Cost Minimization for Hot Gas Defrost System

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Abstract

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Frost accumulation on evaporator coils has been a serious problem that decreases the efficiency of refrigeration systems. Many defrost methods have been used and hot gas defrost is the most common for industrial refrigeration applications. However, it is not a simple task to run an efficient hot gas defrost system. The duration and frequency of the defrost cycle should be properly determined so that the frost is melted but no additional heat transfer is transferred to the refrigerated space.

An experimental investigation was conducted at the Classic Salads Facility in Salinas, California. A single coil was instrumented to study frost build up and defrost during normal operation.

The finite difference method is used to numerically model the temperature of the coil during a defrost cycle. A separate model was developed to determine the pressure drop across evaporator coil with and without frost. Both models are used to determine the presence of frost on the coil tested at Classic Salads.

The pressure drop data did not show an increase in the pressure drop over time, indicating there was no frost accumulation during testing at Classic Salads. Also, comparison between the finite difference model and the experimental temperature data indicate that there was minimal frost. These results indicate that energy savings could be achieved if an accurate defrost termination was implemented.

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Table of Contents

List of Figures	viii
List of Tables	x
Introduction.....	1
Background.....	4
Frost	4
Hot Gas Defrost	5
Literature Review.....	8
Temperature Model.....	11
Pressure Drop Model	17
Testing.....	21
Collecting Data	22
Opto system	22
HOBO System	22
Time Constant.....	24
Results.....	28
Temperature	28
Pressure drop.....	34
Efficiency and Cost.....	38
Conclusions & Recommendations	42
References	45
Appendix A: LRC evaporator specifications	48
Appendix B: Opto22 system and ICTD temperature probe specifications.....	52

Appendix C: HOBO data logger specifications.....	55
Appendix D: Pressure transducer specifications.....	57
Appendix E: RTD sensor specifications	59
Appendix F: EES file for temperature model	61
Appendix G: EES file for pressure drop model	69
Appendix H: Weather data during experiment	71

List of Figures

Figure 1: Layout of Classic Salads facility	2
Figure 2: Refrigeration unit with hot gas defrost arrangement.....	5
Figure 3: Representation of a fin-tube section for a plate-fin heat exchanger.....	11
Figure 4: Schematic of the domain for fin-tube section	12
Figure 5: Isometric view with staggered arrangement.....	18
Figure 6: Friction factor f_{tube} for staggered tube bundle arrangement	19
Figure 7: The evaporator inside the Organic Room that was tested	21
Figure 8: HOBO data recorder for temperature sensors and pressure transducer	23
Figure 9: Temperature data to determine sensor time constant	25
Figure 10: Location of the RTD inserted between two fins.....	26
Figure 11: Locations of RTD on evaporator tube sheet.....	27
Figure 12: Temperature data from Opto22 and HOBO systems during the defrost cycle	28
Figure 13: Comparison between temperature data and model.....	30
Figure 14: Fin temperature during hot gas defrost with 1 mm of frost.....	31
Figure 15: Fin temperature during hot gas defrost with 0.5 mm of frost.....	32
Figure 16: Heat transfer during hot gas defrost with 0.5 mm of frost	33
Figure 17: Heat transfer during hot gas defrost with 1 mm of frost	33
Figure 18: Sample of coil pressure drop data	34
Figure 19: Pressure drop as a function of volumetric flow rate.....	35
Figure 20: Pressure drop as a function of volumetric flow rate for different frost thicknesses	36

Figure 21: Pressure drop as a function of volumetric flow rate for different frost thicknesses and different fan speeds	37
Figure 22: Hot gas defrost efficiency as a function of defrost time for different frost thickness	39
Figure 23: Hot gas defrost cost as a function of defrost time for different frost thickness.....	41

List of Tables

Table 1: Temperature probe specifications for HOBO and Opto systems	24
Table 2: Pressure transducer specifications	24
Table 3: Evaporator coil flow rate and pressure drop as a function of frost thickness.....	37

Nomenclature

Name	Description
A	Air-side surface area
A_c	Minimum free flow area
$A_{c,t}$	Minimum free flow area (for tube bank)
C_p	Specific heat
D	Diameter
f	Friction factor
g_m	Mass transfer coefficient
G_c	Maximum mass velocity
h	Specific enthalpy
h_c	Convection coefficient
k	Thermal conductivity
m	Mass fraction
P	Pressure
Q	Overall energy transfer
r	Radius
Re	Reynolds number
S_T	Transverse pitch
t	Time
t	Thickness
T	Temperature
V	Volume
x	Node spacing in the axial direction
z	Coordinate

Greek Symbol Δ

Difference

 ρ

Density

 η_{defrost}

Defrost efficiency

Subscript

amb

Ambient

 $\text{H}_2\text{O},\text{e}$

Water vapor in the conditioned space

 $\text{H}_2\text{O},\text{s}$

Water vapor at the surface of the frost

ig

Fusion

Introduction

VaCom Technologies is a consulting refrigeration controls company with headquarters in Laverne, CA. They also have an office in San Luis Obispo, CA. They design and install control systems to reduce energy consumption for industrial refrigeration facilities. One of their clients is Classic Salads located in Salinas, CA. The Classic Salads facility was recently retrofitted with new energy saving systems such as variable speed drives on the evaporator coils and floating head pressure control on the condenser. Nevertheless, VaCom Technologies is still interested in reducing energy expense by minimizing the defrost frequency and duration. Due to company workloads and the complexity of this problem, VaCom Technologies contacted Cal Poly for assistance.

The system at Classic Salads is a single stage R22 refrigeration system with four Carrier 5H80 reciprocating compressors and one Frick RWBII-134 screw compressor. There are a total of 26 evaporators, with groups of 2 or 3 in each zone, as shown in Figure 1. There are 6 auxiliary coils, 3 in the Finished Product room and 3 in the Raw Product room, while the rest are direct expansion coils. All of the evaporators are defrosted with hot gas refrigerant except the 6 auxiliary coils which use air to defrost.

Figure 1: Layout of Classic Salads facility

Each zone is scheduled to defrost for either 20 or 30 minutes, 4 or 6 times per day, depending on the location within the facility. For example, in the organic room, the product is brought in through an automatic door that is located next to the evaporator coils. The humidity from the outside tends to make the frost build up quicker in comparison to the other zones. Classic Salads schedules defrost based on experience to make sure that the system runs properly. If the coil ices up, then the evaporator must be shut down for an extended period of time to melt the ice.

Additional heat added to the coil, beyond what is required to melt the frost, and must be removed by the refrigeration system. This increases the refrigeration

load and also the cost of operating the system. The objective of this study is to determine when to initiate and/or terminate the defrost cycle. If the defrost time is too short then there will be some frost remaining on the coil when the normal operation resumes. Energy will be wasted if the defrost time is too long.

Due to limited time and resources, the experimental investigation is limited to the study of a single coil for a 1 week period. Results from the experiments will be composed to mathematical models of the coil.

Background

Refrigeration is the process of removing heat from a space or a substance and rejecting it to the surroundings. The main goal of this process is to lower the temperature of a space or a substance. Refrigeration has been used in wide a range of applications from air conditioning to food preservation. One of the key components of the refrigeration system is the evaporator. In food storage applications, the evaporator sometimes operates below freezing temperature ($<32^{\circ}\text{F}$) and frost can accumulate on the coil.

Frost

Frost accumulation will occur when the surfaces of the operating evaporative coil are below 32°F and the entering air dew point temperature is higher than the coil surface temperature [1]. There is a special case when moisture in the air condenses to liquid water and then freezes to ice. Although, normally the water vapor will transform directly into a solid phase, frost.

There are two major concerns when the frost formation occurs. First, when the frost layers grow, the air passages of the evaporator are narrowed. This will increase the resistance to the air flow. This frost layer will reduce the ability of the evaporator fan to move air through the coil and fan energy consumption will increase. Second, frost accumulation increases the resistance to heat transfer due to

the low conductivity of the frost layer [2]. Both of these factors reduce the effectiveness of the evaporator. Therefore, the frost must be removed periodically.

Hot Gas Defrost

There are many methods for defrosting, such as electric resistance heating, off-cycle (air defrost), reversed cycle, etc. For most industrial refrigeration applications, hot gas defrost is the most common method to remove frost from the evaporator. The basic concept behind hot gas defrost is to stop the cold refrigerant supply and then release high-pressure, high-temperature refrigerant from the compressor discharge to the evaporator. The evaporator will temporarily function as a condenser during the defrost process [3]. A schematic of the typical lay-out for a hot gas defrost system is shown in Figure 2.

Figure 2: Refrigeration unit with hot gas defrost arrangement

The processes of a typical hot gas defrost system are as follows:

Refrigeration (normal operation) phase:

- Saturated liquid refrigerant runs through the liquid feed valve into the evaporator
- Heat from the space is absorbed and the majority of the refrigerant is vaporized
- Saturated vapor from the evaporator outlet flows to the accumulator

Pump out phase (approximately 10 to 30 minutes):

- The liquid feed valve is closed
- The evaporator fans will continue to run in order for the liquid inside the coil to vaporize which will prevent pressure shocks damage
- At the end of this phase, the fans are turned off and the suction stop valve will be closed

Soft gas phase:

- For safety purposes, a solenoid valve is installed parallel to the hot gas valve to allow the hot gas to flow slowly into the coil
- The main hot gas valve is opened while the solenoid valve is closed at the end of this phase

Hot gas phase (approximately 5 to 30 minutes):

- The hot gas flows to warm the drain pan first before going into the coil
- The defrost regulator controls the pressure rise inside the coil
- The hot gas continues to flow until either:
 - 1) The hot gas pre-set time is reached or
 - 2) A sensor terminates this phase and closes the hot gas valve

Equalization phase:

- For safety purposes, an equalizing valve is installed parallel to the hot gas valve to slowly decrease the pressure inside the coil
- At the end of this phase, the equalizing valve is closed and the suction stop and liquid feed valves are opened.

Fan delay phase:

- The fans remain off allowing any remaining water to re-freeze to prevent any blow-over when the fans turn on again

Resume refrigeration phase:

- The fans are energized back to normal operation until the next defrost cycle is initiated

Literature Review

There are two ways for frost to build up on the evaporator coil. The first type is when tiny frost particles accumulate due to impaction on the evaporator coil surfaces. The tiny frost particles are created from the supersaturated air stream that suddenly cools. This type frost can build up very quickly due to its low density. It forms when the moisture content in the air is high, for example, evaporative coils near doors that open to the outside.

The second mechanism is through the diffusion of water vapor onto the cold surface due to the difference in the concentration of water between the air stream and the frost layer. This forms a layer of frost with high density. Due to the dense structure, this type of frost must be removed periodically. This high density frost occurs in places where the air temperature is low, along with low moisture content. This type of frost forms on evaporator coils located inside a refrigerated warehouse where food products are stored for a long period of time [1]. It is important to periodically remove the frost on the coil surface regardless of the frost formation mechanism, so that the evaporator can continue to operate normally.

There are many research studies for the prediction of frost growth. Nevertheless, the frost formation is still an open research topic because there are many different types of models, different assumptions and different experimental conditions. Most of the frost prediction models can be categorized into three main

groups [4]. The first group uses molecular diffusion applied to the frost layer. Then, empirical correlations on the air side are used to calculate heat and mass transfer to predict the frost growth characteristics [5]. The second group applies an improved model to predict the frost properties by using the empirical correlations from the air flow boundary layer equations [6]. The third group uses molecular diffusion in the frost layer combined with the boundary layer equations to analyze the frost formation [7].

There are some studies that do not follow the previous mentioned groups. Some of these studies focus on the analysis of simple geometries that resemble a portion of a heat exchanger, such as a laminar flow analysis over parallel cold plates [8], a frost formation study on a vertical plate [9], and an experimental analysis on a cold cylindrical surface [10]. Some researchers have worked to develop realistic and complex frost growth models. For example, Tso et al. created a more comprehensive model by deriving equations for non-steady-state and quasi-steady state heat transfer through a tube-fin heat exchanger [11]. Lenic et al. developed a transient two-dimensional mathematical model of frost formation to predict the frost growth rate and a change in the thermal conductivity of the frost layer [12].

Although there are many frost growth models, the majority of them do not focus on when to initiate or terminate the hot gas defrost cycle. Hoffenbecker et al developed a model of the hot gas defrost process [13]. For practical applications, this

method is easier to implement than other models and it still can produce reasonable results, which is why Hoffenbecker's model is chosen for this project.

Many models have been developed and experiments have been conducted to study how to detect the frost build up and how the frost accumulation affects the system performance. The pressure drop across the coil has been used numerous times to predict frost growth, especially in heat pump applications. For example, Yao et al. developed a distributed mathematical model of the airside heat exchanger under frost conditions in an air source heat pump unit [14]. Liu et al. created a transient one dimensional model to predict frost growth and validated it experimentally for a heat pump unit [15]. Both researchers show that pressure drop increases dramatically once the frost accumulates. As a result, pressure drop is investigated as a frost detection method in this application as well.

Not many studies focus on cost estimation of the hot gas process. Cole is one of the few researchers who has developed models to analyze the cost of the hot gas defrost process [16].

Temperature Model

In industrial refrigeration systems, air-cooled evaporators are constructed of multiple rows of tubes along with fins that are arranged perpendicular to the tubes. The fin-tube evaporator can be modeled as a hexagon shape due to the symmetry of the geometry as shown in Figure 3a. This can be further approximated as a circular disc as shown in Figure 3b.

Figure 3: Representation of a fin-tube section for a plate-fin heat exchanger (a) and approximation as a circular disc (b) [13]

During the hot gas defrost, the condensing high pressure gas from either the high pressure receiver or the compressor discharge will warm the inside tube surface along with the fins that attach to the tubes. Thermal energy is conducted through the

fin to melt the frost. Due to gravity, the melted frost will fall down from the coil to the drain pan. Some of the frost will also evaporate to the surrounding air.

Figure 4: Schematic of the domain for fin-tube section (a) [13] and two-dimensional finite difference thermal model (b)

During the hot gas process, there are many important heat and mass transfer mechanisms that occur: condensation of high-pressure and high-temperature gaseous refrigerant inside the coil tubes; conduction through the coil tubes and fins; sensible heating of and melting of accumulated frost; and evaporation of moisture from the surface coil to the surrounding space [13].

A thermal model of the geometry shown in Figure 3 was developed to study the hot gas defrost process. The fin-tube was modeled as a two-dimensional, transient conduction problem. The heat equation for two-dimensional, axisymmetric conduction is

$$\rho C_p \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(kr \frac{\partial T}{\partial r} \right) + \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z} \right). \quad (1)$$

The geometry is sub-divided into smaller regions, as shown in Figure 4, and the heat transfer problem is solved using the finite difference method. The boundary conditions for a control volume about each node are:

- Boundary a: thermal conduction from the adjoining nodes at the inner radius.
- Boundary b: conduction/convection in the axial direction at the left side
- Boundary c: thermal conduction to the adjoining nodes at the outer radius
- Boundary d: thermal conduction in the axial direction at the right side

An adiabatic boundary condition is applied at the outer radius and the right side of the model as shown in Figure 4b. The boundary condition at the inside tube wall is assumed to be constant temperature due to the condensing refrigerant. The boundary condition on the left side is convective heat and mass transfer to the surrounding air.

The two dimensional axisymmetric, partial differential equation is solved using a finite difference approach. The difference equation for node T₁ is

$$\rho V \left(\frac{dh}{dt} \right) = k_{fin} 2\pi \frac{\Delta x}{2} \frac{(T_c - T_1)}{\ln(r_1 / r_c)} - k_{fin} 2\pi \frac{\Delta x}{2} \frac{(T_1 - T_2)}{\ln(r_2 / r_1)} - \left[2\pi \left(\left(\frac{r_2 + r_1}{2} \right)^2 - \left(\frac{r_1 + r_c}{2} \right)^2 \right) \frac{(k_{fin} + k_{frost})}{(k_{frost} t_{fin} + 2 k_{fin} t_{frost})} (T_1 - T_{frost1}) \right]. \quad (2)$$

The first term on the left hand side of equation (2) represents the change in energy stored in the node. This can be simplified by changing $dh = C_p dT$ and assuming that C_p is constant. The first and second terms on the right hand side of equation (2) represent thermal conduction in the radial direction. The third term on the right hand side represents the thermal conduction between adjacent nodes from the fin to the frost at the same radial location. The energy balance at different radial locations along the fin will have a similar set up.

The energy balance equation for frost at node T_{f1} (see Figure 4b) can be expressed as

$$\begin{aligned} \rho V \left(\frac{dh}{dt} \right) = & k_{frost} 2\pi \frac{\Delta x (T_c - T_{frost_1})}{2 \ln(r_1 / r_c)} - h_c \pi \left(\left(\frac{r_2 + r_1}{2} \right)^2 - \left(\frac{r_1 + r_c}{2} \right)^2 \right) (T_{frost_1} - T_{amb}) - \\ & k_{frost} 2\pi \frac{\Delta x (T_{frost_1} - T_{frost_2})}{2 \ln(r_2 / r_1)} - \dot{Q}_{evap} + \\ & \left[2\pi \left(\left(\frac{r_2 + r_1}{2} \right)^2 - \left(\frac{r_1 + r_c}{2} \right)^2 \right) \frac{(k_{fin} + k_{frost})}{(k_{frost} t_{fin} + 2k_{fin} t_{frost})} (T_1 - T_{frost_1}) \right], \end{aligned} \quad (3)$$

where

$$\left[\dot{Q}_{evap} = g_m \pi \left(\frac{r_2 + r_1}{2} \right)^2 - \left(\frac{r_1 + r_c}{2} \right)^2 (m_{H_2O,s} - m_{H_2O,e}) h_{ig} \right]. \quad (4)$$

Equation 4 is the latent energy transfer rate due to evaporation, where $m_{H_2O,s}$ represents the mass fraction of water vapor at the surface of the frost and $m_{H_2O,e}$ represents the mass fraction of water vapor in the surrounding air. Both mass fraction values can be determined from mass transfer book by Mills [17]. The energy balance at different radial locations in the frost will be similar. However, this time the energy storage term cannot be simplified any further since the frost control volume undergoes a phase change from ice to liquid during the defrost process. The discrete equations are solved using the Engineering Equation Solver, or EES [18].

One assumption for this model is that the thermal resistance of frost remains constant over time. However, when the frost melts, it is replaced by water or a layer

of air, which cause a change in the thermal resistance. It is difficult to model the change in thermal resistance under these conditions because of the uncertainty in determining the substance/phase within the layer as a function of time. This approach was taken in order to develop a reasonably accurate yet somewhat simple model.

Pressure Drop Model

As frost accumulates on the evaporator, the air side pressure drop will increase. As a result, the heat transfer from the system will decrease due to the lower air flow rate. It was postulated that the presence of frost could be detected by measuring the pressure drop across the coil. Therefore, a model was developed to predict the air-side pressure drop with and without frost.

The pressure drop across the coil is determined by combining the pressure drop due to the tubes and the pressure drop due to the fins

$$\Delta P_{\text{coil}} = \Delta P_{\text{fin}} + \Delta P_{\text{tube}} \quad (5)$$

or

$$\Delta P_{\text{coil}} = f_{\text{fin}} \frac{A_{\text{fin}}}{A_c} \frac{G_c^2}{2\rho} + f_{\text{tube}} \frac{A_{\text{tube}}}{A_{c,t}} \frac{G_c^2}{2\rho} \quad (6)$$

Where G_c is the maximum mass velocity, A_c is the minimum flow area, $A_{c,t}$ is minimum flow area for the tube bank only, and A_{fin} and A_{tube} are the surface areas for the fins and tubes respectively.

Figure 5: Isometric view with staggered arrangement (a), front view that parallel the air-flow (b), side view which perpendicular to the air-flow (c) of the evaporator

This model neglects the pressure losses at the inlet and outlet of the evaporator. The friction factor for the fin is approximated by an empirical correlation equation [19]:

$$f_{fin} = 0.508 \text{Re}_D^{-0.521} \left(\frac{S_T}{D} \right)^{1.138} \quad (7)$$

where

$$\text{Re}_D = \frac{G_c D}{\mu}, \quad (8)$$

S_T is defined as the transverse pitch or tube spacing normal to the air flow (shown in Figure 5c), and μ is defined as the viscosity of air. The Reynolds number is based on the tube outer diameter and the maximum fluid velocity occurring within the tube bank. The maximum velocity is defined as

$$V_{\max} = \frac{S_T (\text{fin pitch} - \text{fin thickness})}{(S_T - D_{\text{tube}})(\text{fin pitch} - \text{fin thickness})} V. \quad (9)$$

Some researchers use the friction factor for the tube based on the Zukauskas correlation for both normal and staggered banks of tubes. For this pressure drop model, the friction factor for the tube can be approximate from Figure 6 [20],

Figure 6: Friction factor f_{tube} for staggered tube bundle arrangement [20]

where P_T is defined as S_T/D which is the dimensionless transverse pitch.

The pressure drop for the coil with frost is modeled by assuming that the frost layer simply increases the fin thickness and the tube outer diameter. The frost layer is assumed to have uniform thickness and the increased roughness of the frost layer is neglected.

Testing

Figure 7: The evaporator inside the Organic Room that was tested

As mentioned previously, the evaporator coils in the organic room tend to accumulate more frost than the coils in the other rooms. This is because the evaporators are located near the automatic door that is frequently opened to bring product into the facility. The moisture from the outside air makes it more likely that frost will accumulate on the evaporators in this space. Therefore, one of the evaporators in the Organic room, shown in Figure 7, was chosen for testing.

Collecting Data

Opto system

The Opto22 system is used as the main control system for the Classic Salads facility [21]. The following is the list of sensors connected to the Opto22 system:

- Zone temperature - air temperature measured at the inlet of the coil
- Zone suction temperature – refrigerant temperature measured at the outlet of the coil
- AU VFD SR – fan speed
- Zone coil temperature – temperature measured at the tube sheet on evaporator
- Discharge pressure – main system discharge pressure at the outlet of the condenser

Two wire RTD are used to measure temperature for the Opto22 system.

HOB0 System

The coil was instrumented with additional sensors and the output from these sensors was recorded with a HOB0 U12-006 model data logger, which is shown in Figure 8. The HOB0 data logger has 4 channels and can record up to 43,000 measurements [22]. The coil was instrumented with 3 temperature sensors and 1 pressure transducer. Data was logged from November 9, 2008 at 8 am to November 15, 2008 at 7 pm. The temperature sensors were two wire RTD. Each RTD utilized a

separate circuit box to produce 4-20 mA current signal. The specifications for the temperature sensors and the pressure transducer are shown in Table 1 and Table 2, respectively.

Figure 8: HOBOT data recorder for temperature sensors and pressure transducer

The pressure transducer range is unidirectional from 0 to 0.5 InchWg. The output is the same as temperature sensor's output, which is 4-20 mA. The high pressure port was exposed to the ambient air while the lower pressure port was connected to a tube with the exposed end placed between the coil and the fan. This arrangement effectively measures the static pressure difference across the coil.

Table 1: Temperature probe specifications for HOBO and Opto systems

	HOBO	Opto
Input Temperature Range	- 30 °C to 130 °F	- 40 °C to 100 °C
Output	4 mA @ -30 °F 20 mA @ 130 °F	233 μ A @ -40 °C 373 μ A @ 100 °C
Input Response Time	63.2 % at 95 seconds	63.2 % at 150 seconds
Output Accuracy	± 0.3 °C @ 0 °C	± 0.5 °C @ 25 °C
Thermal Time Constant	1.58 minutes	2.5 minutes typical (still air)
Dimensions	Cylinder tip: 1" (2.54 cm) L, 0.25" (0.635 cm) D	Cylinder tip: 0.70" (1.78 cm) L, 0.375" (0.95 cm) D

Table 2: Pressure transducer specifications

Model: 264 0R5WD	
Range	0 to 0.5 InchWg
Output	4 - 20 mA
Circuit	2 wires
Operating Temperature	0 to 175 °F

All the data from the Opto22 system is recorded every 2.3 seconds. As for the HOBO system, all the temperature and pressure drop data were recorded at one minute intervals due to the limited storage capability of the logger.

Time Constant

The temperature sensors used by the HOBO data logger and the Opto22 system have different diameter size and consequently the response time will be different. There was no available information about thermal time constant for the

RTD used by the HOBO data logger. Therefore, an experiment was conducted to determine the response time.

The experiment was conducted in the thermal science lab at Cal Poly. A computer data acquisition system using Labview was used to record the data. The RTD was placed inside a freezer until the temperature reading was constant. The RTD was then removed from the freezer and held in air. The resulting temperature data is shown in Figure 9.

Figure 9: Temperature data to determine sensor time constant

The time constant is the time it takes the sensor to achieve 63.2 % of the temperature change or $0.632 \times (73 + 2) ^\circ\text{F}$, which is 47.4 $^\circ\text{F}$ for this experiment. From

Figure 9, the time constant is determined to be approximately 95 seconds. The first 53 seconds is when the RTD is still inside the refrigerator at a temperature of approximately -2°F . After that, the RTD is taken out from the refrigerator to the ambient air, which is at approximately 73°F . The time to reach steady state is approximately 600 seconds. This is approximately 37% faster than the response time of the temperature sensor used by the Opto22 system (see Table 1).

Figure 10: Location of the RTD inserted between two fins

Figure 11: Locations of RTD on evaporator tube sheet

Results

Temperature

Figure 12: Temperature data from Opto22 and HOBO systems during the defrost cycle

A sample of the measured temperatures during a defrost cycle is shown in Figure 12. From the HOBO system, Fin temp represents the temperature reading between two fins, as shown in Figure 10. Tube sheet temp 1 and Tube sheet temp 2 are the temperature sensors attached at the middle and at the bottom of the tube sheet, as shown in Figure 11. From the Opto22 system, the inlet air temperature refers to the air temperature at the inlet to the coil. The outlet temperature is the

sensor attached to the suction line of the coil. The termination temperature is attached on the tube sheet which is shown in Figure 11.

Note that none of the temperature readings are below 32 °F which indicates that there might not be much frost accumulated. This result is representative for data collected during the entire week. The temperature readings from the tube sheet seem to be more stable in comparison to the reading for the sensor between the fins. This may be due to the insulation that covering around the temperature sensors on the tube sheet, while the temperature sensor attached between the fins is exposed to the surrounding air.

The termination temperature sensor (Opto22 system) attached to the tube sheet responded slowly compared to the HOBO temperature sensors that were also attached to the tube sheet. The Opto22 sensor took more than 10 minutes to reach steady state while it took about 5 minutes for the HOBO sensors. Also, the Opto22 temperature measurement is approximately 5 degrees lower in comparison to the measurements made by the HOBO sensors. The reason for these different values is unknown.

The temperature sensor attached between the fins appeared to have a quick response time compared to the other sensors. This indicates that temperature sensor between the fins should yield a more accurate representation of the temperature of

the coil surface where the frost accumulates. The experimental temperature data was compared to the results from the model described previously.

The time has been adjusted between the two, so the temperature model and the experimental data begin at the same time when the defrost cycle has been initiated. The temperature sensor has a finite response time and so the slope is not as steep in comparison to the thermal model. Once that consideration has been taken into account, there appears to be good agreement between the data and the results from the model for a frost thickness layer of 0.25 mm or less.

Figure 13: Comparison between temperature data and model

Figure 13 shows measured data for the temperature of the fin (Fin temp), along with results from the finite difference model for frost layers with different thicknesses. The result for node 5 of the model (midway between tubes, see Figure 4b) was used for the comparison. For all the nodes, the initial temperature starts at the freezing point of water but experimentally none of the fin's temperature readings reached 32 °F. This was done so the accumulation of frost could be modeled.

Figure 14: Fin temperature during hot gas defrost with 1 mm of frost

Figure 15: Fin temperature during hot gas defrost with 0.5 mm of frost

Figures 14 and 15 show the results of the nodal temperatures during a defrost simulation. The initial temperature for all nodes is 32 F. Figure 14 is for a frost layer of 1 mm and Figure 15 is for a frost layer of 0.5 mm. The subscript for each temperature is defined in Figure 4b with the inner radius being 1 and the other radius being 5. It takes approximately 250 seconds the temperatures to reach steady state with 0.5 mm of frost, while it takes about 400 seconds for the temperature to reach steady state with 1 mm of frost.

Figure 16: Heat transfer during hot gas defrost with 0.5 mm of frost

Figure 17: Heat transfer during hot gas defrost with 1 mm of frost

Figures 16 and 17 show the resulting heat rates during a defrost simulation.

$Q_{in, total}$ is defined as the total heat going into system from the hot gas. $Q_{conv, total}$ is the heat lost due to convection and Q_{melt} is the heat required to melt the frost. The figures show the total energy integrated over time. The time to melt the frost can be observed from the figures by noting when Q_{melt} becomes constant. After this point, heat from the hot gas is lost due to convection.

Pressure drop

Figure 18: Sample of coil pressure drop data

Figure 18 shows the measured pressure drop as a function of time during normal operation. The mean value, of the pressure for the period shown in Figure 18 is 0.403 InchWg. The standard deviation of the readings is 0.019 InchWg. The data is a representative of the data collected for the entire week.

There is significant scatter in the data, which may indicate the pressure tap was not rigidly fixed. Regardless, there is no general trend of increasing pressure drop with time, which would indicate the buildup of frost on the coil.

Figure 19: Pressure drop as a function of volumetric flow rate

Figure 19 shows system curves (different sources) and fan curve for the evaporator. The fan curve is based on data from the manufacturer (see Appendix A

for more information). The “manufacturer” system curve is extrapolated from a single data point assuming $\Delta P \propto Q^2$. The pressure drop as determined by the model is also shown over the range of flow rates.

Figure 19 shows reasonably good agreement between the manufacturer’s data and the pressure drop model. The measured pressure drop exceeds the pressure drop from the manufacturer’s data and the model. This may be due to foreign particles, such as cardboard fibers, which have buildup on the coil surface.

Figure 20: Pressure drop as a function of volumetric flow rate for different frost thicknesses

Table 3: Evaporator coil flow rate and pressure drop as a function of frost thickness

Evaporator's operating point		
Frost (mm)	Q(cfm)	ΔP (InchWg)
1	1,740	0.408
0.75	1,930	0.378
0.5	2,090	0.343
0.25	2,220	0.315
0	2,330	0.282

Figure 20 and Table 3 show the results from the pressure drop model for varying frost thicknesses. The results show that the volumetric flow rate decreases and the pressure drop increases as there is more frost accumulation on the coil. Table 3 shows that the pressure drop increases by approximately 0.03 InchWg for each increment of 0.25 mm of frost.

Figure 21: Pressure drop as a function of volumetric flow rate for different frost thicknesses and different fan speeds

Figure 21 shows the same set of system curves from Figure 20 along with a set of fan curves at speeds varying from 70% to 100% of full speed (1050 rpm). The pressure drop will increase from 0.282 InchWg (dry coil) to 0.408 InchWg with 1 mm of frost when the system is operated at 100% fan speed. If the system operates at 70% of the fan speed (735 rpm), then the pressure drop will increase from 0.147 InchWg (dry coil) to 0.205 InchWg with 1 mm of frost. This shows that as the fan speed decreases the difference in the pressure drop will also decrease. Therefore, it will be harder to detect the amount of frost on the coil using a pressure sensor when the system operates at lower fan speed.

Efficiency and Cost

The defrost efficiency is defined as

$$\eta_{\text{defrost}} = \frac{Q_{\text{melt}}}{Q_{\text{total}}} \quad (10)$$

Where Q_{melt} is total heat required to warm and melt the frost on the coil and Q_{total} is the total heat transfer to the coil during the defrost. A defrost efficiency closer to 1 indicates that a majority of the heat is used to melt the frost.

Figure 22: Hot gas defrost efficiency as a function of defrost time for different frost thickness

Figure 22 shows the defrost efficiency as a function of time for varying frost thicknesses. The results were obtained from the thermal model. Figure 22 shows that the efficiency of the hot gas defrost decreases with increasing time. Also, the efficiency of the defrost decreases as the frost thickness decreases. These results make sense because initially most of the energy is used to melt the frost. After that, heat is transferred to the space by convection. A thinner frost layer will defrost more rapidly and the defrost efficiency will be lower for a fixed defrost time. As mentioned previously, the thermal resistance of the frost layer was not reduced as the frost melted. Therefore, the efficiency of the real system will be lower than what is shown in Figure 22.

The work input for a defrost system is defined as

$$W_{\text{cycle}} = \frac{Q_{\text{in}}}{\beta}, \quad (11)$$

where Q_{in} is the excess heat generated from the hot gas after the frost has melted and β is the coefficient of performance of the refrigeration system. Q_{in} can be expressed as: $Q_{\text{total}} - Q_{\text{melt}}$. After substituting $Q_{\text{melt}} = \eta_{\text{defrost}} Q_{\text{total}}$ from equation 11, the work input can now be expressed as

$$W_{\text{cycle}} = \frac{Q_{\text{total}}(1 - \eta_{\text{defrost}})}{\beta}. \quad (12)$$

Therefore, the cost for a defrost cycle is defined as:

$$\text{Cost} = \frac{\$}{\text{kWh}} \left(\frac{Q_{\text{total}}(1 - \eta_{\text{defrost}})}{\beta} \right). \quad (13)$$

For sample calculation, the coefficient of performance is assumed to be 3 and the electricity cost is assumed to be \$ 0.12 per kWh, based on 2009 Energy Information Administration data [23].

Figure 23: Hot gas defrost cost as a function of defrost time for different frost thickness

Figure 23 shows the cost of operation will consequently increase as the longer the system has to defrost. For thicker frost layers, all the energy from the hot gas is used to melt the frost and so the cost is low initially. Thinner frost layers are melted very quickly and the remaining heat goes to heat up the space instead. For 1 mm of frost, it costs about \$ 0.08 for 20 minutes of defrost. If the system is terminated 10 minutes sooner, it will cost only \$ 0.035, which will save \$ 0.045 per defrost cycle.

Conclusions & Recommendations

Both the results from the measurements and the modeling indicate that there was no frost build up during the testing at the Classic Salads facility. Although there was scatter in the pressure measurements, there was no overall increase in the pressure measurement with time, which would indicate the presence of frost on the coil. The finite difference model with little or no frost was in good agreement with the measured temperatures during a defrost cycle.

The Opto 22 temperature measurement on the tube sheet was significantly different than the HOBO measurements on the fin and the tube sheet. First, the Opto22 sensor had a longer lag time. Second, at steady state, there was about a 5°F temperature difference between the measurements using the two systems. It may be possible to use a temperature measurement to terminate the defrost cycle. A sensor located on the fins should give a more accurate measure of the temperature of the coil surface and whether frost is present. However, the lag in the Opto22 sensor (due to the larger size) would make this measurement more conservative.

There was significant scatter in the pressure measurements, much larger than the accuracy of the transducer. It is believed that the exposed end of the tube between the fan and the coil was not properly attached and consequently was moving due to the turbulence within the air stream. Also, the measured pressure drop across

the coil was higher than that measured by the manufacturer and predicted by the model. The reason for this could be due to cardboard fiber accumulation on the surfaces of the coil.

The purpose of the pressure measurement was to determine whether the defrost cycle could be initiated based on a measured increase in the pressure drop across the coil, indicating the presence of frost. This is a common technique in heat pump applications. Unfortunately, the experimental results could not prove the usefulness of this method since there was little or no frost during the testing.

The pressure drop model was used to determine the change in the coil pressure drop as a function of frost thickness and fan speed. The changes in the pressure drop are much greater than the repeatability of the transducer, indicating that this method could prove useful. Also, the fans do not have to be maintained at fixed speed in order to accurately detect the presence of frost since this can be modeled.

The results from the testing indicate that the defrost cycle could be terminated sooner. If the defrost cycle could be terminated 10 minutes sooner during a regularly scheduled 20 minute defrost cycle, the savings per defrost are estimated to be \$0.045 per coil. Extending this to the entire facility, and assuming the Classic Salads facility is operated for 9 months during the year and each of the 26 evaporator

coils are scheduled to defrost 6 times per day for 20 minutes, the annual savings would be approximately \$1,900 per year.

Many things can be done to improve the experimental testing. First, it is necessary to repeat the testing under conditions where frost will form. Second, photographic evidence would be very helpful in determining the presence of frost. It should also be possible to use this to determine the thickness of the frost layer. It is also recommended that the testing be conducted at a location on campus. This would facilitate easy access to the testing facility and the ability to run multiple tests under different, and hopefully more controllable, conditions.

Also, the models need to be verified with experimental testing. While this investigation was an important first step, both the thermal and pressure drop models need to be tested against measurements on a coil, under frosting conditions, with a known layer of frost.

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FIN DIMENSIONS

Fin Length of unit - 108"

expected static pressure drop on

coil $\Rightarrow$.297 in H_2O

Motor rpm - 1040

Evaporator Coil

Model

LRC (5/8" tube)
610 x 2743 6r 4f 10c

Air

Barometer	29.891	InchHg
Altitude	-0.114	ft
Airflow	10800	cfm
Velocity	600	fpm
On Coil	38.0/31.9	°F
Off Coil	32.0/28.4	°F
Cond Rate	0	g/s
Pressure drop	0.297	InchWg

Refrigerant (Evaporating R22)

Temperature	25.0	°F
Abs. Press	63.3	Psi
Density	80.5	lb/ft ³
Enthalpy Change	71.5	Btu/lb
Mass Flow	1040	lb/hr
dP Rate	0.00532	Psi/ft
Press Drop	0.561	Psi
Pipe Conn	1.38	inch

Performance

Sensible Duty	74354	Btu/hr
Total Duty	74354	Btu/hr

DXC 1.2.1.0 registered to LRC Coil Co., Mike Williams

Appendix B: Opto22 system and ICTD temperature probe specifications

OPTO 22

DATA SHEET

Form 446-050803

I/O MODULES STANDARD ANALOG INPUT

page 1/2

Description

The AD4 module provides a single channel of optically-isolated temperature-to-digital conversion. Modules plug into any Opto 22 standard analog I/O rack and are secured by a captive screw. Field connections to the module are made via two terminals on the analog I/O rack.

The AD4 module is designed for use with an ICTD probe from Opto 22 or from other vendors. The ICTD probe is encapsulated and completely isolated from ground loop possibilities. The ICTD can be used in energy management, freezer control, and similar applications.

Part Number	Description
AD4	ICTD Temperature Input
ICTD	ICTD Temperature Probe

Specifications

AD4

Input Temperature Range	- 188.4 °C to 150 °C
Input Response Time	5% full scale step change in 7.8 seconds 63% full scale step change in 150 seconds
Accuracy of AD4 Accuracy with ICTD Probe	± 0.3 °C ± 0.8 °C
Resolution with ICTD Probe	0.083 °C
Power Requirements:	16 mA at +15 VDC 11 mA at -15 VDC 1.6 mA at 5 VDC
Thermal Time Constant	2.5 minutes typical (still air)
Cable Length	> 2,000 feet (610 meters)
Isolation Input-to-Output	4,000 Vrms
Ambient Temperature Operating Storage	0 °C to 70 °C -25 °C to 85 °C

OPTO 22**DATA SHEET**

Form 446-060803

**I/O MODULES
STANDARD
ANALOG INPUT**

page 2/2

Specifications (continued)**ICTD**

Input Temperature Range	- 40 °C to 100 °C
Output	233 μ A @ -40 °C 373 μ A @ 100 °C
Sensitivity	1 μ A/°K
Input Response Time (% of Span/ Δ °C/ Δ Time)	5 %/7 °C/7.8 seconds 20 %/28 °C/33.6 seconds 63.2 %/88.48 °C/150 seconds
Output Accuracy	± 0.5 °C @ 25 °C
Repeatability	± 0.25 °C
Thermal Time Constant	2.5 minutes typical (still air)
Cable Length	>2,000 ft (610 meters)
Dimensions	Overall length: 39" (1 m) Cylinder tip: 0.70" (1.78 cm) L, 0.375" (0.95 cm) D

Connections**AD4 CONNECTION DIAGRAM**

Opto 22 • 43044 Business Park Drive • Temecula, CA 92590-3614 • Phone: (951) 695-3000 • (800) 321-OPTO • Fax: (951) 695-3095 • www.opto22.com

Inside Sales: (800) 321-OPTO • Product Support: (800) TEK-OPTO • (951) 695-3080 • Fax: (951) 695-3017 • E-mail: sales@opto22.com

Dimension and specifications are subject to change. All products and/or company names throughout this data sheet are generally trademarks or registered trademarks of their respective companies.

ICTD

Input Temperature Range	- 40 °C to 100 °C
Output	233 μ A @ -40 °C 373 μ A @ 100 °C
Sensitivity	1 μ A/°K
Input Response Time (% of Span/ Δ °C/ Δ Time)	5 %/7 °C/7.8 seconds 20 %/28 °C/33.6 seconds 63.2 %/88.48 °C/150 seconds
Output Accuracy	$\pm$ 0.5 °C @ 25 °C
Repeatability	$\pm$ 0.25 °C
Thermal Time Constant	2.5 minutes typical (still air)
Cable Length	>2,000 ft (610 meters)
Dimensions	Overall length: 39" (1 m) Cylinder tip: 0.70" (1.78 cm) L, 0.375" (0.95 cm) D

Appendix C: HOBO data logger specifications

Overview

Specs

Compatible/Req. Items

Acc.

Resources

Support

The HOBO U12 4-External Channel Data Logger accepts a wide range of external sensors, including temperature, AC current, AC voltage, CO₂, 4-20mA, and 0 - 2.5 vdc. 12-bit resolution provides great data accuracy.

Features:

- 12-bit resolution provides high accuracy
- Large memory for long-term deployments or fast sampling
- Five models to choose from, with flexible measurement options
- Programmable and push button start

Environment: Measurement:

Indoor
AC Current
Temperature
DC Voltage
4-20mA
AC Voltage

Product ID:

U12-006

Product Name:

4-Channel External
Data Loggers

Product Type:

Overview

Specs

Compatible/Req. Items

Acc.

Resources

Support

Measurement range: External input channels (see sensor manual): 0 to 2.5 DC Volts

Accuracy (logger only): $\pm 2 \text{ mV} \pm 2.5\%$ of absolute reading; $\pm 2 \text{ mV} \pm 1\%$ of reading for logger-powered sensors

Resolution: 0.6 mV

Time accuracy: ± 1 minute per month at 25°C (77°F), see Plot A

Operating range: -20 to 70°C (-4° to 158°F)

Operating temperature:

Logging: -20° to 70°C (-4° to 158°F)

Launch/readout: 0° to 50°C (32° to 122°F), per USB specification

Plot A

Humidity range: 0 to 95% RH, non-condensing

Battery life: 1 year typical use

Memory: 64K bytes (43,000 12-bit measurements)

Weight: 46 g (1.6 oz)

Dimensions: 58 × 74 × 22 mm (2.3 × 2.9 × 0.9 inches)

The CE Marking identifies this product as complying with all relevant directives in the European Union (EU).

Appendix D: Pressure transducer specifications

Model 264
Very Low Differential Pressure Transducer
Unidirectional Ranges: 0 - 0.1 to 0 - 100 In. W.C.
Bidirectional Ranges: 0 - ± 0.5 to 0 - ± 50 In. W.C.
Air or Non-Conducting Gas

**3-year
Unconditional
Warranty**

**HIGH OVERPRESSURE
CAPABILITY
Up to 10 PSI**

Setra Systems 264 pressure transducers sense differential or gauge (static) pressure and convert this pressure difference to a proportional electrical output for either unidirectional or bidirectional pressure ranges. The 264 Series is offered with a high level analog 0 to 5 VDC or 4 to 20 mA output.

Used In Building Energy Management Systems, these transducers are capable of measuring pressures and flows with the accuracy necessary for proper building pressurization and air flow control.

The 264 Series transducers are available for air pressure ranges as low as 0.1 In. W.C. full scale to 100 In. W.C. full scale. Static standard accuracy is $\pm 1.0\%$ full scale in normal ambient temperature environments, but higher accuracies are available. The units are temperature compensated to 0.033% FS/°F thermal error over the temperature range of 0°F to +150°F.

The Model 264 utilizes an improved all stainless steel micro-tig welded sensor. The tensioned stainless steel diaphragm and insulated stainless steel electrode, positioned close to the diaphragm, form a variable capacitor. Positive pressure moves the diaphragm toward the electrode, increasing the capacitance. A decrease in pressure moves the diaphragm away from the electrode, decreasing the capacitance. The change in capacitance is detected and converted to a linear DC electrical signal by Setra's unique electronic circuit.

The tensioned sensor allows up to 10 PSI overpressure (in either direction) with no damage to the unit. In addition, the parts that make up the sensor have thermally matched coefficients, which promote improved temperature performance and excellent long term stability.

NOTE: Setra quality standards are based on ANSI-Z540-1. The calibration of this product is NIST traceable.

U.S. Patent nos. 4093915; 4358814; 4434203; 6019002; 6014800. Other Patents Pending.

159 Swanson Rd., Boxborough, MA 01719/Telephone: 978-263-1400/Fax: 978-264-0292

Applications

- Heating, Ventilating and Air Conditioning (HVAC)
- Energy Management Systems
- Variable Air Volume and Fan Control (VAV)
- Environmental Pollution Control
- Lab and Fume Hood Control
- Oven Pressurization and Furnace Draft Controls

Features

- Up to 10 PSI Overpressure on All Ranges
- Installation Time Minimized with Snap Track Mounting and Easy-To-Access Pressure Ports and Electrical Connections
- 0 to 5 VDC or 2-wire 4 to 20 mA Analog Outputs Are Compatible with Energy Management Systems
- Reverse Wiring Protection
- Internal Regulation Permits Use with Unregulated DC Power Supplies
- Fire Retardant Case (UL 94 V-0 Approved)
- Meets CE Conformance Standards

When it comes to a product to rely on - choose the Model 264. When it comes to a company to trust - choose Setra.

Visit Setra Online:
<http://www.setra.com>

setra
800-257-3872

Model 264 Specifications

Performance Data

	Standard	Optional
Accuracy* RSS (at constant temp)	±1.0% FS	±0.4% FS ±0.25% FS
Non-Linearity, BSL	±0.96% FS	±0.38% FS ±0.22% FS
Hysteresis	0.10% FS	0.10% FS 0.10% FS
Non-Repeatability	0.05% FS	0.05% FS 0.05% FS

Thermal Effects**

Compensated Range °F(°C)	0 to +150 (-18 to +65)
Zero/Span Shift %FS/°F(°C)	0.033 (0.06)
Maximum Line Pressure	10 psi
Overpressure	Up to 10 psi (Range Dependent)
Long Term Stability	0.5% FS/1 YR

Position Effect	Range	Zero Offset (%FS/G)
(Unit is factory calibrated at 0g effect in the vertical position.)	To 0.5 in. WC	0.60
	To 1.0 in. WC	0.50
	To 2.5 in. WC	0.22
	To 5 in. WC	0.14

*RSS of Non-Linearity, Hysteresis, and Non-Repeatability.

**Units calibrated at nominal 70°F. Maximum thermal error computed from this datum.

Environmental Data

Temperature	
Operating* °F (°C)	0 to +175 (-18 to +79)
Storage °F (°C)	-65 to +250 (-54 to +121)

*Operating temperature limits of the electronics only. Pressure media temperatures may be considerably higher.

Physical Description

Case	Fire-Retardant Glass Filled Polyester (UL 94 V-0 Approved)
Mounting	Four screw holes on removable zinc plated steel base (designed for 2.75" snap track)
Electrical Connection	Screw Terminal Strip
Pressure Fittings	3/16" O.D. barbed brass pressure fitting for 1/4" push-on tubing
Zero and Span Adjustments	Accessible on top of case
Weight (approx.)	10 ounces

Pressure Media

Typically air or similar non-conducting gases.

Specifications subject to change without notice.

Electrical Data (Voltage)

Circuit	3-Wire (Com, Exc, Out)
Excitation	9 to 30 VDC
Output*	0 to 5 VDC**

Bidirectional output at zero pressure:

2.5 VDC**

Output Impedance

100 ohms

*Calibrated into a 50K ohm load, operable into a 5000 ohm load or greater.

**Zero output factory set to within ±50mV (±25 mV for optional accuracies).

***Span (Full Scale) output factory set to within ±50mV (±25 mV for optional accuracies).

Electrical Data (Current)

Circuit

2-Wire

Output*

4 to 20mA**

Bidirectional output at zero pressure:

12mA**

External Load

0 to 800 ohms

Minimum supply voltage (VDC) = 9+ 0.02 x

(Resistance of receiver plus line).

Maximum supply voltage (VDC) = 30+ 0.004 x

(Resistance of receiver plus line).

*Calibrated at factory with a 24 VDC loop supply voltage and a 250 ohm load.

**Zero output factory set to within ±0.16mA (±0.06 mA for optional accuracies).

***Span (Full Scale) output factory set to within ±0.16mA (±0.06 mA for optional accuracies).

Outline Drawings

Code T1 Electrical Termination Dimensions

Optional 1/2" Conduit Electrical Enclosure Dimensions

ORDERING INFORMATION

Code all blocks in table.

Example: Part No. 26412

Model	Differential	Bidirectional	Output	Elec. Termination	Accuracy
2641 = 264	0R1WD = 0 to 0.1 in. WC R25WD = 0 to 0.25 in. WC 0R5WD = 0 to 0.5 in. WC 001WD = 0 to 1 in. WC 2R5WD = 0 to 2.5 in. WC 003WD = 0 to 3 in. WC 005WD = 0 to 5 in. WC 010WD = 0 to 10 in. WC 015WD = 0 to 15 in. WC 025WD = 0 to 25 in. WC 050WD = 0 to 50 in. WC 100WD = 0 to 100 in. WC	R05WB = ±0.05 in. WC 0R1WB = ±0.1 in. WC R25WB = ±0.25 in. WC 0R5WB = ±0.5 in. WC 001WB = ±1 in. WC 1R5WB = ±1.5 in. WC 2R5WB = ±2.5 in. WC 005WB = ±5 in. WC 7R5WB = ±7.5 in. WC 010WB = ±10 in. WC 025WB = ±25 in. WC 050WB = ±50 in. WC	11 = 4-20 mA 20 = 0 to 5 VDC	Standard T1 = Terminal Strip Optional A1 = 1/2" Conduit Enclosure	C = ±1% FS Optional (w/Cal. Cert.) E = ±0.4% FS F = ±0.25% FS G = ±1% FS

Please contact factory for versions not shown.

We will provide application assistance on all Setra products, both personally and through our literature, it is the customer's responsibility to determine the suitability of the product in the application.

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Toll Free: 800-257-3872; Fax: 978-264-0292; email: sales@setra.com

setra

SSP 264 Rev F 06/24/08

Appendix E: RTD sensor specifications

Product Data

Low Temperature Transmitters

I/TT100-LT I/TT1000-LT

Product Description

The I/TT100-LT and I/TT1000-LT Series temperature transmitters incorporate a Class B wire wound platinum RTD sensor which conforms to DIN-IEC-751 platinum RTD standards.

INTEC provides two temperature transmitters, the I/TT100-LT for use with a 100 Ohm RTD input and the I/TT1000-LT for use with a 1000 ohm RTD input. The transmitters produce a linear 4-20 mA output carried on a two-wire system. Optional output voltage ranges of 1 to 5 or 2 to 10 VDC can also be ordered. The input voltage to the transmitter may vary from 7-35 VDC. The transmitter is supplied factory calibrated and is protected from EMI and load resistances up to 800 ohms, when using a 4-20 mA output. The transmitter's output current, may be limited upon request.

One of the most important features of the I/TT100-LT and I/TT1000-LT temperature transmitters is the use of a strain free wire wound RTD. This allows the sensor to have a more stable output since the sensing element isn't subjected to the constant expansion and contraction of the encapsulation materials. The sensors are mounted in a 1/4" diameter stainless steel probe and come with standard 24" Teflon leads, except for the remote probes which come with 6' Teflon leads.

Some of the typical applications are for Freezers, Chemical Plants, Fuel Processing Facilities, Meat Packing Plants, and Scientific Laboratories.

These units are offered in Duct, Duct without box, Immersion, and Remote Probe configurations.

Product Specifications

Input	2-wire, Platinum 100/1000RTD	Transmitter Accuracy	+/- 0.1% of span
Output	2-wire, Linear 4 - 20mA DC Current	Sensor Accuracy	+/- 0.3°C @ 0°C Single Point
Supply Voltage	7 to 35 VDC, No Polarity Sensitivity	Temperature Sensor Range	-328 to 392°F (-200 to 200°C)
Linearity	+/- 0.05% of span	Transmitter Operating Range	32 to 158° F

Dimensions

DUCT

IMMERSION

OUTSIDE

REMOTE PROBE

Ordering Information

Sensor I / []	Span []	Configuration []	Output []
↑ TT100LT (100 Ohm RTD) TT1KLT (1000 Ohm RTD)	↑ (Specify)	↑ D- Duct - 4", 8", 12", 18" DO- Duct without box 4", 8", 12", 18" I- Immersion 4", 8" RP- Remote Probe	↑ (1) 1 to 5 VDC (2) 2 to 10 VDC (4) 4 to 20 mA

Example: I/TT1KLT -(-80 to 0°C) -I -4" -(4) or I/TT100LT -(-30 to -150°F) -RP -(4)

Appendix F: EES file for temperature model

"Determination of the relationship between temperature vs. time during the defrost"

"Known Information for aluminum fin"

rho_fin = 173 [lb/ft^3]	"Assume density of the fin is constant through out"
thickness = 0.0003333333333 [feet]	"This must change to feet"
h_c = 0.88 [Btu/hr ft^2 F]	"Using natural convection"
T_amb = 35 [F]	"Temperature of the room"
Cp_fin = 0.21 [Btu/lbm*F] through out the fin"	"Assume specific heat to be the same"
k_fin = 101 [Btu/hr*ft*F]	"Assume Thermal conductivity to be the same throughout the fin"
T_hot_gas = 55 [F]	"This takes into account of Q_in by knowing boundary condition temperature"
T_i = 35 [F]	"Initial condition"

"Known Information for copper tube wall"

rho_copper = 557.7 [lb/ft^3] out"	"Assume density of the copper is constant through out"
h_R22 = 440.3 [Btu/h ft^2 F] condensation"	"Using convection with phase change"
fin_pitch = 0.018 [feet]	
Cp_copper = 0.0906 [Btu/lbm*F] the copper"	"Assume specific heat to be the same through out the copper"
k_copper = 234 [Btu/hr*ft*F] through out the copper"	"Assume Thermal conductivity to be the same"
"Radius of the copper tube"	
r_a = 0.025125	[feet]
r_b = 0.026375	[feet]
r_c = 0.027625	[feet]

"Volume of copper tube"

$$\begin{aligned} \text{volume_tube_a} &= \pi * (((r_a + r_b)/2)^2 - r_a^2) * (\text{fin_pitch}/2) \\ \text{volume_tube_b} &= \pi * (((r_b + r_c)/2)^2 - ((r_a + r_b)/2)^2) * (\text{fin_pitch}/2) \\ \text{volume_tube_c} &= \pi * (r_c^2 - ((r_b + r_c)/2)^2) * (\text{fin_pitch}/2) \\ \text{volume_fin_0} &= \pi * (((r_c + r_1)/2)^2 - r_c^2) * (\text{thickness}/2) \end{aligned}$$

"Solving Equation at the tube wall"

$$f_tTa = (Q_conv_R22_to_a - Q_cond_a_to_b) / (\rho_{copper} * \text{volume_tube_a} * C_{p_copper})$$

$$T_a = T_i + \text{integral}(f_tTa, t)$$

$$f_tTb = (Q_cond_a_to_b - Q_cond_b_to_c) / (\rho_{copper} * \text{volume_tube_b} * C_{p_copper})$$

$$T_b = T_i + \text{integral}(f_tTb, t)$$

$$f_tTc = (Q_cond_b_to_c - Q_conv_c_to_amb - Q_cond_c_to_1 - Q_cond_c_f0) / (\rho_{copper} * \text{volume_tube_c} * C_{p_copper} + \rho_{fin} * \text{volume_fin_0} * C_{p_fin})$$

$$T_c = T_i + \text{integral}(f_tTc, t)$$

"Radius of the Rings"

$$\begin{aligned} r_0 &= 0.027625 & [\text{feet}] \\ r_1 &= 0.038768 & [\text{feet}] \\ r_2 &= 0.049913 & [\text{feet}] \\ r_3 &= 0.061056 & [\text{feet}] \\ r_4 &= 0.072200 & [\text{feet}] \\ r_5 &= 0.083344 & [\text{feet}] \end{aligned}$$

"Volume of the Rings"

$$\begin{aligned} \text{volume_fin_1} &= \pi * (((r_1 + r_2)/2)^2 - ((r_c + r_1)/2)^2) * (\text{thickness}/2) \\ \text{volume_fin_2} &= \pi * (((r_2 + r_3)/2)^2 - ((r_1 + r_2)/2)^2) * (\text{thickness}/2) \\ \text{volume_fin_3} &= \pi * (((r_3 + r_4)/2)^2 - ((r_2 + r_3)/2)^2) * (\text{thickness}/2) \\ \text{volume_fin_4} &= \pi * (((r_4 + r_5)/2)^2 - ((r_3 + r_4)/2)^2) * (\text{thickness}/2) \\ \text{volume_fin_5} &= \pi * (r_5^2 - ((r_5 + r_4)/2)^2) * (\text{thickness}/2) \end{aligned}$$

"Solving for temperature at fin"

"Ring #1"

$$f_tT1 = (Q_cond_c_to_1 - Q_cond_1_to_2 - Q_cond_1_to_f1) / (\rho_{fin} * \text{volume_fin_1} * C_{p_fin})$$

$$T_1 = T_i + \text{integral}(f_tT1, t)$$

"Ring #2"

$$f_tT2 = (Q_cond_1_to_2 - Q_cond_2_to_3 - Q_cond_2_to_f2) / (\rho_fin * volume_fin_2 * Cp_fin)$$

$$T_2 = T_i + \text{integral}(f_tT2, t)$$

"Ring #3"

$$f_tT3 = (Q_cond_2_to_3 - Q_cond_3_to_4 - Q_cond_3_to_f3) / (\rho_fin * volume_fin_3 * Cp_fin)$$

$$T_3 = T_i + \text{integral}(f_tT3, t)$$

"Ring #4"

$$f_tT4 = (Q_cond_3_to_4 - Q_cond_4_to_5 - Q_cond_4_to_f4) / (\rho_fin * volume_fin_4 * Cp_fin)$$

$$T_4 = T_i + \text{integral}(f_tT4, t)$$

"Ring #5"

$$f_tT5 = (Q_cond_4_to_5 - Q_cond_5_to_f5) / (\rho_fin * volume_fin_5 * Cp_fin)$$

$$T_5 = T_i + \text{integral}(f_tT5, t)$$

"Solving for the frost Temperature"

"Information about frost"

$$k_frost = 1.283 \text{ [Btu/hr*ft*F]}$$

"Get the info from online: http://www.engineeringtoolbox.com/ice-thermal-properties-d_576.html"

$$frost_thickness = 0.5 * 0.003281 \text{ [feet]} \quad \text{"Convert from 1 millimeter of thickness"}$$

$$\rho_frost = 18.7 \text{ [lb/ft}^3\text{]}$$

"Get the info from online: http://www.engineeringtoolbox.com/ice-thermal-properties-d_576.html"

$$g_md = 0.000369 \text{ [lbm/ft}^2 \text{ s]}$$

"Calculate from the mass transfer book by Mills page 53 to 54"

$$m_H2O_s = 0.003745$$

"Calculate from the mass transfer book by Mills page 53 to 54"

$$m_H2O_e = 0.003476$$

"Calculate from the mass transfer book by Mills page 53 to 54"

$h_{ig} = 143.4$ [Btu/lbm]

"Enthalpy fusion of water"

$h_i = \text{Enthalpy}(\text{Water}, T=30, p=1)$

"Q_evap ring"

$Q_{\text{evap_0}} = (g_md * \pi * (((r_1 + r_c)/2)^2 - (r_c)^2) * (m_H2O_s - m_H2O_e) * h_{ig})$
 $Q_{\text{evap_1}} = (g_md * \pi * (((r_2 + r_1)/2)^2 - ((r_1 + r_c)/2)^2) * (m_H2O_s - m_H2O_e) * h_{ig})$
 $Q_{\text{evap_2}} = (g_md * \pi * (((r_3 + r_2)/2)^2 - ((r_2 + r_1)/2)^2) * (m_H2O_s - m_H2O_e) * h_{ig})$
 $Q_{\text{evap_3}} = (g_md * \pi * (((r_4 + r_3)/2)^2 - ((r_3 + r_2)/2)^2) * (m_H2O_s - m_H2O_e) * h_{ig})$
 $Q_{\text{evap_4}} = (g_md * \pi * (((r_5 + r_4)/2)^2 - ((r_4 + r_3)/2)^2) * (m_H2O_s - m_H2O_e) * h_{ig})$
 $Q_{\text{evap_5}} = (g_md * \pi * (r_5^2 - ((r_5 + r_4)/2)^2) * (m_H2O_s - m_H2O_e) * h_{ig})$

"Frost ring"

$\text{volume_frost_0} = \pi * (((r_c + r_1)/2)^2 - (r_c)^2) * ((\text{fin_pitch}/2) - (\text{thickness}/2))$
 $\text{volume_frost_1} = \pi * (((r_1 + r_2)/2)^2 - ((r_c + r_1)/2)^2) * \text{frost_thickness}$
 $\text{volume_frost_2} = \pi * (((r_2 + r_3)/2)^2 - ((r_1 + r_2)/2)^2) * \text{frost_thickness}$
 $\text{volume_frost_3} = \pi * (((r_3 + r_4)/2)^2 - ((r_2 + r_3)/2)^2) * \text{frost_thickness}$
 $\text{volume_frost_4} = \pi * (((r_4 + r_5)/2)^2 - ((r_3 + r_4)/2)^2) * \text{frost_thickness}$
 $\text{volume_frost_5} = \pi * (r_5^2 - ((r_5 + r_4)/2)^2) * \text{frost_thickness}$

"Ring #0"

$f_{th0} = (Q_{\text{cond_c_f0}} - Q_{\text{conv_f0_to_amb}} - Q_{\text{cond_f0_to_f1}} - Q_{\text{evap_0}}) / (\rho_{\text{frost}} * \text{volume_frost_0})$

$h_{f_0} = h_i + \text{integral}(f_{th0}, t)$

$\{T_{f_0} = \text{TEMPERATURE}(\text{Water}, h=h_{f_1}, p=1)\}$
 $T_{f_0} = \text{Interpolate1}(\text{'Lookup 1'}, 'T', 'h', h=h_{f_0})$

"Ring #1"

$f_{th1} = (Q_{\text{cond_f0_to_f1}} - Q_{\text{conv_f1_to_amb}} - Q_{\text{cond_f1_to_f2}} + Q_{\text{cond_1_to_f1}} - Q_{\text{evap_1}}) / (\rho_{\text{frost}} * \text{volume_frost_1})$

$h_{f_1} = h_i + \text{integral}(f_{th1}, t)$

$\{T_{f_1} = \text{TEMPERATURE}(\text{Water}, h=h_{f_1}, p=1)\}$
 $T_{f_1} = \text{Interpolate1}(\text{'Lookup 1'}, 'T', 'h', h=h_{f_1})$

"Ring #2"

$f_{th2} = (Q_{\text{cond_f1_to_f2}} - Q_{\text{conv_f2_to_amb}} - Q_{\text{cond_f2_to_f3}} + Q_{\text{cond_2_to_f2}} - Q_{\text{evap_2}}) / (\rho_{\text{frost}} * \text{volume_frost_2})$

$h_{f_2} = h_i + \text{integral}(f_{th2}, t)$

$\{T_{f_2} = \text{TEMPERATURE}(\text{Water}, h=h_{f_2}, p=1)\}$
 $T_{f_2} = \text{Interpolate1}(\text{'Lookup 1'}, 'T', 'h', h=h_{f_2})$

"Ring #3"

$$f_{th3} = (Q_{cond_f2_to_f3} - Q_{conv_f3_to_amb} - Q_{cond_f3_to_f4} + Q_{cond_3_to_f3} - Q_{evap_3}) / (\rho_{frost} \cdot volume_{frost_3})$$

$$h_{f_3} = h_i + \text{integral}(f_{th3}, t)$$

$$\{T_{f_3} = \text{TEMPERATURE}(\text{Water}, h=h_{f_3}, p=1)\}$$

$$T_{f_3} = \text{Interpolate1}('Lookup\ 1', 'T', 'h', h=h_{f_3})$$

"Ring #4"

$$f_{th4} = (Q_{cond_f3_to_f4} - Q_{conv_f4_to_amb} - Q_{cond_f4_to_f5} + Q_{cond_4_to_f4} - Q_{evap_4}) / (\rho_{frost} \cdot volume_{frost_4})$$

$$h_{f_4} = h_i + \text{integral}(f_{th4}, t)$$

$$\{T_{f_4} = \text{TEMPERATURE}(\text{Water}, h=h_{f_4}, p=1)\}$$

$$T_{f_4} = \text{Interpolate1}('Lookup\ 1', 'T', 'h', h=h_{f_4})$$

"Ring #5"

$$f_{th5} = (Q_{cond_f4_to_f5} - Q_{conv_f5_to_amb} + Q_{cond_5_to_f5} - Q_{evap_5}) / (\rho_{frost} \cdot volume_{frost_5})$$

$$h_{f_5} = h_i + \text{integral}(f_{th5}, t)$$

$$\{T_{f_5} = \text{TEMPERATURE}(\text{Water}, h=h_{f_5}, p=1)\}$$

$$T_{f_5} = \text{Interpolate1}('Lookup\ 1', 'T', 'h', h=h_{f_5})$$

"List of all the Heat energy terms"

$$Q_{conv_R22_to_a} = (h_{R22} \cdot \pi \cdot r_a \cdot fin_pitch \cdot (T_{hot_gas} - T_a)) / 3600$$

$$Q_{CONV_R22_to_a_sum} = \text{integral}(Q_{conv_R22_to_a}, t)$$

$$Q_{cond_a_to_b} = (k_{copper} \cdot \pi \cdot fin_pitch \cdot (T_a - T_b) / \ln(r_b / r_a)) / 3600$$

$$Q_{COND_a_to_b_sum} = \text{integral}(Q_{cond_a_to_b}, t)$$

$$Q_{cond_b_to_c} = (k_{copper} \cdot \pi \cdot fin_pitch \cdot (T_b - T_c) / \ln(r_c / r_b)) / 3600$$

$$Q_{COND_b_to_c_sum} = \text{integral}(Q_{cond_b_to_c}, t)$$

$$Q_{conv_c_to_amb} = (h_c \cdot \pi \cdot r_c \cdot ((fin_pitch) / 2 - (thickness) / 2 - frost_thickness) \cdot (T_c - T_{amb})) / 3600$$

$$Q_{CONV_c_to_amb_sum} = \text{integral}(Q_{conv_c_to_amb}, t)$$

$$Q_{cond_c_to_1} = (k_{fin} \cdot \pi \cdot thickness \cdot (T_c - T_1) / \ln(r_1 / r_c)) / 3600$$

$$Q_{COND_c_to_1_sum} = \text{integral}(Q_{cond_c_to_1}, t)$$

$$Q_{cond_c_f0} = (((2 \cdot \pi \cdot ((r_1 + r_c) / 2)^2 - (r_c)^2) \cdot (T_c - T_{f_0}) \cdot (k_{fin} \cdot k_{frost})) / ((thickness \cdot k_{frost}) + (2 \cdot frost_thickness \cdot k_{fin}))) / 3600 + ((k_{frost} \cdot 2 \cdot \pi \cdot (fin_pitch / 2 - thickness / 2) \cdot (T_c - T_{f_0}) / \ln(r_0 / r_c)) / 3600)$$

$$Q_COND_c_f0_sum = \text{integral}(Q_cond_c_f0, t)$$

$$Q_cond_1_to_2 = (k_fin * \pi * \text{thickness} * (T_1 - T_2) / \ln(r_2/r_1)) / 3600$$

$$Q_COND_1_to_2_sum = \text{integral}(Q_cond_1_to_2, t)$$

$$Q_cond_1_to_f1 = ((2 * \pi * (((r_2 + r_1)/2)^2 - ((r_1 + r_c)/2)^2) * (T_1 - T_f_1) * (k_fin * k_frost)) / ((\text{thickness} * k_frost) + (2 * \text{frost_thickness} * k_fin))) / 3600$$

$$Q_COND_1_to_f1_sum = \text{integral}(Q_cond_1_to_f1, t)$$

$$Q_cond_2_to_3 = (k_fin * \pi * \text{thickness} * (T_2 - T_3) / \ln(r_3/r_2)) / 3600$$

$$Q_COND_2_to_3_sum = \text{integral}(Q_cond_2_to_3, t)$$

$$Q_cond_2_to_f2 = ((2 * \pi * (((r_3 + r_2)/2)^2 - ((r_2 + r_1)/2)^2) * (T_2 - T_f_2) * (k_fin * k_frost)) / ((\text{thickness} * k_frost) + (2 * \text{frost_thickness} * k_fin))) / 3600$$

$$Q_COND_2_to_f2_sum = \text{integral}(Q_cond_2_to_f2, t)$$

$$Q_cond_3_to_4 = (k_fin * \pi * \text{thickness} * (T_3 - T_4) / \ln(r_4/r_3)) / 3600$$

$$Q_COND_3_to_4_sum = \text{integral}(Q_cond_3_to_4, t)$$

$$Q_cond_3_to_f3 = ((2 * \pi * (((r_4 + r_3)/2)^2 - ((r_3 + r_2)/2)^2) * (T_3 - T_f_3) * (k_fin * k_frost)) / ((\text{thickness} * k_frost) + (2 * \text{frost_thickness} * k_fin))) / 3600$$

$$Q_COND_3_to_f3_sum = \text{integral}(Q_cond_3_to_f3, t)$$

$$Q_cond_4_to_5 = (k_fin * \pi * \text{thickness} * (T_4 - T_5) / \ln(r_5/r_4)) / 3600$$

$$Q_COND_4_to_5_sum = \text{integral}(Q_cond_4_to_5, t)$$

$$Q_cond_4_to_f4 = ((2 * \pi * (((r_5 + r_4)/2)^2 - ((r_4 + r_3)/2)^2) * (T_4 - T_f_4) * (k_fin * k_frost)) / ((\text{thickness} * k_frost) + (2 * \text{frost_thickness} * k_fin))) / 3600$$

$$Q_COND_4_to_f4_sum = \text{integral}(Q_cond_4_to_f4, t)$$

$$Q_cond_5_to_f5 = ((2 * \pi * ((r_5)^2 - ((r_5 + r_4)/2)^2) * (T_5 - T_f_5) * (k_fin * k_frost)) / ((\text{thickness} * k_frost) + (2 * \text{frost_thickness} * k_fin))) / 3600$$

$$Q_COND_5_to_f5_sum = \text{integral}(Q_cond_5_to_f5, t)$$

$$Q_cond_f0_to_f1 = (k_frost * 2 * \pi * \text{frost_thickness} * (T_f_0 - T_f_1) / \ln(r_1/r_0)) / 3600$$

$$Q_COND_f0_to_f1_sum = \text{integral}(Q_cond_f0_to_f1, t)$$

$$Q_conv_f0_to_amb = (h_c * \pi * (((r_1 + r_c)/2)^2 - (r_c)^2) * (T_f_0 - T_amb)) / 3600 + ((h_c * \pi * (r_1 + r_c)/2) * ((\text{fin_pitch}/2) - (\text{thickness}/2) - \text{frost_thickness}) * (T_f_0 - T_amb)) / 3600$$

$$Q_CONV_f0_to_amb_sum = \text{integral}(Q_conv_f0_to_amb, t)$$

$$Q_conv_f1_to_amb = (h_c * \pi * (((r_2 + r_1)/2)^2 - ((r_1 + r_c)/2)^2) * (T_f_1 - T_amb)) / 3600$$

$$Q_CONV_f1_to_amb_sum = \text{integral}(Q_conv_f1_to_amb, t)$$

```

Q_cond_f1_to_f2 = (k_frost*2*pi*frost_thickness*(T_f_1-T_f_2)/LN(r_2/r_1))/3600
Q_COND_f1_to_f2_sum = integral(Q_cond_f1_to_f2, t)
Q_conv_f2_to_amb = (h_c*pi*(((r_3+r_2)/2)^2-((r_2+r_1)/2)^2)*(T_f_2 -T_amb))/3600
Q_CONV_f2_to_amb_sum = integral(Q_conv_f2_to_amb, t)
Q_cond_f2_to_f3 = (k_frost*2*pi*frost_thickness*(T_f_2-T_f_3)/LN(r_3/r_2))/3600
Q_COND_f2_to_f3_sum = integral(Q_cond_f2_to_f3, t)
Q_conv_f3_to_amb = (h_c*pi*(((r_4+r_3)/2)^2-((r_3+r_2)/2)^2)*(T_f_3 -T_amb))/3600
Q_CONV_f3_to_amb_sum = integral(Q_conv_f3_to_amb, t)
Q_cond_f3_to_f4 = (k_frost*2*pi*frost_thickness*(T_f_3-T_f_4)/LN(r_4/r_3))/3600
Q_COND_f3_to_f4_sum = integral(Q_cond_f3_to_f4, t)
Q_conv_f4_to_amb = (h_c*pi*(((r_5+r_4)/2)^2-((r_4+r_3)/2)^2)*(T_f_4 -T_amb))/3600
Q_CONV_f4_to_amb_sum = integral(Q_conv_f4_to_amb, t)
Q_cond_f4_to_f5 = (k_frost*2*pi*frost_thickness*(T_f_4-T_f_5)/LN(r_5/r_4))/3600
Q_COND_f4_to_f5_sum = integral(Q_cond_f4_to_f5, t)
Q_conv_f5_to_amb = (h_c*pi*(r_5^2-((r_5+r_4)/2)^2)*(T_f_5 -T_amb))/3600
Q_CONV_f5_to_amb_sum = integral(Q_conv_f5_to_amb, t)

Q_conv = (h_c*pi*r_c*(fin_pitch-thickness)*(T_c-T_amb) + h_c*pi*(((r_2+r_1)/2)^2-
((r_1+r_c)/2)^2)*(T_f_1 -T_amb) + h_c*pi*(((r_3+r_2)/2)^2-((r_2+r_1)/2)^2)*(T_f_2 -T_amb)
+ h_c*pi*(((r_4+r_3)/2)^2-((r_3+r_2)/2)^2)*(T_f_3 -T_amb) + h_c*pi*(r_5^2-
((r_5+r_4)/2)^2)*(T_f_5 -T_amb) + h_c*pi*(r_5^2-((r_5+r_4)/2)^2)*(T_f_5 -T_amb))/3600

Q_CONV_TOTAL = integral(Q_conv, t)

Q_evap_sum = (Q_evap_1 + Q_evap_2 + Q_evap_3 + Q_evap_4 + Q_evap_5)
Q_EVAP_TOTAL = integral(Q_evap_sum, t)

Q_in = (h_R22*pi*r_a*fin_pitch*(T_hot_gas - T_a))/3600
Q_IN_TOTAL = integral(Q_in, t)

Q_tube=rho_copper*volume_tube_a*Cp_copper*(T_a-
T_i)+rho_copper*volume_tube_b*Cp_copper*(T_b-
T_i)+rho_copper*volume_tube_c*Cp_copper*(T_c-T_i)

Q_fin=rho_fin*volume_fin_0*Cp_fin*(T_c-T_i)+rho_fin*volume_fin_1*Cp_fin*(T_1-
T_i)+rho_fin*volume_fin_2*Cp_fin*(T_2-T_i)+rho_fin*volume_fin_3*Cp_fin*(T_3-
T_i)+rho_fin*volume_fin_4*Cp_fin*(T_4-T_i)+rho_fin*volume_fin_5*Cp_fin*(T_5-T_i)

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$$Q_{\text{melt}} = \rho_{\text{frost}} \cdot \text{volume_frost}_0 \cdot (h_{f_0} - h_i) + \rho_{\text{frost}} \cdot \text{volume_frost}_1 \cdot (h_{f_1} - h_i) + \rho_{\text{frost}} \cdot \text{volume_frost}_2 \cdot (h_{f_2} - h_i) + \rho_{\text{frost}} \cdot \text{volume_frost}_3 \cdot (h_{f_3} - h_i) + \rho_{\text{frost}} \cdot \text{volume_frost}_4 \cdot (h_{f_4} - h_i) + \rho_{\text{frost}} \cdot \text{volume_frost}_5 \cdot (h_{f_5} - h_i)$$

$$Q_{\text{Total}} = Q_{\text{tube}} + Q_{\text{fin}} + Q_{\text{melt}} + Q_{\text{evap_total}} + Q_{\text{conv_total}}$$

$$\text{ratio} = Q_{\text{total}} / (Q_{\text{in_total}} + 0.0000000001)$$

$$\text{eff} = Q_{\text{melt}} / (Q_{\text{in_total}} + 0.0000000001)$$

$$\text{number_elements} = 70000$$

$$\text{Cost} = 0.1 \cdot ((Q_{\text{IN_TOTAL}} \cdot (1 - \text{eff})) / 3) \cdot (\text{number_elements} / 3412)$$

Lookup Table		
Lookup 1		
Paste Special	1 T	2 h
Row 1	70	38.08
Row 2	65	33.09
Row 3	60	28.09
Row 4	55	23.09
Row 5	50	18.09
Row 6	45	13.09
Row 7	40	8.077
Row 8	35	3.051
Row 9	32	0.02635
Row 10	32	-143.1
Row 11	30	-144.1
Row 12	25	-146.6
Row 13	20	-149.2

Appendix G: EES file for pressure drop model

"Known values from given information"

frost_thickness = 1*0.03937 [inches]

V_1 = 600/60 [ft /s]

fin_pitch = 0.216 "Spacing between each fin"

fin_thickness = 0.004 + 2*frost_thickness

A_c = (fin_pitch - fin_thickness)*(S_T - D_tube) "Minimum free flow area"

A_c_t = fin_pitch *(S_T - D_tube) "For bare tube bank"

S_T = 1.75 [inches] "Transverse pitch"

S_L = 1.5 [inches] "Longitudinal pitch"

D_tube = 0.663 +2* frost_thickness

Viscosity_air = 0.041633 [lb/ft h]

Re = (rho_1)*(V_max)*(D_tube)*3600*(1/12)*(1/Viscosity_air)

"Known values from assumption"

specific_v_1 = 12.515 [ft^3/lb] "Assuming for air at 34 F"

rho_1 = 0.0799 [lb/ft^3]

"From this website: <http://www.denysschen.com/catalogue/density.asp>"

specific_v_2 = 12.453 [ft^3/lb] "Assuming for air at 32 F"

A_tube = pi*D_tube*(fin_pitch-fin_thickness)

A_fin = 2*((S_T*S_L) - ((pi/4)*(D_tube)^2)) "Exchanger total heat transfer Area on one side"

sigma = (S_T*0.212)/((S_T - D_tube)*(fin_pitch-fin_thickness))

V_max = V_1*sigma

G = rho_1*V_max

"Calculate friction factor"

f_tube = INTERPOLATE1('Re_vs_f_tube','Re','f_tube',Re=Re)

"Based on heat transfer textbook"

$$f_{fin} = (0.508 \cdot (Re)^{-0.521}) \cdot (S_T/D_{tube})^{1.318}$$

"Solving for pressure drop"

$$\Delta P_{fin} = (f_{fin} \cdot (6 \cdot A_{fin}/A_c) \cdot (G^2) \cdot (0.004015/0.020886)) / (64.4 \cdot \rho_{1})$$

$$\Delta P_{tube} = (f_{tube} \cdot (6 \cdot A_{tube}/A_{c_t}) \cdot (G^2) \cdot (0.004015/0.020886)) / (64.4 \cdot \rho_{1})$$

$$\Delta P = \Delta P_{tube} + \Delta P_{fin}$$

Re_vs_f_tube		
	1	2
	Re	f _{tube}
Row 1	1,000	0.43
Row 2	2,000	0.39
Row 3	4,000	0.37
Row 4	6,000	0.33
Row 5	8,000	0.31
Row 6	10,000	0.3

Appendix H: Weather data during experiment

History for Salinas, CA

Week of November 9, 2008 through November 15, 2008 — [View Current Conditions](#)

Weekly Summary				
« Previous Week November 15 2008 View Next Week »				
Daily Weekly Monthly Custom				
	Max:	Avg:	Min:	Sum:
Temperature:				
Max Temperature	89 °F	72 °F	60 °F	
Mean Temperature	72 °F	61 °F	54 °F	
Min Temperature	54 °F	49 °F	44 °F	
Degree Days:				
Heating Degree Days (base 65)	11	5	0	38
Cooling Degree Days (base 65)	7	1	0	9
Growing Degree Days (base 50)	22	11	4	74
Dew Point:				
Dew Point	55 °F	48 °F	34 °F	
Precipitation:				
Precipitation	0.00 in	0.00 in	0.00 in	0.00 in
Snowdepth	-	-	-	-
Wind:				
Wind	24 mph	6 mph	0 mph	
Gust Wind	28 mph	23 mph	20 mph	
Sea Level Pressure:				
Sea Level Pressure	30.33 in	30.14 in	29.88 in	

Daily Observations																				
2008	Temp. (°F)			Dew Point (°F)			Humidity (%)			Sea Level Pressure (in)			Visibility (mi)			Wind (mph)		Gust Speed (mph)	Precip (in)	Events
November	high	avg	low	high	avg	low	high	avg	low	high	avg	low	high	avg	low	high	avg	high	sum	
9	62	58	53	52	50	47	93	79	64	30.03	29.92	29.88	10	10	7	21	11	25	0.00	
10	60	55	49	50	49	45	93	78	62	30.21	30.09	30.00	10	9	7	14	4	16	0.00	
11	64	54	44	52	45	43	100	81	62	30.32	30.21	30.19	10	7	0	14	5	15	0.00	Fog
12	72	60	47	55	48	46	100	79	57	30.33	30.28	30.21	10	6	0	15	5	18	0.00	Fog
13	74	60	46	55	52	46	100	71	41	30.26	30.22	30.12	10	4	0	13	4	16	0.00	Fog
14	86	67	47	55	49	41	96	60	23	30.15	30.11	30.04	10	9	7	17	3	20	0.00	
15	89	72	54	49	43	34	72	43	14	30.14	30.12	30.06	10	10	10	24	9	28	0.00	
Comma Delimited File