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The Effect of Water Vapour on Food Refrigeration Systems

by

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(Session 2011-2012)

To be presented before the Institute of Refrigeration at London Chamber of Commerce and Industry, 33 Queen Street, London, EC4R 1AP On Thursday 2nd February 2012 at 5.45pm

Abstract

Air relative humidity (*RH*) significantly affects the storage life of many refrigerated products and the performance of the refrigeration system. This paper describes methods to quantify the main moisture transport processes and predict *RH*, and discusses how water vapour and air *RH* can be optimally managed in refrigerated facilities. Air *RH* is determined by the balance of moisture entry and removal. Quantitative methods to predict the main latent heat loads due to door air infiltration, sorption by packaging, defrost, product weight loss, people and equipment, and dehumidification by the air cooling coils are presented that can be solved iteratively to find *RH* using a standard spreadsheet. A coolstore for horticultural products with a loading dock is analysed as a case study. To increase *RH*, sensible heat loads (such as conduction through insulation, product cooling and fan power) and the air to refrigerant temperature difference should be minimised and latent heat loads maximised (and vice versa to reduce *RH*). Active humidification and dehumidification should only be used as a last resort. Particularly unfavourable frost, that causes very rapid deterioration in cooling coil performance, forms when the air approach line on a psychrometric chart passes through the super-saturated region. The best ways to avoid such frost are to reduce latent loads or lower air to refrigerant temperature differences, but sometimes increased sensible loads or special coil designs are necessary. Avoidance of condensation or icing on surfaces other than the coil is best achieved by breaking thermal bridges or insulating surfaces to increase the surface temperature above the dewpoint, rather than trace heating. Defrost frequency needs to be adjusted dynamically to match latent loads especially if the coil performance is sensitive to frosting, or the defrost-related energy use can become excessive.

Introduction

Water vapour (moisture) concentration in food refrigeration facilities is most commonly quantified in terms of the air relative humidity (*RH*). Air *RH* has significant effects on the design, operation and performance of refrigerated facilities but is often considered in a superficial manner relative to temperature and sensible heat loads.

The storage life of many refrigerated products, particularly those without plastic packaging, is sensitive to air *RH* as well as temperature. In particular, low *RH* increases evaporative moisture loss resulting in quality deterioration as well as the loss of saleable weight, while high *RH* can lead to increased microbial growth (e.g. rots and moulds). High *RH* can also lead to problems such as condensation of water or icing on surfaces and loss of paper-based packaging strength. Additionally, for facilities operating near 0°C, *RH* affects the rate of frost formation on air cooling coils and hence the need for defrost to maintain refrigeration performance. Defrosts result in temperature fluctuations that may contribute to accelerated product quality deterioration.

This paper describes the main moisture transport processes in refrigerated facilities, methods to quantify them, and discusses how water vapour and air *RH* can be optimally managed in the food cold chain.

Moisture Transport

Air temperature is determined by the balance between sensible heat entry and removal rates for a refrigerated facility. Temperature control usually involves modulating the rate of sensible heat removal by the refrigeration system. Similarly, air *RH* is determined by the balance between rates of entry and removal of moisture (latent heat) from the air in a facility [23]. The main mechanisms for moisture entry and removal are:

1. dehumidification (condensation or frosting) by the air cooling coils
2. adsorption or desorption of water from paper-based packaging materials
3. evaporation/condensation of free water from/onto internal surfaces
4. infiltration of warm, moist ambient air through doors
5. sublimation of coil frost during defrost;
6. evaporative moisture loss or transpiration from the product;
7. exhalation and evaporation of moisture from people and equipment
8. active humidification (e.g. water atomisation) or dehumidification (e.g. desiccants).

Mechanisms 1 to 3 generally result in moisture removal while 4 to 7 generally result in moisture entry into the air. For most facilities, the moisture transfer is dominated by 1, 4 and 5, while occasionally 2 and 6 are significant. Mechanisms 3 and 7 are seldom significant compared with the other mechanisms. Active humidification or dehumidification should generally only be employed when other measures do not achieve the desired *RH* conditions, due to the significant capital and operating costs for such systems.

Prediction of *RH*

Prediction of air *RH* for a refrigerated facility is an iterative process that involves:

- Estimation of the sensible and latent heat loads (ϕ_{sen} and ϕ_{lat}) for an assumed value of the *RH*.
- Calculation of cooling coil performance for these heat loads to get a new *RH* estimate assuming that the refrigeration system has sufficient capacity to control temperature at the set-point.
- Repeating the above steps until *RH* estimates converge.

Total and Latent Heat Load Models

The main heat loads for a refrigerated facility are conduction through the insulated surfaces and the floor, air infiltration through doors, product cooling, fans, coil defrost, lighting, people, equipment, sorption by packaging and humidification systems. Methods to estimate total heat loads or sensible heat loads are well-known (e.g. [2]) so the following focuses on methods to estimate the loads with significant latent heat components.

Door Air Infiltration

The total and latent heat load due to air infiltration through doors is given by:

$$\phi_{do} = Q_{do}\,\rho_{a,in}\left(h_{a,out} - h_{a,in}\right) \quad \text{and} \quad \phi_{lat,do} = Q_{do}\,\rho_{a,in}\left(H_{out} - H_{in}\right)L \qquad (1)$$

The average rate of air interchange through doors (Q_{do}) can be estimated by methods described by ASHRAE [2], Chen *et al.* [3] and Cleland *et al.* [5] for forklift doors, East *et al.* [12] for external truck-trailer doors, and East *et al.* [12] for hinged personnel doors. For example, for forklift doors with strip curtain protection, Cleland *et al.* [5] sums the contribution when the door is closed (air-tightness), when it is open without traffic based on the Gosney and Olama [14] correlation for fully developed flow through doors, and the additional air exchange caused by forklift traffic pushing through the curtain:

$$Q_{do} = (1 - F_{do})Q_{at} + F_{do}(1 - E)\,0.221\ W\ dh\ (g\ dh)^{0.5}\left(\frac{2\left(1 - \frac{\rho_{a,out}}{\rho_{a,in}}\right)^{0.333}}{1 + \left[\frac{\rho_{a,in}}{\rho_{a,out}}\right]^{0.333}}\right)^{1.5} + N_{tr}\,0.087\,(T_{out} - T_{in})^{1.76} \tag{2}$$

For sliding or rapid roll doors with seals in good condition, the air-tightness leakage, Q_{at} is about 0.0006 or 0.003 m^3/s per metre of seal length respectively. There is a paucity of information on the effect of forklift traffic on air infiltration through doors without protection, or with air curtain or air lock protection.

Product Weight Loss

Weight loss from product represents a conversion of sensible heat load into latent heat load [7]. For product cooling, the total and latent loads are given by:

$$\phi_{prod} = m_{prod}\, c_{prod}\,(T_{in} - T_{out}) \text{ or } \phi_{prod} = m_{prod}\,(h_{prod,in} - h_{prod,out}) \quad \text{and} \quad \phi_{lat,prod} = m_{ml}\,L \tag{3}$$

The rate of moisture loss from product can be approximated using [7]:

$$m_{ml} = \int k\left(p_{ws}a_w - p_{wa}\, RH/_{100}\right)\partial A \tag{4}$$

The mass transfer coefficient, *k*, includes the effect of any packaging and product skin resistance to mass transfer. It is usually measured experimentally because it can be difficult to estimate from first principles. The product water activity (a_w) and the water vapour pressure at the surface temperature can also be difficult to measure or predict especially because the product surface temperature is often highly variable with position and/or time. For product at storage temperature, the rate of weight loss is proportional to the difference between a_w and the *RH* (as a fraction).

Sorption by Packaging

Paper-based packaging materials are hygroscopic and can represent a significant fraction of the packaged product mass. The total and latent heat loads represented by packaging materials entering the facility can be estimated for each packaging component [8,9]):

$$\phi_{pack} = m_{pack}\, c_{pack}\,(T_{in} - T_{out}) \qquad \text{and} \qquad \phi_{lat,pack} = \frac{m_{pack}\,(X_{in} - X_{out})L}{1 + X_{in}} \tag{5}$$

Note that moisture sorption also represents a conversion of sensible heat into latent heat, or vice versa, as the heat of adsorption/desorption is countered by sensible heat transfer between the packaging and the air by convection (i.e. there is no net heat transfer unless the packaging enters at a different temperature to the facility). The exit moisture content, X_{out}, can be estimated from the fractional approach to equilibrium:

$$X_{out} = EMC + Y\left(X_{in} - EMC\right) \tag{6}$$

For example, for bulk-stacked cardboard cartons on pallets, Figure 1 can be used to estimate the fractional unaccomplished moisture change, *Y* [8]. The equilibrium moisture content, *EMC*, can be estimated using the moisture sorption isotherms based on the air temperature and relative humidity in the refrigerated space. For typical cardboards used to package refrigerated food [8]:

$$EMC = \frac{0.001066\exp\left(498/(T_{pack} + 273.15)\right)RH}{(1 - 0.00961RH)(1 + 0.14289RH)} \tag{7}$$

The equivalent of Eq (7) for wood and specific cardboards, and of Figure 1 for individual cartons plus the equations and data underpinning these diagrams are available [8]. At high moisture content, paper-based packaging loses it mechanical strength, so long periods at high *RH* need to be avoided.

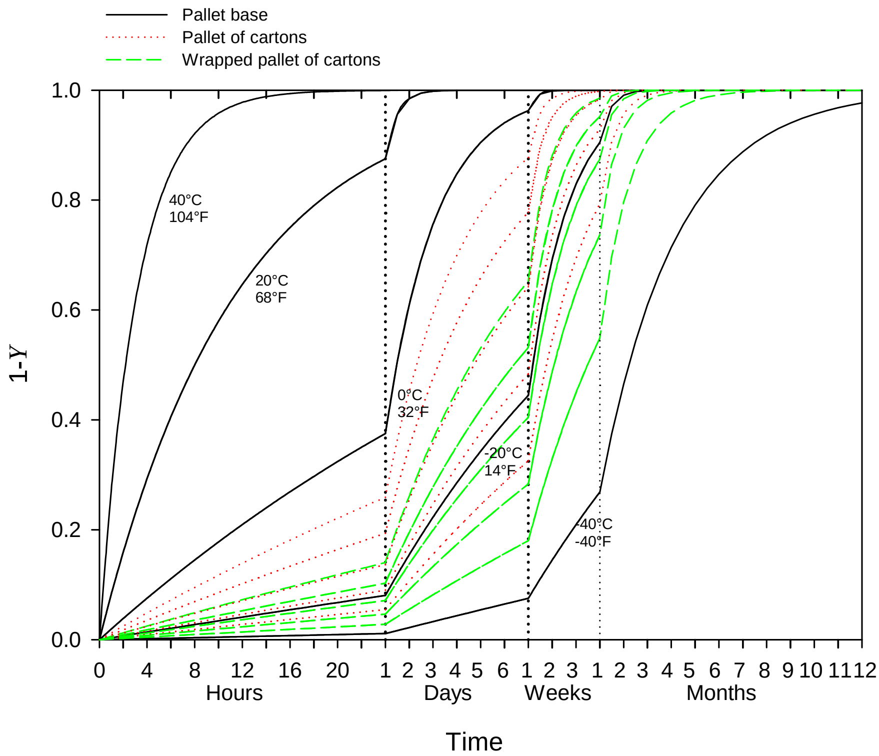

Figure 1: Fractional unaccomplished moisture change (Y) as a function of time and temperature for wooden pallet bases, unwrapped pallets of cartons and wrapped pallets of cartons (all plots are given in the same temperature order as shown for a pallet base) [8].

People

The total heat load due to people in a refrigerated facility depends on the temperature and assuming that 50% of the load is latent heat due to exhaled moisture [2]:

$$\phi_{peo} = N_{peo} \left(272 - 6\, T_{a,on}\right) \qquad \text{and} \qquad \phi_{lat,peo} = 0.5\, \phi_{peo} \tag{8}$$

Equipment

Active humidification/dehumidification systems or other equipment usually add water vapour (m_{hum}) or sensible heat (ϕ_{reh}). The heat loads can be approximated by:

$$\phi_{hum} = m_{hum}\, L + \phi_{reh} \qquad \text{and} \qquad \phi_{lat,hum} = m_{hum}\, L \tag{9}$$

If water enters the air as liquid droplets then it does not contribute to the total load but is a latent load only as it eventually evaporates. Generally, equipment used in refrigerated facilities is electrically driven with no moisture transfer. Shrink-wrapping equipment using hot water can be a source of moisture in refrigerated process areas unless the vapour is ventilated directly outside.

Cooling Coil Defrost

During defrost, cooling coils contribute both sensible and latent heat load because not all of the supplied heat melts frost that drains away. Assuming that defrost has a fixed efficiency [11], that all of the net heat load enters the air space, and that the sensible heat ratio (*SHR*) is constant and equal to that for the non-defrost loads, the average defrost heat load and the latent component can be estimated using [10]:

$$\phi_{def} = \frac{0.13\left(1-\eta_{def}\right)\phi_{lat\,excl.def}\,\phi_{tot\,excl.def}}{\eta_{def}\,\phi_{tot\,excl.def} - 0.13\left(1-\eta_{def}\right)\phi_{lat\,excl.def}} \qquad \text{and} \qquad \phi_{lat,def} = \phi_{def}\,\frac{\phi_{lat\,excl.def}}{\phi_{tot\,excl.def}} \tag{10}$$

Surface Deposition and Evaporation/Sublimation

The equivalent of Eq. (4) can be used to estimate the rate of evaporation/sublimation of water from or condensation/frosting of water onto other surfaces within the refrigerated facility. The challenges are to estimate the temperature and mass transfer coefficient, *k*, for such surfaces.

Total Heat Loads
Having considered the various contributions to heat and moisture loads, the overall latent, sensible and total heat loads that must be removed by the air-cooling coil and the load *SHR* can be estimated from:

$$\phi_{tot} = \phi_{surf} + \phi_{light} + \phi_{peo} + \phi_{equip} + \phi_{do} + \phi_{pack} + \phi_{prod} + \phi_{fan} + \phi_{hum} + \phi_{def} \tag{11}$$

$$\phi_{lat} = \phi_{lat,peo} + \phi_{lat,do} + \phi_{lat,pack} + \phi_{lat,prod} + \phi_{lat,hum} + \phi_{lat,def} \tag{12}$$

$$\phi_{sen} = \phi_{tot} - \phi_{lat} \quad \text{and} \quad SHR = \frac{\phi_{sen}}{\phi_{tot}} = \frac{\phi_{tot} - \phi_{lat}}{\phi_{tot}} \tag{13}$$

Air Cooling Coil Performance

In cooling mode, if the coil operates with on/off control with a fixed evaporation temperature, the fractional operating time (F_{on}) is defined by:

$$F_{on} = \frac{\phi_{sen}}{UA(T_{on} - T_e)} \tag{14}$$

If temperature control is an evaporator pressure regulator (EPR) valve then F_{on}=1 and Eq. (14) is used to calculate the refrigeration evaporation temperature, T_e. For a given air-on temperature (T_{on}), the air-off temperature (T_{off}) is given by:

$$T_{off} = T_{on} - \frac{\phi_{sen}}{F_{on}\, Q_a\, \rho_a\, c_a} \tag{15}$$

The absolute humidity of the air coming onto the coil (H_{on}) can be related to the coil surface temperature (T_s), T_{on}, T_{off} and the latent load by assuming a straight line approach from the air-on condition to saturation at the coil surface as shown in Figure 2 [21]:

$$H_{on} = H_{ws} + \frac{\phi_{lat}}{Q_a\, \rho_a\, L} \frac{(T_{on} - T_s)}{(T_{on} - T_{off})} \tag{16}$$

Based on an estimate of the ratio of the refrigerant-side to the overall heat transfer resistance for the coil, R, T_s is calculated from:

$$T_s = T_e + R(T_{on} - T_e) \tag{17}$$

The air-on RH can be calculated from T_{on} and H_{on} using normal psychrometric relationships:

$$p = \frac{2937700H}{29H + 18} \qquad H = \frac{18p}{29 \cdot (101300 - p)} \qquad RH = \frac{p}{p_w} \cdot 100 \tag{18}$$

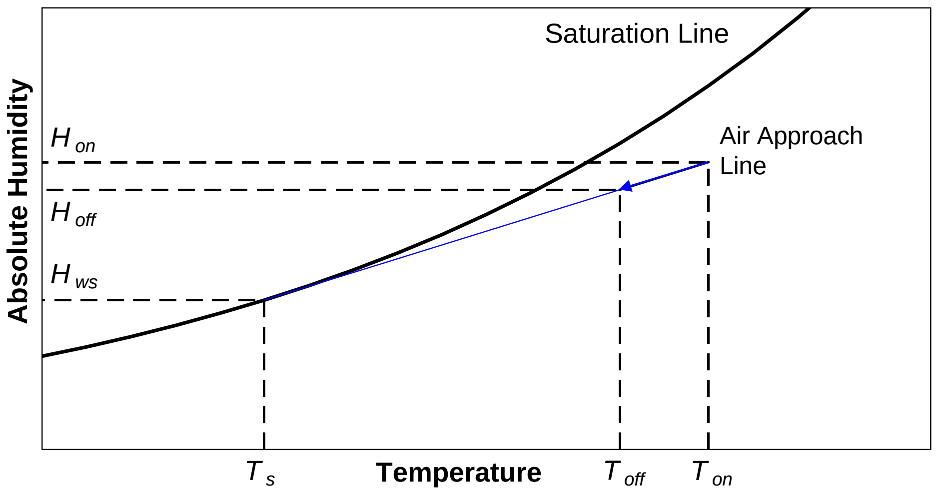

Figure 2: Psychrometric chart showing the concept of the straight line air approach to the saturation condition at the cooling coil surface temperature [21].

$$p_w = e^{\left(28.7775 - \frac{6071.67}{(T\ +\ 271.511)}\right)} \text{ for } T \leq 0°C \qquad p_w = e^{\left(23.4795 - \frac{3990.56}{(T\ +\ 233.833)}\right)} \text{ for } T > 0°C \qquad (19)$$

The above methods have been shown to accurately predict *RH* in both a large horticultural coolstore and a small walk-in cold room under a wide range of operating conditions [1, 10, 22].

Factors Affecting Air *RH*

The appendix provides an example of use of the above methods to find *RH* for a coolstore at 0°C. Only the first iteration is shown. Table 1 gives the heat loads for the ultimate balanced *RH* of 85.9% for day-time operation.

Table 2 gives predicted heat loads, evaporation temperature, *SHR* and balanced air *RH* for various independent scenarios based on the case study. For example, at night when heat loads are significantly lower, the evaporation temperature and *RH* are significantly higher than during the day (scenario 1). The main design and operating strategies to control *RH* that can be deduced from these scenarios are:

1. Minimise sensible heat loads if high *RH* is desired (scenarios 1 to 4). Key design aspects are increased insulation levels and reduced lighting and fan power. Operational aspects include reducing fan speed when heat loads are low and turning off lights when practicable. For the case study, reducing fan speed was predicted to slightly reduce *RH* due to a commensurate decline in air flowrate and coil performance. However, if the same air flowrate and coil performance can be achieved at lower fan power then *RH* will increase.
2. Minimise latent heat loads if low *RH* is desired (scenarios 5 and 6). Key design aspects are protection of external doors, reduced air flow over unpackaged stored product and use of moisture impermeable packaging for product to reduce weight loss. An operational aspect is management of external doors to minimise openings. For scenario 6, the *RH* increased slightly when door openings reduced because the *SHR* for the infiltration load was higher than for other heat loads. This unusually low overall *SHR* was due to the high product weight loss. Generally, reduced door loads reduce *RH*.
3. Maximise the active coil surface area (minimise air to refrigerant temperature difference) if high *RH* is desired (opposite of scenarios 7 and 8). Key design aspects are coil size and refrigerant supply method (e.g. flooded coils and coils using secondary refrigerants rather than direct expansion coils). An important operational aspect is the superheat setting for direct expansion coils because the air to refrigerant temperature difference cannot normally be lower than the superheat at the evaporator exit [15]. Also, it is important to choose air temperature control systems that keep the cooling coil fully wetted e.g. EPR rather than on/off temperature control.

	ϕ_{sen} (kW)	ϕ_{lat} (kW)	ϕ_{tot} (kW)
Insulation	10.94	0.00	10.94
Lighting	4.00	0.00	4.00
People	0.14	0.14	0.27
Equipment	2.00	0.00	2.00
Air Infiltration	9.15	4.64	13.80
Product	17.07	17.65	34.72
Packaging	5.51	-5.19	0.33
Fans	6.00	0.00	6.00
Defrost	4.29	1.35	5.64
TOTAL	59.11	18.58	77.69
Balanced Air *RH* (%)			85.9
Sensible Heat Ratio (*SHR*)			0.761

Table 1: Estimated heat loads for the coolstore during daytime operating hours with 20°C ambient and balanced air RH.

Scenario	T_e (°C)	*SHR*	ϕ_{tot} (kW)	Balanced Air *RH* (%)
Base Case – Daytime Operation	-4.9	0.76	77.7	85.9
(1) Night-time Operation (10°C ambient)	-3.5	0.78	53.6	88.4
(2) No product cooling in coolstore	-2.6	0.81	39.0	89.8
(3) Floor insulation (100 mm)	-4.7	0.76	75.2	86.5
(4) Reduced fan speed (70%)	-5.7	0.75	73.8	85.7
(5) Halved weight loss e.g. plastic liner	-5.3	0.84	76.1	79.8
(6) Reduced door opening (1 hour per day)	-4.5	0.77	70.6	86.1
(7) Smaller cooling coil (60% of base size)	-8.0	0.74	78.5	83.3
(8) On/off control with T_e = -10°C	-10.0	0.73	78.7	82.6
(9) Increased cardboard packaging (double)	-5.1	0.79	77.4	83.2

Table 2: Predicted heat loads and balanced air RH for various independent scenarios based on the case study coolstore.

4. Selection and equilibration of packaging materials (scenario 9). If large amounts of paper-based packaging pass through a coolstore, then air *RH* where the packaging was stored before entering the cool store can influence cool store *RH*. If the previous *RH* is lower than the cool store *RH* then moisture adsorption by the packaging will act to lower the *RH*, whereas if the previous *RH* is higher than the store *RH* then the packaging will be a source of moisture acting to raise *RH*.

Figure 3 shows measured air *RH* for a large apple coolstore operating with EPR control [1]. Consistent with the above, the *RH* is low early in the season when sensible loads are high (warm ambient, product being precooled, high fan power) and packaging is absorbing moisture, although this is partially offset by high door infiltration loads. Later in the season, *RH* increases as sensible loads decrease and packaging approaches equilibrium moisture levels, even although door use and air infiltration load decrease.

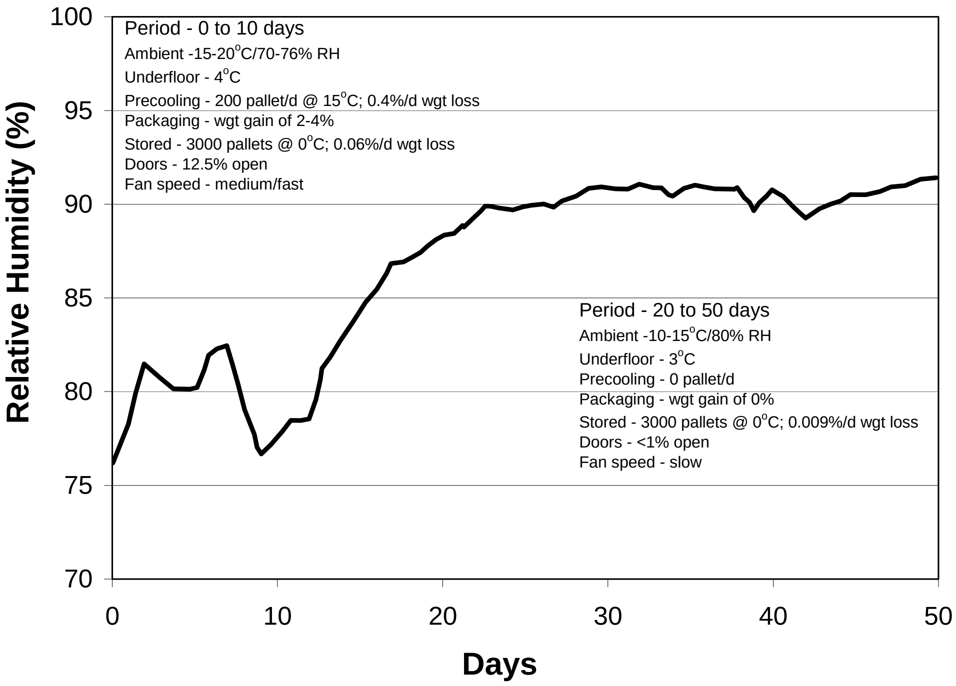

Figure 3: Measured air RH for a large apple coolstore precooling fruit early in the season[1].

Moisture Related Issues

In addition to the effect of *RH* on weight loss, packaging moisture sorption and coil frosting considered above, there are a number of other moisture related issues that are common in refrigerated facilities.

Surface Condensation/Frosting

Condensation/frosting of water vapour will occur if a surface is below the dew-point temperature of the air. Such deposition is undesirable because it can be a source of microbiological growth, or it can be unsightly or a safety hazard e.g. slippery floor.

Except for the cooling coil which is designed for condensation/frosting, most surfaces inside a refrigerated facility are similar to or slightly warmer than the air temperature so condensation/frosting is only probable if the local *RH* becomes extremely high or super-saturated. This is only likely near doorways or if hot unwrapped product is loaded into the facility. Protection of doorways, minimising open times and pre-cooling or wrapping product are obvious solutions, which also reduce the overall refrigeration load.

The other common situation where condensation/icing can occur is in a refrigerated space adjacent to a colder space. Conduction through structural elements (thermal bridges) to the colder space can cool a surface below the air dewpoint in the warmer space. Examples include door frames in loading docks outside coldstores (conduction to the inside through the metal frame) and air locks retrofitted inside rather than outside a coldstore (conduction through the concrete floor if base isolation not adequate). A similar example is deposition on the bottom of defrost drain trays due to conduction through hanging brackets to the cooling coil. The simplistic solution is to heat the surface using trace heating but this increases energy costs. The preferable approach is to break the thermal bridge (e.g. plastic inserts in metal conduction paths) or to insulate the surface so the surface approaches the air temperature and becomes above the dewpoint.

Unfavourable Frosting
While all frost is undesirable, under certain conditions particularly unfavourable frost can form on air cooling coils [19,20]. Unfavourable frost has lower density than more "favourable" frost types, is more insulating, has higher impact on air flowrate through the coil and can be more difficult to defrost. Smith [19,20] proposed that unfavourable frost occurs when the air approach line crosses the saturation line of the psychrometric chart (i.e. becomes super-saturated). This corresponds to an air approach line above that shown in Figure 2 (higher air-on *RH; lower SHR*) or a lower coil surface temperature for the same air-on condition. Note that this does not necessarily mean that the coil air-off condition is super-saturated; rather that some part of the full approach line crosses the saturation line.

The concept of unfavourable frost and the supersaturated transition was confirmed experimentally by O'Hagan *et al.* [16,17] and Sherif *et al.* [18]. The mechanism of unfavourable frost formation was postulated to be the formation of airborne water droplets or ice-crystals, which physically deposit on any surface they encounter, in addition to the normal frost formation by diffusion of water from high water vapour partial pressure conditions in the air stream to low water vapour partial pressures conditions at the coil (or frost) surface.

Based on the supersaturated theory, the demarcation between frost types can be defined as an air approach line at a tangent to the air saturation line (Figure 2). For a given coil surface temperature, the slope of the coil transitional air approach line on the psychrometric chart is approximated by [4]:

$$\frac{H_{on,crit} - H_{ws,crit}}{T_{on} - T_{s,crit}} = \frac{1.1708\times10^{11}\cdot\exp\left(\frac{6071.67}{T_{s,crit}+271.511}\right)\left(T_{s,crit}+271.511\right)^{-2}}{\left(\exp\left(\frac{6071.67}{T_{s,crit}+271.511}\right)-3.106705\times10^{7}\right)^{2}} \qquad (20)$$

This allows the critical H_{on} or T_s and consequently the critical *SHR* for onset of unfavourable frosting to be calculated. For example, for the case study in the Appendix, the critical coil surface temperature is -8.1°C (corresponding to a refrigerant evaporation temperature of -10.8°C) and the critical air-on *RH* is 96.5% so the existing coil (T_e of -4.9°C; RH_{on} of 85.9%) will not operate unfavourably. For facilities such as blast freezers operating at very low temperatures, it can be very difficult to avoid unfavourable frosting conditions, even if latent loads due to air infiltration and product weight loss are low. In such cases, special coil designs to cope with unfavourable frosting are necessary (e.g. large fin spacing or physical removal of frost by air knives).

The concept of unfavourable frost explains the following situations observed in some facilities [6]:

- Coils frost up almost immediately after defrost and effectively remain blocked with frost most of the time (e.g. coils in air locks or above doors)
- The whole refrigerated air space becomes foggy (air-off condition is supersaturated)
- Addition of sensible heat (e.g. air heaters in doorways) can surprisingly result in significantly lower air temperature in the refrigerated space (due to better coil performance if unfavourable frosting is avoided)
- Lowering the refrigerant suction pressure setpoint in order to lower the air temperature unexpectantly leads to higher air temperature (or conversely raising refrigerant suction pressure leads to lower air temperature) due to the onset of unfavourable frosting and loss of coil performance.

Insulation Saturation
Another possible effect of water vapour in refrigerated facilities is saturation of insulation with inadequate vapour barriers on the outside. Without a complete vapour barrier, water vapour in the ambient air can enter the insulation until it reaches a position in the insulation below the dewpoint of

the air. At this point, water vapour will condense out of the air saturating the insulation and causing deterioration of its insulation value. If the water reaches part of the insulation below 0°C it will freeze. The saturated insulation is heavier and, particularly if freezing expansion occurs, the insulation can be physically damaged and may lose its structural strength. The solution is to carefully install and keep vapour barriers intact.

Defrost Frequency
For refrigerated facilities operating at about 0°C or lower, then coil frosting is inevitable. Defrosting is essential to maintain coil performance but defrosting too frequently increases defrost heat load and causes greater fluctuations in temperature due to the loss of cooling capacity while defrosting. Figure 4 shows measured defrost efficiency and duration as a function of defrost frequency for a small walk-in coolstore with constant latent load [24]. In general, the optimum defrost frequency increases as the latent load increases and if the rate of deterioration in coil performance is large, but is difficult to quantify in a generic manner.

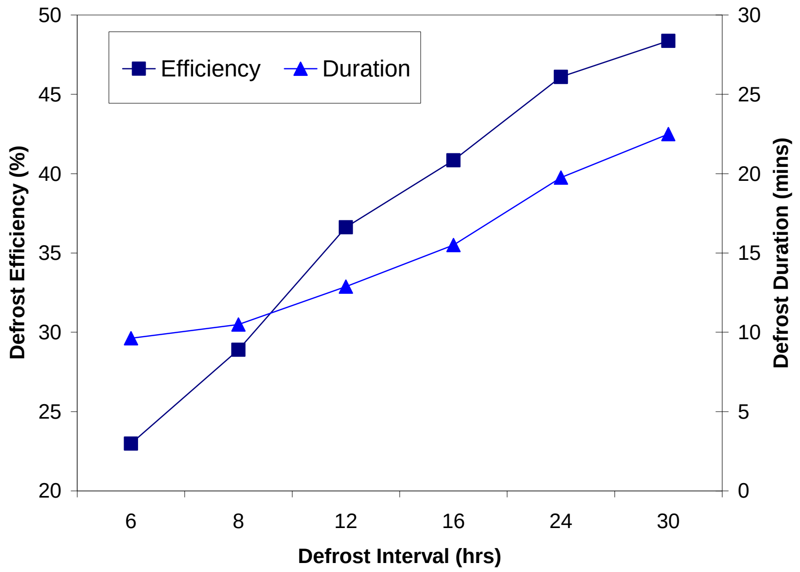

Figure 4: Defrost efficiency and duration as a function of defrost frequency for a small walk-in coolstore [26].

Conclusions
The above methodologies allow latent heat loads and balanced air *RH* to be predicted for a refrigerated facility. Important ways to influence *RH* are: control of sensible heat entry, management of air infiltration rates, design and operation of cooling coils with appropriate air-to-refrigerant temperature difference, choice of temperature control systems, and selection and equilibration of packaging materials. Optimisation of air *RH* by such methods allows cold chain improvement without the unnecessary expense of dedicated humidity control or advanced packaging systems. Avoidance of condensation or icing on surfaces other than the coil is best achieved by breaking thermal bridges or insulation of surfaces rather than trace heating. Poor coil design or extreme latent loads can lead to particularly unfavourable frosting of air cooling coils. The criteria for onset of this phenomenon have been defined. Preferred ways to avoid unfavourable frost are to reduce latent loads or lower air to refrigerant temperature differences, but sometimes increased sensible loads or special coil design are necessary. Defrost frequency needs to be adjusted dynamically to match latent loads, especially if the coil performance is sensitive to frosting, in order to avoid poor refrigeration performance.

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Nomenclature

A	area (m^2)
a_w	product water activity
c	specific heat capacity (kJ/kg·K)
dh	door height (m)
E	door protection effectiveness
EMC	equilibrium moisture content (kg/kg dry basis)
exp	the function e^x
F	fraction of time in use or open
h	enthalpy (kJ/kg or kJ/kg dry air)
H	air absolute humidity; humidity ratio (kg/kg dry air)
k	mass transfer coefficient (kg/m^2·Pa·s)
L	latent heat of condensation (and freezing if below 0°C) (kJ/kg)
m	mass flowrate (kg/s)
N	number (s^{-1})
p	pressure of water vapour (Pa)
Q	air volumetric flowrate (m^3/s)
R	ratio of air-side to overall heat transfer resistance for the cooling coil
RH	air relative humidity (%)
SHR	sensible heat ratio
T	temperature (°C)
UA	coil sensible heat transfer capacity rating (kW/°C)
W	width of door (m)
X	packaging moisture content (kg/kg dry basis)
Y	fractional unaccomplished moisture change
ϕ	heat load; total unless indicated otherwise (kW)
ρ	density (kg/m^3)
η	efficiency

Subscripts

a	air
at	door air-tightness
$crit$	critical value at unfavourable transition
def	defrost
do	door
e	refrigerant evaporation
$equip$	equipment
fan	fan
hum	humidification
in	inlet or inside condition
ml	mass loss
lat	latent heat
$light$	lights
off	coil air-off condition
on	coil air-on condition or coil operation
out	outlet or outside condition
$pack$	packaging
peo	people
$prod$	product
reh	reheat
s	coil surface
sen	sensible heat
$surf$	insulated surfaces
tr	traffic
tot	total
w	water vapour saturation

Appendix – Prediction of *RH* Case Study

Coolstore Description

A coolstore for horticultural products operates at 0°C, is 6 m high, 12 m wide and 33 m long, holds 700 tonnes of product, and is constructed from 100 mm polystyrene sandwich panel. The floor is uninsulated and the deep ground (2 m deep) temperature is 15°C. The cool store has 4 kW of lights, 6 kW of fans that circulate the air at 24 m³/s, and the air cooling coils have a combined sensible rating of 12 kW/°C and use an evaporator pressure regulator (EPR) for temperature control. The coolstore operates for 12 hours each day from 6 am to 6 pm. There is a 2.5 m wide by 3.5 m high rapid-roll door between the cool store and a packing area at 15°C and 60% *RH* that is open 2 hours each day in order to load-in 50 tonnes of product (specific heat capacity of 4 kJ/kg K) at 15°C that is cooled to 0°C before dispatch. The average weight loss is 1% during cooling and 0.5% per month thereafter if the *RH* is 85%.(assume a_w is 1.0). Each tonne of product has 25 kg of cardboard packaging associated with it. On average during work hours there are one person and one forklift (average output 2 kW) in the cool store. The typical ambient temperature during normal daytime operation is 20°C.

Heat Loads

The heat loads for the coolstore for the daytime operation are estimated assuming that the *RH* is 85%. The insulation, lighting, equipment and fans loads are sensible only, are unaffected by *RH*, can be estimated using standard methods [2], and are given in Table 1.

Door infiltration is estimated using Eq (1) and (2). The last term in Eq (2) does not apply because the door does not have strip curtain protection (E = 0). During work hours, F_{do} = 2/12 = 0.167 and the door seal length is 2×(2.5+3.5) = 12 m. Using Eq (2) to calculate the average air infiltration rate and then Eq (1) and taking psychrometric properties from standard charts or programs:

for air at 15°C and 60% *RH*: ρ_{out} = 1.223 kg/m³, h_{out} = 131.2 kJ/kg, H_{out} = 0.00634 kg/kg
for air at 0°C and 85% *RH*: ρ_{in} = 1.292 kg/m³, h_{in} = 108.0 kJ/kg, H_{in} = 0.00319 kg/kg

$$Q_{do} = (1-F_{do})Q_{at} + F_{do}(1-E)\,0.221\,W\,dh\,(g\,dh)^{0.5}\left(\frac{2\left(1-\frac{\rho_{a,out}}{\rho_{a,in}}\right)^{0.333}}{1+\left[\rho_{a,in}/\rho_{a,out}\right]^{0.333}}\right)^{1.5}$$

$$= (1-0.167)\times 0.003\times 12 + 0.167\times(1-0)\times 0.221\times 2.5\times 3.5\times(9.81\times 3.5)^{0.5}\left(\frac{2\times\left(1-\frac{1.223}{1.292}\right)^{0.333}}{1+\left(1.292/1.223\right)^{0.333}}\right)^{1.5}$$

$$= 0.03 + 0.432 = 0.462\ \mathrm{m^3/s}$$

$$\phi_{do} = Q_{do}\,\rho_{a,in}\,(h_{a,out} - h_{a,in}) = 0.462\times 1.292\times(131.2-108.0) = 13.85\ \mathrm{kW}$$

$$\phi_{lat,do} = Q_{do}\,\rho_{a,in}\,(H_{out} - H_{in})L = 0.462\times 1.292\times(0.00634-0.00319)\times 2500 = 4.70\ \mathrm{kW}$$

Heat loads are calculated for the stored and entering product separately on a daily basis due to the slow heat transfer rate. For the entering product, m_{prod} = 50,000/(24×3600) = 0.579 kg/s and c_{prod} = 4.0 kJ/kg K, and the rate of weight loss, m_{ml} = (50,000×0.01)/(24×3600) = 0.0058 kg/s. Using Eq. (3) the heat loads are:

$$\phi_{prod} = m_{prod}\,c_{prod}\,(T_{in} - T_{out}) = 0.579\times 4\times(15-0) = 34.72\ \mathrm{kW}$$

$$\phi_{lat,prod} = m_{ml}\,L = 0.0058\times 2500 = 14.47\ \mathrm{kW}$$

For the stored product, m_{prod} = 0 kg/s, m_{ml} = (700,000×0.005)/(30×24×3600) = 0.00135 kg/s so:

$$\phi_{prod} = 0\ \mathrm{kW} \qquad \phi_{lat,prod} = m_{ml}\,L = 0.00135\times 2500 = 3.38\ \mathrm{kW}$$

For the combined product excluding packaging:

$$\phi_{prod} = 34.72 + 0 = 34.72\ \mathrm{kW} \qquad \phi_{lat,prod} = 14.47+3.38 = 17.84\ \mathrm{kW} \qquad \phi_{sen,pack} = 34.72 - 17.84 = 16.88\ \mathrm{kW}$$

The packaging enters equilibrated at 15°C and 60% *RH* and as there is no information on how long it stays, it will be assumed that it leaves equilibrated in the air at 0°C and 85% *RH* (i.e. Y = 0 and X_{out} = EMC from Eq (6)). The heat load is spread out over the whole day so for cardboard the effective rate of entry, m_{pack} = (50×25)/(24×3600) = 0.0145 kg/s and c_{pack} = 1.5 kJ/kg K. Using Eq (7) to find X_{in} and EMC and then Eq. (5):

$$X_{in} = \frac{0.001066\exp(498/(T_{pack}+273.15))RH}{(1-0.00961RH)(1+0.14289RH)} = \frac{0.001066\times\exp(498/(15+273.15))\times 60}{(1-0.00961\times 60)(1+0.14289\times 60)} = 0.089$$

$$X_{out} = EMC = \frac{0.001066\exp(498/(T_{pack}+273.15))RH}{(1-0.00961RH)(1+0.14289RH)} = \frac{0.001066\times\exp(498/(0+273.15))\times 85}{(1-0.00961\times 85)(1+0.14289\times 85)} = 0.233$$

$$\phi_{pack} = m_{pack}\, c_{pack}\,(T_{in} - T_{out}) = 0.0145 \times 1.5 \times (15 - 0) = 0.33\,\text{kW}$$

$$\phi_{lat,pack} = \frac{m_{pack}\,(X_{in} - X_{out})\,L}{1 + X_{in}} = \frac{0.0145 \times (0.089 - 0.233) \times 2500}{1 + 0.089} = -4.79\,\text{kW}$$

Hence, from Eq. (13): $\phi_{sen,pack} = \phi_{tot,pack} - \phi_{lat,pack} = 0.33 - -4.79 = 5.11\,\text{kW}$

The personnel load for N_{peo} = 1 is estimated using Eq. (8):

$$\phi_{peo} = N_{peo}\,(272 - 6\,T_{a,on}) = 1 \times (272 - 6 \times 0) = 272\ \text{W} = 0.27\,\text{kW}$$

$$\phi_{lat,peo} = 0.5\,\phi_{peo} = 0.5 \times 272 = 136\ \text{W} = 0.14\,\text{kW}$$

No active dehumidification or humidification equipment is used in the coolstore so using Eq. (9):

$\phi_{hum} = 0\,\text{kW}$ $\qquad\qquad$ $\phi_{lat,hum} = 0\,\text{kW}$

Assuming that η_{def} = 0.3, the defrost heat loads are estimated using Eq. (10) to (12):

$$\phi_{tot\ excl.\ def} = 10.94 + 4.0 + 0.27 + 2.0 + 13.85 + 0.33 + 34.72 + 6.0 + 0 = 72.11\,\text{kW}$$

$$\phi_{lat\ excl.\ def} = 0.14 + 4.70 - 4.79 + 17.84 + 0 = 17.89\,\text{kW}$$

$$\phi_{def} = \frac{0.13\,(1 - \eta_{def})\,\phi_{lat\ excl.\ def}\,\phi_{tot\ excl.\ def}}{\eta_{def}\,\phi_{tot\ excl.\ def} - 0.13\,(1 - \eta_{def})\,\phi_{lat\ excl.\ def}} = \frac{0.13 \times (1 - 0.3) \times 17.89 \times 72.11}{0.3 \times 72.11 - 0.13 \times (1 - 0.3) \times 17.89} = 5.87\,\text{kW}$$

$$\phi_{lat,def} = \phi_{def}\,\frac{\phi_{lat\ excl.\ def}}{\phi_{tot\ excl.\ def}} = 5.87 \times \frac{17.89}{72.11} = 1.46\,\text{kW}$$

The overall heat loads and the sensible heat ratio are given by Eq.(11) to (13):

$\phi_{tot} = 72.11 + 5.87 = 77.97\,\text{kW}$ $\qquad\qquad$ $\phi_{lat} = 17.89 + 1.46 = 19.35\,\text{kW}$

$\phi_{sen} = 77.97 - 19.35 = 58.63\,\text{kW}$ $\qquad\qquad$ $SHR = \dfrac{58.63}{77.97} = 0.752$

Cooling Coil Performance

The cooling coil uses EPR temperature control, so F_{on} = 1 and Eq.(14) can be used to estimate the refrigerant evaporation temperature required to balance the sensible heat load with UA = 12 kW/°C:

$$T_e = T_{on} - \frac{F_{on}\phi_{sen}}{UA} = 0 - \frac{1 \times 58.63}{12} = -4.89^\circ\text{C}$$

The air-off temperature (T_{off}) is given by Eq.(15) given Q_a=24 m³/s and c_a=1.005 kJ/kg K:

$$T_{off} = T_{on} - \frac{\phi_{sen}}{F_{on}\,Q_a\,\rho_a\,c_a} = 0 - \frac{58.63}{1 \times 24 \times 1.29 \times 1.005} = -1.88^\circ\text{C}$$

Eq.(16) to (19) are used to estimate T_s, H_s and a new H_{on} assuming R = 0.25 (a typical value):

$$T_s = T_e + R\,(T_{on} - T_e) = -4.89 + 0.25 \times (0 - -4.89) = -3.66^\circ\text{C}$$

$$p_{ws} = e^{\left(28.7775 - \frac{6071.67}{(T_s\ +\ 271.511)}\right)} = e^{\left(28.7775 - \frac{6071.67}{(-3.66\ +\ 271.511)}\right)} = 449.9\ \text{Pa}$$

$$H_{ws} = \frac{18\,p_{ws}}{29 \cdot (101300 - p_{ws})} = \frac{18 \times 449.9}{29 \times (101300 - 449.9)} = 0.00277\ \text{kg/kg}$$

$$H_{on} = H_{ws} + \frac{\phi_{lat}}{Q_a\,\rho_a\,L}\,\frac{(T_{on} - T_s)}{(T_{on} - T_{off})} = 0.00277 + \frac{19.35}{24 \times 1.292 \times 2500} \times \frac{(0 - -3.66)}{(0 - -1.88)} = 0.00326\,\text{kg/kg}$$

The new estimate of air-on RH can be calculated from Eq. (16) and (17):

$$p = \frac{2937700 H_{on}}{29 H_{on} + 18} = \frac{2937700 \times 0.00326}{29 \times 0.00326 + 18} = 528.5\,\text{Pa}$$

$$p_w = e^{\left(28.7775 - \frac{6071.67}{(T_{on}\ +\ 271.511)}\right)} = e^{\left(28.7775 - \frac{6071.67}{(0\ +\ 271.511)}\right)} = 610.1\,\text{Pa} \qquad RH_{on} = \frac{p}{p_w} \cdot 100 = \frac{528.5}{610.1} \times 100 = 86.6\%$$

Iteration Summary

This result is slightly different from the initial guess of 85% so the heat load and cooling coil calculations should be repeated using a RH between 85% and 86.6% as the next estimate of the air RH, until agreement between sequential estimates is achieved. Table 1 gives the heat loads for the ultimate balanced RH of 85.9%. Note that the loads are only slightly different to those above because the first guess of 85% was very close to the balanced RH and the effect of RH on heat loads is small.