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EXPERIMENTAL INVESTIGATIONS INTO FROST FORMATION ON DISPLAY CABINET EVAPORATORS IN ORDER TO IMPLEMENT DEFROST ON DEMAND

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ABSTRACT

This paper reports on experimental investigations carried out on a vertical high temperature display cabinet under controlled laboratory conditions and analyses the observed results to identify parameters which best represent the degradation of the system performance due to frosting. The effects of air temperature, air humidity and evaporator inlet temperature on frost growth and thermal performance are studied by using frost accumulation and an energy transfer coefficient based on the log mean enthalpy difference. The tests showed that for a standard evaporator coil and fan arrangement, with the fan speed being constant, air humidity is the most dominating factor for frost formation. Thus methods are suggested to monitor the difference in air absolute humidity based on a combination of temperatures and space humidity.

NOMENCLATURE

A_t = Heat/mass transfer surface area (m^2)
Q_t = Energy transfer (kW)
E_o = Energy transfer coefficient (W/m^2)
C_p = Specific heat capacity (kJ/kg-K)
Δi_m = Mean enthalpy difference

INTRODUCTION

Display Cabinet evaporators are prone to frost formation due to water vapour in the air condensing and freezing when the surface temperature of the evaporator falls below 0°C. A small amount of frost may improve the heat transfer performance of the coil by increasing the surface area and surface roughness that induces increased turbulence, Stoecker (1957). However, significant frost accumulation deteriorates the coil performance by reducing the air flow and thereby the refrigerating capacity of the evaporator. Use of air curtains to reduce penetration of store air into the display cabinets and maintaining the store humidity at low levels reduces the rate of frost formation on the evaporators to some extent, but does not eliminate it completely. Hence, the evaporators need to be actively defrosted periodically to maintain system performance and temperature control in the display cabinets.

Defrost in supermarket refrigeration systems is most frequently controlled by a preset time cycle as it is simple to install and easy to service. Defrosting involves the application of heat to the coil in order to melt the ice and this penalises the refrigeration system performance due to the fact that during the defrosting process energy is used while producing no useful cooling. Also, during the defrost cycle the cabinet and thus the product temperature rise above the set limits for normal operation as seen in figure 1. It is widely acknowledged that timed defrost may cause a number of unnecessary defrost cycles and this reduces the energy efficiency of refrigeration systems as well as the accuracy of temperature control of the cabinets.

Implementing defrost on demand should reduce the number of defrost cycles and thus will potentially lead to savings in energy. Demand defrost techniques which have been investigated over the years include air pressure differential measurement across the evaporator, sensing the temperature difference between the air and the evaporator surface, using fibre optic sensors to measure frost thickness etc. Due to excessive capital cost and reliability problems associated with complex and unreliable sensing methods none of these demand defrost techniques has gained wide application in the food retail industry. Furthermore, in supermarket refrigeration there are additional problems such as devising a defrost schedule so that all the display cabinets do not switch to defrost at the same time.

This paper aims to use the results of experimental investigations conducted on a display cabinet under controlled conditions to identify the parameters that best represent the degradation in system performance under frosting conditions. The overall objective is to use the identified parameters to devise an intelligent defrost controller based on these parameters.

EXPERIMENTAL PROCEDURE

A high temperature display cabinet 1.25 m (8 ft) long and 3.7kW (12,625 Btu/hr) cooling capacity at +5°C (41°F) was employed for testing purposes. Figure 2 shows a schematic diagram of the display cabinet whilst figure 3 is a photograph of the coil and fan arrangement. The cabinet was loaded with plastic containers filled with water and food substitute material. The refrigerant used was R22, fed from a mini-compressor pack designed to emulate the operation of standard supermarket pack. The refrigerant flow rate was measured using a Coriolis mass flow meter. Air and refrigerant temperatures were measured at various points in the cabinet using thermocouples. Measuring points included evaporator air on and air off temperatures, evaporator surface temperature and product temperatures at various positions in the cabinet. The operation of the system was controlled by a standard supermarket controller board. A computer, along with a data-acquisition module and software was used to record the various parameters at regular intervals.

In order to achieve steady state test conditions, the refrigerated cabinet was run overnight to bring the products down to an equilibrium temperature. Prior to all tests, a defrost cycle was initiated to ensure that the evaporator was clear of frost The test period was extended from the start of the first defrost to the end of the next defrost cycle. During the testing period the climatic condition in the environmental chamber was maintained at a constant temperature and relative humidity and the measured parameters were logged at 1-minute intervals. Figures 4 and 5 show the variation of the product and air temperatures for varying relative humidity conditions for constant refrigeration system operation. As expected, the air on the coil temperature increases with the increasing relative humidity, which in turn increases the load on the system. The air off temperature decreases more rapidly for higher relative humidity conditions and gradually increases as frost builds up, increasing the thermal resistance of the evaporator coil. Consequently, the product temperature as seen in figure 4 gradually increases at certain positions in the display cabinet.

DATA ANALYSIS

The formation of frost on evaporator coils has two major effects. Firstly, the thermal resistance of the system rises and secondly the air flow across the coil gradually deteriorates with time. The thermal resistance reflects the quality of frost formed and the air flow degradation is indirectly a measure of the amount of frost formed. The logged data were used to evaluate the amount of frost accumulated and the energy transfer coefficient as defined by Kondepudi et al (1989) in order to estimate both the quantity and quality of the frost formed. The air on and air off velocities were also measured using a hot wire anemometer to observe the degradation in the air flow due to frosting.

Frost Accumulation

It is difficult to estimate the height and growth rate of frost on the evaporator directly. An indirect method of estimating the amount of frost on the coil is to use the difference in measured absolute humidities before and after the test coil and numerically integrate this value over time. Figures 6, 7 and 8 show these calculated values for various conditions. The total amount of condensate at the end of the tests was collected and measured to check the validity of the estimated value. On average, the condensate was within 10-20% of the estimated value. This discrepancy can be attributed to the fact that some condensate evaporates into space while some water remains trapped between the fins.

Energy Transfer Coefficient

The thermal resistance of the system is the inverse of the product of the overall heat transfer coefficient U and the surface area of the coil, hence if U is low then the system has a high thermal resistance. Researchers have found that the coefficient decreases with the formation of frost, as the coefficient is a function of the frost thickness, frost properties, the geometry of the coil and the flow rate of the air across the coil. Predicting U is complex analytically. Usually log mean temperature difference (LMTD) is used to evaluate the U value but since frosting includes both sensible and latent energy transfer processes, log mean enthalpy difference is a more appropriate value to be calculated. The energy transfer coefficient is defined as

$$E_o = \frac{C_p \dot{Q}_t}{\Delta i_m A_t}$$

where E_o, based on the coil geometry is an easily measurable quantity and provides a qualitative and quantitative indication of the performance of the coil under frosting conditions. Figures 9 and 10 show the variation of the energy transfer coefficient for various conditions while table 1 provides the average values for them. In this evaluation, since the coil under testing is the same and the C_p does not vary significantly in the range of air temperatures considered the C_p/A_t is considered to be constant throughout for all the tests. The logarithmic mean enthalpy difference Δi_m is defined as $\Delta i_m = [(\Delta i_o - \Delta i_L)/\{\ln(\Delta i_o/\Delta i_L)\}]$ where $i_o = (i_{air\ in} - i_{ref\ out})$ and $i_L = (i_{air\ out} - i_{ref\ in})$. The derivation of Δi_m is along the similar lines to the LMTD with appropriate modifications and E_o is analogous to the sensible heat transfer coefficient.

RESULTS AND DISCUSSIONS

All tests carried out for this paper were with the same fan and coil arrangement and a constant fan speed. The variable parameters considered were the air dry and wet bulb temperatures and evaporator refrigerant inlet temperature. The air conditions were varied by an air handling unit while the evaporator refrigerant inlet temperature was varied by varying the cut-in temperature of the cabinet.

Effect of Air Humidity

Higher relative humidities lead to more rapid frost growth (figure 6). At the end of 320 minutes of cooling frost accumulated for relative humidity of 57% RH frost accumulated was about 1.8 times more than that accumulated for 40% RH for the same space air temperature of 22°C.

Figure 9 shows the effect of relative humidity on E_o. As the humidity increases, E_o increases. For a constant refrigerant temperature, higher humidities produce an increase in mass transfer due to more moisture content in the air. Hence the latent energy transfer is increased. For a 17% RH rise the energy transfer coefficient rises by 35%. Previous works (Stoeker, 1957 and Kondepudi et al, 1989) had reported the energy transfer coefficient to rise and then fall. The fall was not noticed in these experiments as the space relative humidity considered was lower than those reported in the literature and also, probably the duration of the tests where not long enough to show the decrease in value. The duration of the tests will be increased to verify this phenomenon in future investigations. Figure 11 shows the degradation of the airflow with time due to frosting for varying humidity. Air off velocity was measured at different point across the coil and an average value was obtained in order to monitor the degree of coil blockage. Higher space relative humidity encourages rapid degradation of the air off velocity. Figure 11 shows that it takes 1.3 times more time for the air off velocity to decrease by 70% of its initial value when the cabinet is operating at space condition of 50% RH than when it is at 65% RH.

Table 1. Energy Transfer Coefficient values for various conditions

Space Air Humidity (at constant temperature of 22°C)	40%RH	50%RH	57%RH	
Energy Transfer coefficient (Average) (W/m^2)	0.491154	0.623437	0.662555	

Space Air Temperature (at constant humidity of 50% RH)	22°C	24°C	26°C	28°C
Energy Transfer coefficient (Average) (W/m^2)	0.623437	0.702584	0.767027	0.912571

Refrigerant temperature at Evaporator Inlet	-6°C	-8°C	-10°C	
Energy Transfer coefficient (Average) (W/m^2)	0.636322	0.652539	0.632296	

Effect of Air Temperature

The amount of frost formed increases with increasing air temperature, figure 7. For 290 minutes cooling period the amount of frost at 28°C was 1.7 times that at 22°C. This is because the amount of moisture transferred into the frost layer increases with increasing temperature gradient. The energy transfer coefficient increases with the increase in air temperature. For a 6°C rise in temperature the energy transfer coefficient rises by 47%. Air off velocity shows that the difference in blockage of the coil with varying temperature is not significant, figure 12.

Effect of Evaporator Inlet Temperature

The evaporator inlet temperature and therefore the evaporator surface temperature do have significant effect on the formation of frost. However, this is not shown in figure 8 and table 1 as no significant changes are observed in both the amount of frost accumulated and the energy transfer coefficient. This is because the range of evaporator inlet temperature selected for this investigation was not large enough to show any notable trend.

Air Humidity Monitoring Scheme

Space air humidity and thus the air on humidity has the most dominating effect on the formation of frost. Monitoring air on humidity for each cabinet is not only difficult but also expensive as far as accuracy is concerned. Hence a temperature-based parameter along with the space humidity can be used to estimate the moisture content in the air. Kuwahara (1983) devised a method of determining the air on humidity of a coil by using the air on temperature, evaporator temperature and a fin surface temperature. In this case, it was observed that the relative humidity on the face of the coil is proportional to the ratio of (Air On temperature – Coil fin temperature adjacent to the refrigerant inlet point)/(Air On temperature – Coil inlet Temperature) and the space relative humidity. Both these factors can be combined to estimate the air on relative humidity and an example of using this method to determine the air on humidity is shown in figure 13.

CONCLUSIONS

1. Experimental investigations have identified the moisture content of the air onto the coil as the most dominating factor for frost formation. This is in agreement with all the past work. It was found that although air on temperature increases the rate of heat transfer of the coil, changes in the air on relative humidity have a more significant impact on the moisture transfer potential and hence the rate of frost formation on the coil.
2. Also, the degradation of air flow across the evaporator coil is more significant with increasing space relative humidity than with increasing space temperature as the amount of frost on the coil directly effect the air flow pattern.
3. Since monitoring air on humidity for every display cabinet is not financially viable air on humidity can be determined by the use of temperature ratio. Estimating the air on humidity will assist in the evaluation of the amount of frost collected at a given time and in turn, determine the defrost initiation and termination time.

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2. Kondepudi, S. N. and O'Neal, D L, May 1989, "Louvred finned tube heat exchangers", Rev. Int. Froid Vol 12.
3. Kuwahara, E, 1985, "A Humidity Detection Method for Use with a Room Air Conditioner", ASHRAE Trans. Vol. 91, pt 2A.

FIGURES

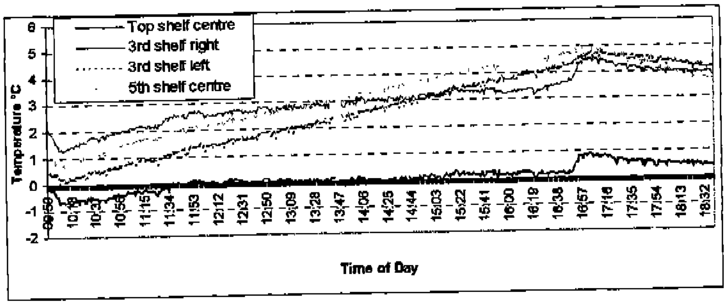

Figure 1: Product temperature variation during cooling and defrosting cycle

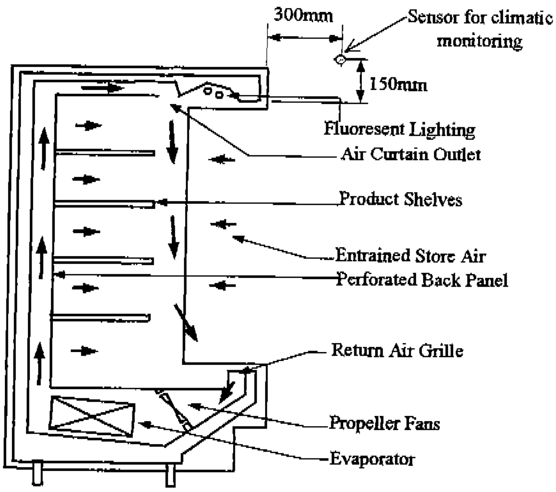

Figure 2 Schematic Diagram of the Display Cabinet

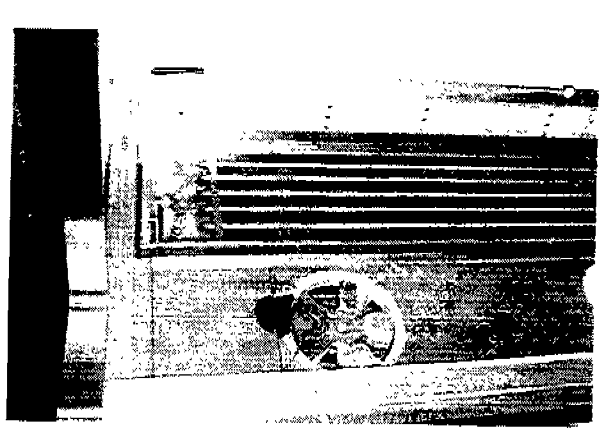

Figure 3: Fan and coil arrangement

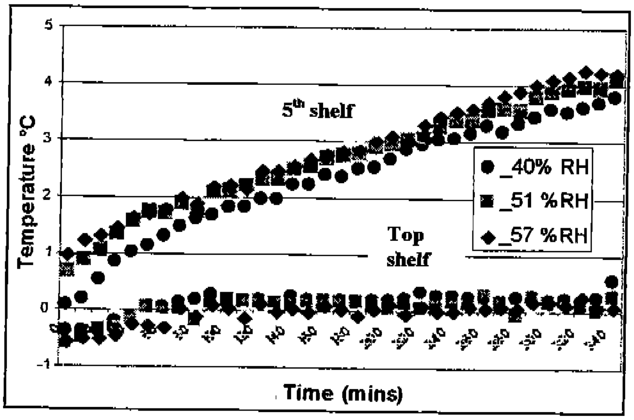

Figure 4: Product Temperature variation for varying relative humidity and constant temperature of 22°C

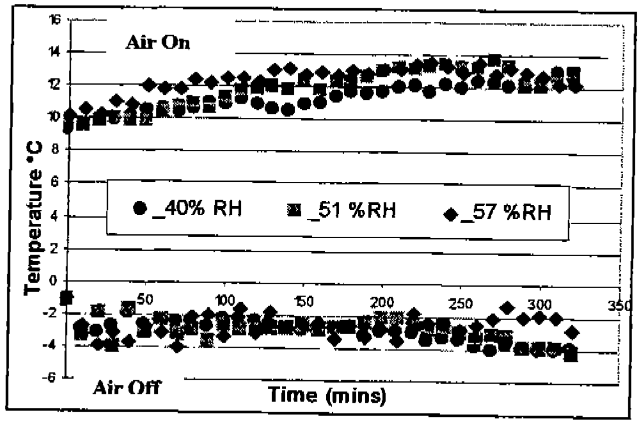

Figure 5: Air Temperature variation for varying relative humidity and constant temperature of 22°C

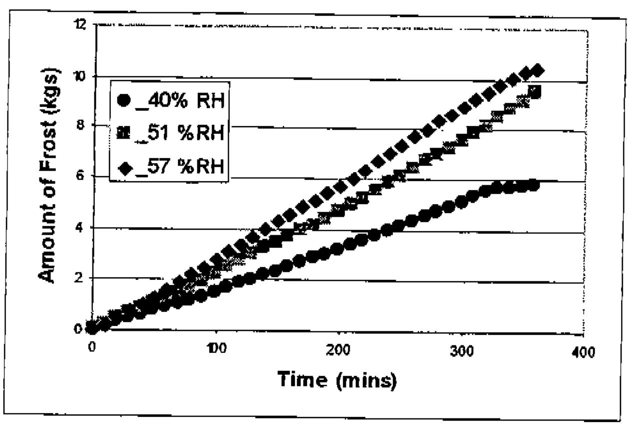

Figure 6: Frost accumulation for varying relative humidity and constant temperature of 22°C

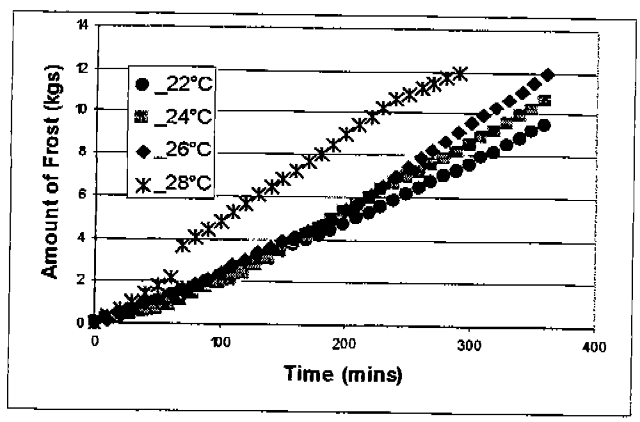

Figure 7: Frost accumulation for varying space temperature for constant relative humidity of 50%RH

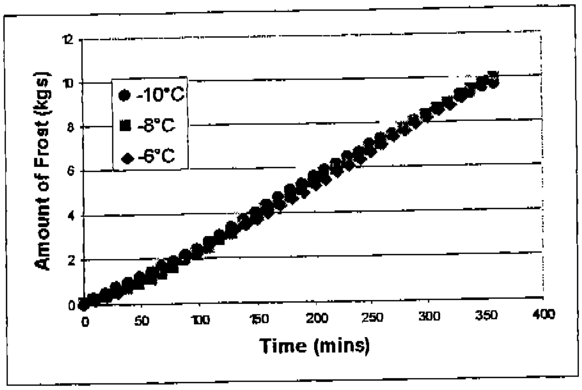

Figure 8: Frost accumulation for varying evaporator refrigerant inlet temperature for constant space conditions of 22°C and 55 %RH

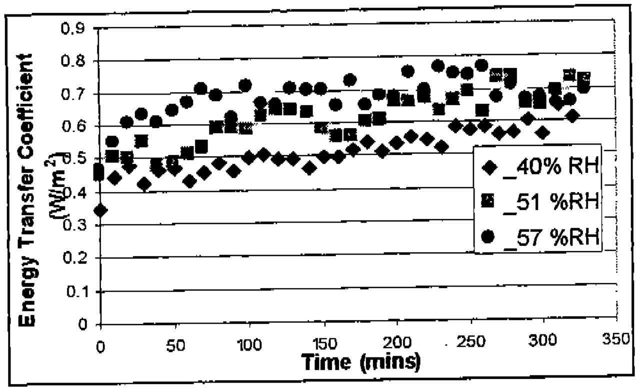

Figure 9: Energy transfer coefficient for varying space humidity for constant space temperature of 22°C

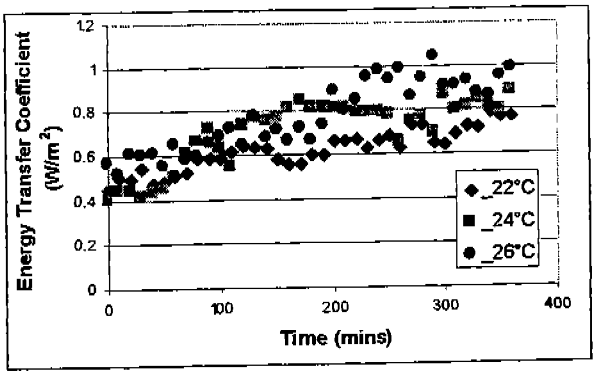

Figure 10: Energy transfer coefficient for varying space temperature for constant humidity of 50%RH

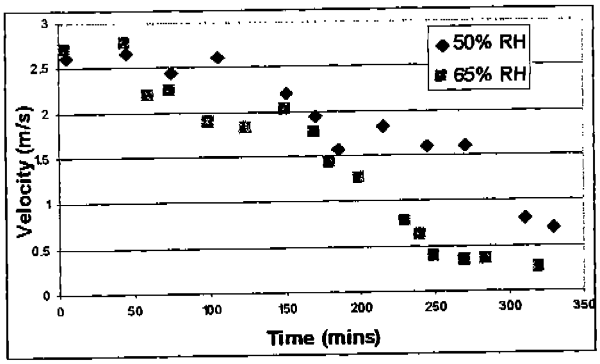

Figure 11: Air off velocity profile for varying space humidity at constant temperature of 22°C

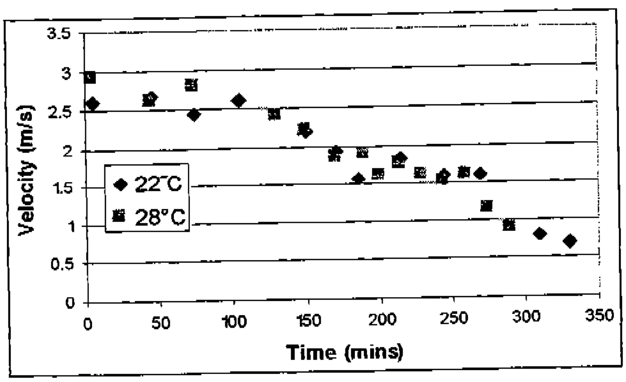

Figure 12: Air off velocity profile for varying space temperature for constant relative humidity of 50% RH

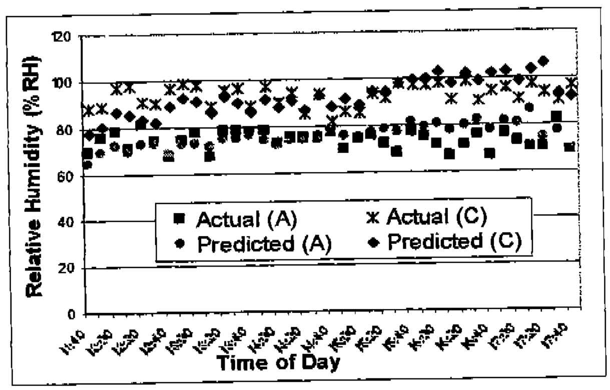

Figure 13: Predicted vs Actual air on humidity for space conditions 40% RH (A) and 60% RH (B) at constant space temperature of 22°C