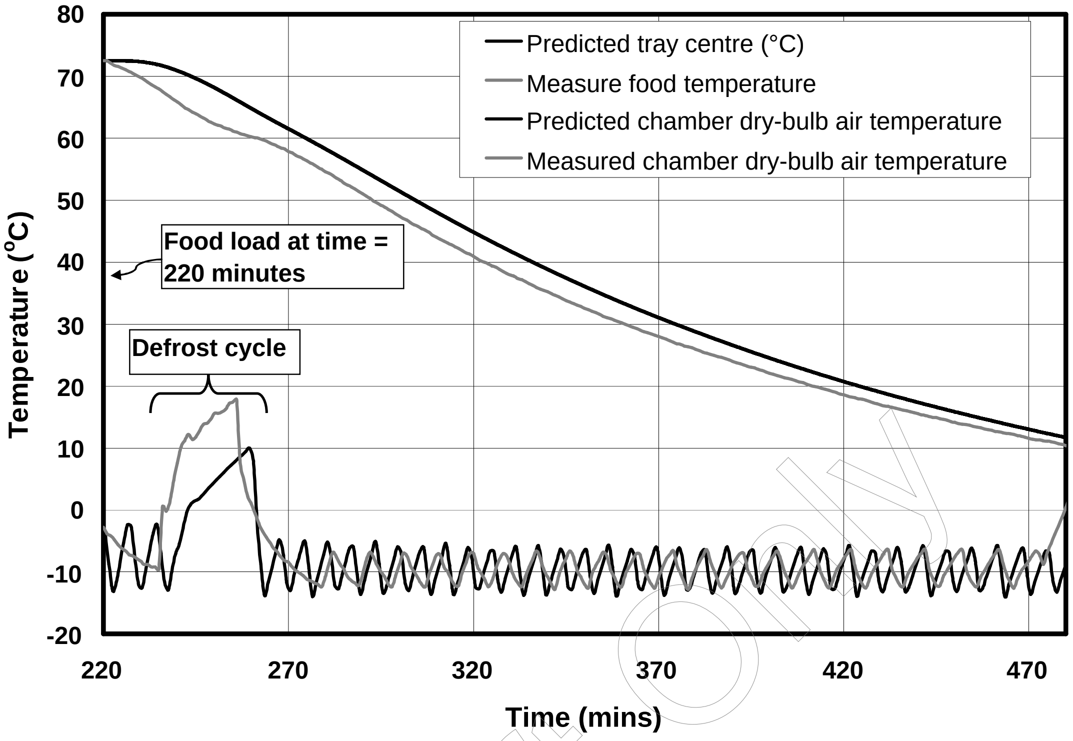

Figure 10: Comparison between measured and predicted food product temperatures and air-blast cooling cabinet dry-bulb air temperature.

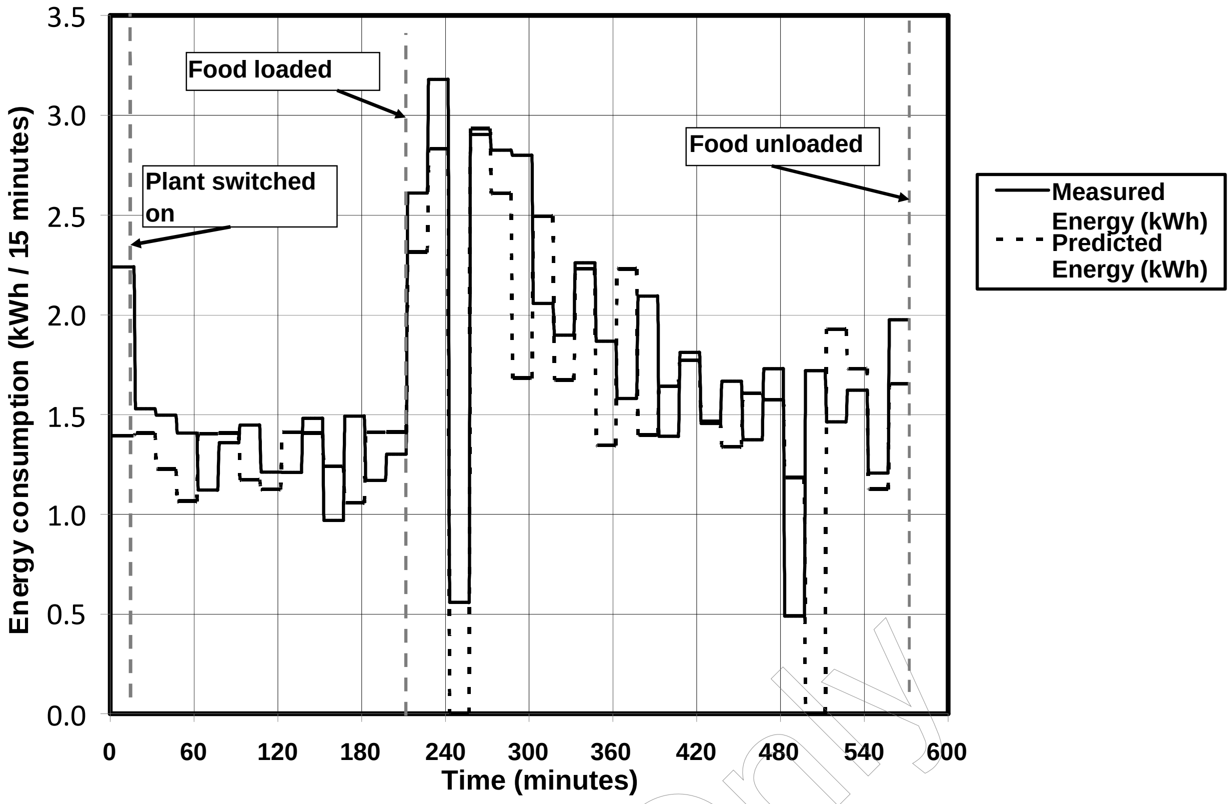

Figure 11: Measured and predicted energy consumption (average values over 15 minute intervals)