An active defrost scheme with a balanced energy consumption and food quality loss in supermarket refrigeration systems

Cai, Junping; Stoustrup, Jakob; Rasmussen, Bjarne Dindler

Published in:
Elsevier IFAC Publications / IFAC Proceedings series

Publication date:
2008

Document Version
Early version, also known as pre-print

Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

An Active Defrost Scheme with a Balanced Energy Consumption and Food Quality Loss in Supermarket Refrigeration Systems

J. Cai, J. Stoustrup * **B. D. Rasmussen** **

* *Automation and Control, Department of Electronic Systems, Aalborg University, 9220 Aalborg, Denmark. (e-mail: jc@es.aau.dk, jakob@es.aau.dk)*
** *Business Unit Industry and Water Services, Grundfos Management A/S, 8850 Bjerringbro, Denmark. (e-mail: brasmussen@grundfos.com)*

Abstract: This paper introduces food quality as a new parameter, together with energy, to determine an optimal cooling time between defrost cycles. A new defrost-on-demand scheme is proposed. It uses a feedback loop consisting of on-line model updating and estimation as well as a model based optimization. This scheme automatically adjusts the time interval between defrost cycles with varying operating conditions, continuously seeking an optimal time interval, featuring either an energy optimal time, or a trade-off between energy consumption and food quality loss. This adaptive approach is compared with traditional defrost schemes, found to be able to reduce energy consumption significantly.

1. INTRODUCTION

Supermarkets are one of the most energy-intensive types of commercial buildings. Refrigeration is the largest component of their electric energy usage, accounting for half or more of the store total. Among others, energy associated with defrosts and anti-sweat heaters in supermarkets may exceed 30% of the total energy requirement, see Howell et al. [1999]. Thus, from an energy point of view, determining how often to defrost is an important issue. From a food quality point of view, it is also important. During defrosting, the air temperature inside a display cabinet will normally increase, and so will the food temperature. Depending on the defrost method, the food temperature will stay out of the usually controlled range for a period of time, which is harmful to the food quality, see Cai et al. [2006].

Defrost approaches can be classified in two major schemes: one is scheduled defrost, another is defrost-on-demand. Fahlen [1996] studied and compared these two schemes, and found that both of them have some excellent features, but various limitations and drawbacks. Food quality has never been used as an active decision factor in either of these schemes.

Biotechnology provides us the knowledge of food quality change during the refrigerated storage. Research on frost formation enables us to predict the system performance degradation under frosting conditions.

This paper introduces food quality as a new parameter, together with energy, to determine an optimal time between defrost cycles depending on ambient parameters. Based on this, a new defrost-on-demand scheme is proposed.

The paper starts with a simple introduction to supermarket refrigeration systems and the most commonly used defrost methods, defrost schemes, which is in Section 2. We propose a new defrost-on-demand control scheme in Section 3. In Section 4, we discuss the problem connected with traditional defrost schemes by using three models, two for energy and one for food quality. Simulation results are presented in Section 5. Gains by the new defrost-on-demand control scheme is demonstrated in Section 6. Finally some discussions and conclusions are given in Section 7.

2. SIMPLE INTRODUCTION OF REFRIGERATION SYSTEM AND DEFROST

2.1 *Refrigeration of foodstuffs in a supermarket*

The display cabinet depicted in Fig. 1 consists of a food container and an air tunnel. The evaporator inside the air tunnel cools the passing air, which circulates around the food container and creates an air blanket on the top of foodstuffs.

To control properly the temperature inside a cabinet, one or more thermal sensors are required in the system. The number and function of those sensors differ from one application to another. For example in our system, there are 3 sensors: S_3, S_4, S_5. S_3 and S_4 are used to measure the temperature of air inlet $T_{a,i}$ and outlet $T_{a,o}$ respectively, S_5 is a defrost stop sensor.

To increase sales, any physical obstacles between the products and customers should be avoided. As a consequence, for most display cabinets, one or more forced air blankets (for horizontal display) or air curtains (for vertical display) are the only barrier between the refrigerated foodstuffs and warm ambient air. Due to the disturbance of air flow, a more or less significant amount of warm ambient air is always entrained, which introduces the frost formation, reduces the temperature control capabilities and increases energy consumption.

2.2 *Requirements on food storage temperature*

In supermarkets, there are general requirements regarding the storage temperature for different foodstuffs in display cabinets. For example in Denmark, according to Announcement [2004] and DSK et al. [2004].

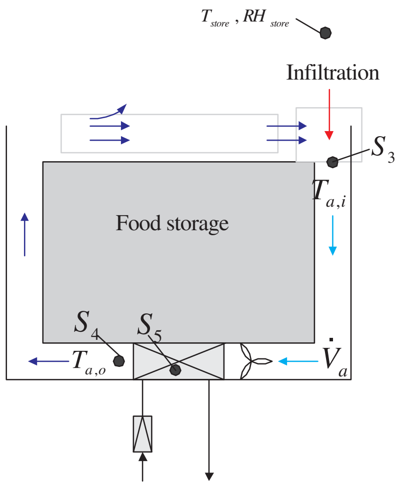

Fig. 1. A simplified display cabinet in a supermarket

- Frozen food, the max. temperature is -18°C.
- Fresh fish and fish products, the max. temperature is +2°C.
- Milk, the max. temperature is +5°C.

Usually, the legally constrained temperature is the air temperature.

2.3 *Frost formation and defrost methods*

Frosting is a well known and undesirable phenomenon on evaporator coils. It happens whenever the surface temperature of the evaporator is below 0°C and humid air passes by. Frost decreases the performance of the heat exchanger. In order to maintain a satisfactory performance, defrosting needs to be done regularly.

The most common defrost methods for medium and low temperature applications are:

- Off-cycle defrost: This is the simplest defrost methods: refrigeration is stopped, evaporator fans continue to move room air over the frosted coil surface, which warms and melts the frost.
- Hot/cool gas defrost: During a hot/cool gas defrost, the normal supply of cold refrigerant is stopped. The former involves the circulation the hot gas from the compressor discharge manifold directly to the display cabinet, and the latter utilizes cooler gas from the liquid receiver. The cool or hot gas condenses in the evaporator, releasing heat which melts the ice from the coil.
- Electric defrost: This approach uses the electric heater embedded on the fin surface to supply heat, warm and melt the frost.

Normally, for medium temperature applications, the cheapest and least energy consuming means of defrost is off-cycle. For low temperature applications, electric defrost is the most commonly used.

2.4 *Defrost schemes*

In a simple taxonomy, there are the following two classes of defrost schemes.

- Scheduled defrost: Initiating the defrost cycle by a timer, normally with a fixed number of defrost cycles per day. Defrost is terminated either based on a fixed time or on a stop temperature while with a maximum defrost time as a security.
- Defrost-on-demand: Initiating the defrost cycle only when necessary, see Llewelyn [1984]. This approach normally uses one parameter to initiate and terminate the defrost process, such parameters could be: air pressure difference across the evaporator, fan power sensing etc.

Features and shortcomings of the current defrost schemes are:

- Scheduled defrost is simple and uses a low cost controller, so it is the most commonly used defrost scheme in today's supermarkets, but the time schedule is normally determined based on experience and observation, most cases, based on worst case conditions. It is configured during the commission stage and can not automatically adapt to the varying shop conditions under which the system is working, so the time between two defrost cycles may be either too long or too short.
- Existing defrost-on-demand schemes typically involve the installation of an additional sensor to detect frost build-up, and use one parameter to initiate and terminate the defrost cycle. The threshold of this detected parameter is determined mainly to ensure a safe operation, or maintain the performance degradation of the system within fixed limits over the whole range of operating conditions. No energy optimality is guaranteed.
- None of the existing schemes have used food quality as a decision factor.

3. DEFROST CONTROLLER DESIGN

To overcome the shortcomings of the current defrost schemes, and realize an objective with a balanced system energy consumption and food quality loss, we propose a new defrost-on-demand control scheme, see Fig. 2. It uses a feedback loop consisting of an on-line model updating and estimation by an Extended Kalman Filter (EKF), as well as a model based optimization (see below). z_m is the measured output. It could advantageously use some extra sensors, but this scheme would also work just with the existing sensors in the system. For example, for the refrigeration system in Fig. 1, if S_3 is the controlled temperature by the normal controller, S_4 could be a good candidate for on-line measurement. Here we assume that the store temperature T_{store} and relative humidity RH_{store} are measured, such as at every half or at every full hour, depending on the stability of store indoor conditions. $d_{fr,opt,ave}$ is the average optimal frost thickness for defrosting generated by the optimization, but defrost will only be initiated when the estimated frost thickness d_{fr} from the EKF is equal to or larger than $d_{fr,opt,ave}$. This initiating signal is sent to the normal defrost controller, defrosting starts, and it is terminated in a normal way. After the defrost is complete, a reset signal is sent to the EKF, and the process for the next defrost cycle starts again.

3.1 *Model based optimization*

The optimization objective is described as the following:

$$\min_{t_{opt}} E(t) + k \cdot Q_{food,loss}(t) \quad (1)$$

Where $E(t)$ is the system energy consumption, which includes two parts, one is the energy used direct for defrosting, another is the extra energy used for compensating the degraded system

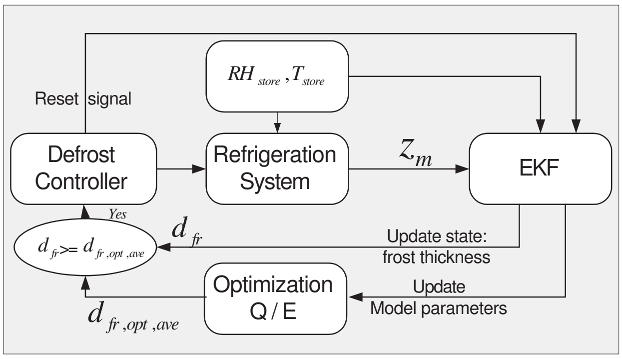

Fig. 2. On line new defrost-on-demand control scheme

efficiency due to frost build-up. $Q_{food,loss}(t)$ is the quality loss of foodstuffs, k is a weighing factor based on costs or shop owners' priorities, t_{opt} is the optimal cooling time between defrosting.

To determine an optimal frost thickness threshold under dynamic situations is not easy; a simple method is proposed, $d_{fr,opt,ave}$ is approximated by the average value of optimal frost thickness under the whole working range. Simulation results under the following conditions: [20°C, 25°C] [50%, 60%] showed that by using the average to initiate defrosting, the maximal energy loss compared with using the true optimal value is less than 4%. The alternative using a detailed model for determining the value given by Eq. (1) has from a control point of view a serious lack of robustness, and realistic modeling errors could easily cause larger deviations than 4%. Moreover this 4% in worst-case is insignificant relative to the potential savings demonstrated below.

3.2 Features and advantages of the new controller

- Adaptivity: The approach suggested uses real time disturbance measurements (store temperature and humidity) for on-line model updating, estimation and optimization, continually seeking an optimal time for defrosting under dynamic conditions.
- Optimality: The approach suggested is based on an optimality condition which is a weighted function between system energy consumption and food quality loss, so the resulting closed-loop system will always operate on the Pareto-optimal trade-off curve between system energy consumption and food quality loss.
- Feasibility: The proposed method is a model based control method, and introduction of an EKF can avoid some special sensors. Those sensors are normally expensive, and often some reliability and feasibility problems are associated with the complex and unreliable sensing methods. The EKF can infer those values of interest from some available measurements. Thus the new controller can be implemented directly on the top of existing systems, no physical rearrangements or extra components installations are required.

4. ENERGY AND QUALITY MODELING

4.1 Extra energy to compensate for the reduced performance under frosting

Frosting of the heat exchanger surface affects its thermal performance in the following ways, see Chen et al. [2003]:

- It increases the thermal resistance between the fin and airflow, and decreases the cooling capacity of heat exchangers.
- It substantially reduces the airflow through heat exchangers, and increases the air pressure drop through heat exchangers. Depending on the characteristic of the fan, several hours later, the airflow path may be nearly or completely blocked.

Frost build up is a complex process even on a flat plate. It is affected by many factors, such as air flow velocity, air and plate temperature, air humility ratio etc. Frost growth on a real evaporator becomes even more complex, limited modeling exists, see Yang et al. [2006].

This paper is not aiming at developing a detailed model to predict the frost formation under varying conditions. Instead it uses energy correlations to calculate how much water is condensed on the surface of evaporator as frost. The purpose of modeling here is for controlling, so the model itself is extremely simple but still captures the main dynamical features seen from an input / output point of view.

Frost growth rate estimation: The nominal cooling demand Q_{nom} for a display cabinet can be calculated as follows (without special notification, all the units are standard *SI* units):

$$Q_{nom} = Q_0 \sum_{i=1}^{4} (G_i \cdot X_i \cdot Y_i) \tag{2}$$

Where Q_0 is the standard cooling demand, can be calculated according to cabinet category and dimension. G_i is the correction factor for the difference between testing conditions and measured actual operating conditions, X_i is the load distribution factor, Y_i is the load reduction factor related to the covering of the display case, if no physical covering $Y_i = 1$. i indicate the load type, 1 for the load from heat conduction, 2 for infiltration, 3 for radiation and 4 for the load from electric equipments, such as light, fan, anti-sweat etc.

The correction factor for the infiltration and the infiltration load can be calculated as follows:

$$G_2 = \frac{I_{store,m} - I_{cab,m}}{I_{store,test} - I_{cab,test}} \tag{3}$$

$$Q_{inf} = Q_0 \cdot (G_2 \cdot X_2 \cdot Y_2) \tag{4}$$

Where I are specific enthalpy, which can be calculated based on temperature and RH. Subscript *store*, *cab* represent store and cabinet respectively, $test$ and m refer to test and measurement. Parameters and details see Holm et al. [1996].

Infiltration is caused by an amount of hot humid air from the store entrained in the display cabinet. The load of infiltration can also be calculated in another way:

$$Q_{inf} = \dot{m}_{a,ent}(I_{store,m} - I_{cab,m}) \tag{5}$$

From the above correlation, we can calculate the mass flow rate of air entrained $\dot{m}_{a,ent}$.

Infiltration is the only source of water condensates on the surface of evaporator and eventually becomes frost. As time goes, it will increase both the thickness and density of frosts.

$$\dot{m}_{fr} = \dot{m}_{a,ent}(x_{store} - x_{cab}) \tag{6}$$

$$\dot{m}_{fr} = \rho_{fr} \cdot \dot{\delta}_{fr} \cdot A_{fr} \tag{7}$$

Where x is the specific humidity or humidity ratio of air, based on temperature and RH, $\dot{m}_{fr}$ is the frost mass growth rate. ρ_{fr} and A_{fr} are the frost density, frosting area respectively. $\dot{\delta}_{fr}$ is the frost thickness growth rate. Here we assume the density of frost is a constant. A review on frost properties and modeling was given by Iragorry et al. [2004].

Frosted fin efficiency: Frosted fin efficiency η_f according to Barrow [1985] can be calculated as follows. Eq. (8) and (9) applies to both dry and frosted conditions.

$$\eta_f = \tanh mL/(mL) \tag{8}$$

$$m = \sqrt{\frac{h_a}{k_{fr}\delta_{fr} + k_f t_f/2}} \tag{9}$$

Where L is the effective length of fin, m is a fin parameter, k is used for conductivity, h for heat transfer coefficient, δ and t for thickness, subscript a, fr and f refer to air, frost and fin.

The overall heat transfer coefficient U based on the total air side area is given by:

$$\frac{1}{U} = \frac{A_a}{h_r A_r} + \left(h_a\left(\frac{A_f}{A_a}\right)\eta_f + \left(\frac{h_a k_{fr}}{k_{fr} + h_a\delta_{fr}}\right)\left(1 - \frac{A_f}{A_a}\right)\right)^{-1} \tag{10}$$

Where A is used for area, subscript r refer to refrigerant.

Evaporator fan: In refrigeration systems, axial fans or centrifugal fans are commonly used. The operating point of the fan installed in a system is established at the intersection of the fan and device curve. Fig. 3 shows a system and fan interaction.

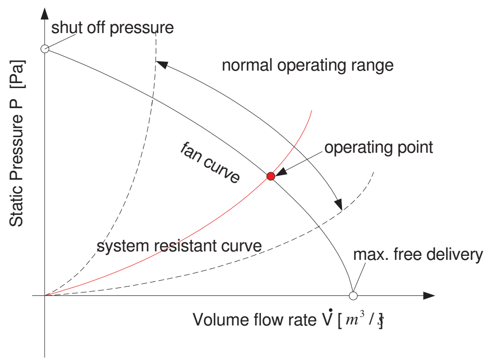

Fig. 3. Fan and system interaction

COP (Coefficient Of Performance): System COP and compressor power W_C can be calculated as follow:

$$\eta_{c,nom} = Z(C + D \cdot \dot{V}_r) \tag{11}$$

$$\varepsilon_c = \frac{T_e + 273.15}{T_c - T_e} \tag{12}$$

$$COP = \eta_{c,nom}\varepsilon_c \tag{13}$$

$$W_C = Q_{nom}/COP \tag{14}$$

where Z, C, D are constants for compressors. $\dot{V}_r$ is the volume flow rate of refrigerant, T_e is the evaporating temperature. T_c is the condensing temperature, assumed to be a constant. Parameters and details see Holm et al. [1996].

Overall calculation procedure: Here we use the heat exchanger from Blundell [1977] as an e example. It has wavy continuous fins on circular tubes, with a staggered array. Slight modifications have been made on the dimension to meet our cooling demand. The overall calculation procedure includes 2 loops: one is the time integration loop for frost growth, another is an internal iterative loop for finding the operating point of fan. Here we use an island site medium temperature display case for fresh fish products as an example, according to the requirements from food authorities, the maximum storage temperature is +2°C. It uses electric defrost. Details see Cai [2007].

Power and extra energy consumption: In this system, we focus on two power consuming components: compressor and evaporator fan (the power consumption of the condenser fan has no direct relation with frosting). Extra energy means that if we do not defrost, the efficiency of the system will degrade with frosting, in order to meet the same cooling demand, more power is needed compared with frost free conditions.

$$W_{tot}(t) = W_{comp}(t) + W_{fan}(t) \tag{15}$$

$$W_{extra}(t) = W_{tot}(t) - W_{tot}(0) \tag{16}$$

Where 0 is the frost free time, t is the time for frost growth.

4.2 Direct energy use for defrosting

In order to maintain a satisfactory performance of heat exchangers, a periodic defrosting is required to remove frost. During a defrost cycle, the cooling system is shut down, and heat is supplied to the heat exchanger to raise its temperature well above freezing.

Energy distribution in a defrost cycle is:

- Energy used to warm and melt frost $E_{df,fr}$.
- Energy used to heat the coil of heat exchanger $E_{df,coil}$.
- Energy used to heat the refrigerant $E_{df,r}$.
- Energy wasted (the defrosting efficiency).

4.3 Food quality loss under defrosting

Food quality decay is determined by its composition factors and many environmental factors, such as temperature, relative humidity, light etc. Of all the environmental factors, temperature is the most important.

Food temperature T_{food} is determined by the cabinet air temperature T_{air}. For simple calculations, we can lump the food into one thermal mass.

$$(mCp)_{food}\frac{dT_{food}}{dt} = UA(T_{food} - T_{cab}) \tag{17}$$

Food quality loss $Q_{food,loss}$ can be calculated as follows:

$$Q_{food,loss} = \int_{t_0}^{t_f} 100 \cdot D_{T,ref}\exp(\frac{T_{food} - T_{ref}}{z})dt \tag{18}$$

Where $D_{T,ref}$, T_{ref}, Z are quality parameters, see Cai et al. [2006]. UA is the heat transfer coefficient and area from air to products, mCp_{food} is the thermal mass and properties of foodstuffs. Here we assume T_{cab} is the same as $T_{a,i}$, during normal operation, T_{food} and T_{cab} are equal, $UA/(mCp)_{food} = 3.97 \cdot 10^{-4}$.

5. SIMULATION RESULT

We are aiming at finding an optimal time interval between defrost cycles to meet our optimization objective, which is a weighted function between system energy consumption and food quality loss. Regarding the energy consumption, we need to consider two aspects: extra energy and energy for defrosting.

5.1 Extra energy

Simulation is carried out for 11 hours under frost formation, with a store temperature of 25°C and RH of 55%. Fig. 4

shows the operating points of the fan as a function of time, due to frosts build up. When the evaporator is clean, the fan provides a reasonable high air flow. As pressure drop increases, the air flow is dramatically decreased. After 11 hours, the fan is already working out of its normal operating range. When the air flow rate decreases, the overall heat transfer coefficient between the air and evaporator will decrease. In order to meet the same cooling demand, the temperature drop of the air across the coil must increase. This, in turn requires a lower evaporating temperature, see Fig. 5. The drop in the evaporating temperature will cause a lower COP and increased power consumption. Fig. 6 shows the compressor and fan power consumption for the same cooling demand as a function of time under frosting.

5.2 Defrost energy

Fig. 7 shows the energy used to warm the coil and to melt frost as a function of time between defrosting. From the figure we can see that the longer time we wait for initiating the defrost, the more energy is needed both for melting the frost and warming the coil. This is because, on one hand, frost accumulates with time. On the other hand, the coil will become colder when the evaporating temperature goes down, it needs more energy to be warmed up.

5.3 Food quality loss

Fig. 8 shows the daily food quality loss under different defrost frequencies.

5.4 Energy vs. Quality

We use one day as an example, and assume we defrost the system 2, 3... up to 6 times, then the cooling time between two defrost cycles will be 12, 8... 4 hours (defrost time is ignored), this is also the time that we allow frost to grow and the system performance to degrade. We plot the daily energy consumption and daily food quality loss in Fig. 9. From the figure, we can conclude that from an energy point of view, we should select an optimal cooling time of 5 hours. But from the food quality point of view, we should defrost at a longer interval, such as 11 hours. This is a conflicting requirement to supermarket owners. It is up to them to make the final decision, based on their preference on quality, or cost, or a trade-off.

6. GAINS FROM NEW DEFROST-ON-DEMAND CONTROL SCHEME

The above simulation is based on one specific situation, where the store has a constant temperature and relative humidity, which is more or less true for a store with air conditioning systems, while in some European countries, such as Denmark, this is not the case. The indoor environment will normally vary with outdoor condition, staff and customers' activities. The fixed optimal cooling time which is determined off-line and configured at the commissioning phase, as conditions change, may not be the best choice any more.

Fig. 10 shows the energy optimal cooling time under different store conditions. Generally speaking, a high store temperature and RH gives more load to the system, a faster frost growth, and a quicker performance degradation, which requires more frequent defrost. More precisely, it is the specific enthalpy and humidity ratio that determine the frost formation rate.

From Fig. 11, focusing on the energy aspect, we can see that if we configure the defrosting of the system at an optimal time interval of 9 hours, according to one initial condition of 20°C, 50% RH, when the store temperature rises up to 25°C, same RH, this 9 hours scheme will lead to a daily energy consumption of 229.8 kJ. Compared with its actual energy optimal point of 101.5 kJ at 6 hours, an extra 126.4% of energy is wasted.

7. DISCUSSION AND CONCLUSION

This paper discussed the problems related to the traditional defrost schemes. Through the analysis on both system energy and food quality, we propose a new way of determining the optimal time between defrost cycles, and a new defrost-on-demand control scheme. It on-line adjusts the cooling time between defrost cycles, according to the varying operation condition, continuously seeking an optimal time interval, featuring either an energy optimal point, or a trade-off between system energy consumption and food quality loss.

REFERENCES

Announcement. *Announcement 1271 of 13/12/2004 concerning hygiene of foodstfuffs by Danish authorities, http://www.foedevarestyrelsen.dk*. 2004.

H. Barrow. A note on frosting of heat pump evaporator surfaces. *Heat Recovery Systems*, 5(3):195– 201, 1985.

C. J. Blundell. Optimising heat exchangers for air-to-air space-heating heat pumps in the united kingdom. *Energy Research*, 1:69– 94, 1977.

J. Cai. *Model based control of refrigeration systems*. PhD thesis, Automation and Control, Department of Electronic Systems, Aalborg University, Denmark, 2007.

J. Cai, J. Risum, and C. Thybo. Quality model of foodstuffs in a refrigerated display cabinet. Purdue, USA, 2006. 11th Int. Refrigeration and Air Condition Conference.

H. Chen, L. Thomas, and R. W. Besant. Fan supplied heat exchanger fin performance under frosting conditions. *Int. J. Refrigeration*, 26(1):140– 149, 2003.

DSK, Danish Supermarket Group, and COOP. *Hygiene and Self Control, Regulation for Supermarkets*. 2004.

P. Fahlen. Frosting and defrost of air coils - results from laboratory testing. 1996.

H. V. Holm, P. O. Danig, and B. D. Rasmussen. Energy saving for remote refrigeration and frozen equipments in trading and service sector, calculations of norm energy consumption, report nr. 3 (danish). 1996.

R. H. Howell, L. Rosario, D. Riiska, and M. Bondoc. Potential saving on display case energy with reduced supermarket relative humidity. Sydney, Australia, 1999. 20th Int. Congress of Refrigeration, IIR/IIF.

J. Iragorry, Y. X. Tao, and S. Jia. Review article: A critical review of properties and models for frost formation analysis. *Int. J. Refrigeration*, 10(4):393– 420, 2004.

D. S. Llewelyn. A significant advance in defrost control. *Int. J. Refrigeration*, 7(5):334– 335, 1984.

D. K. Yang, K. S. Lee, and S. Song. Modeling for predicting frosting behavior of a fin tube heat exchanger. *Int. J. Heat and Mass Transfer*, 49:1472– 1479, 2006.

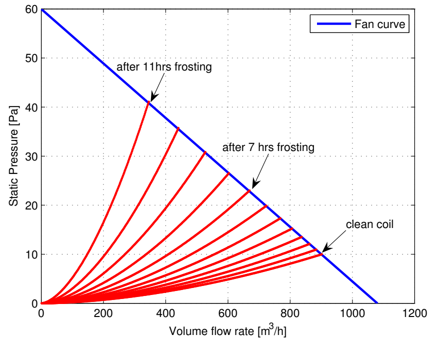

Fig. 4. Operating point of fan as a function of time under frost build-up

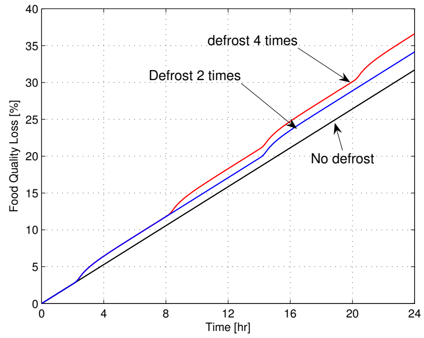

Fig. 8. Food daily quality loss under defrosting frequencies

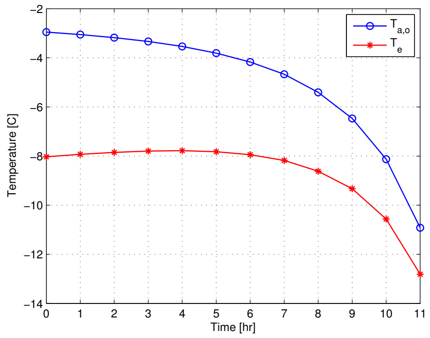

Fig. 5. Evaporation and air outlet temperature as a function of time under frost build-up

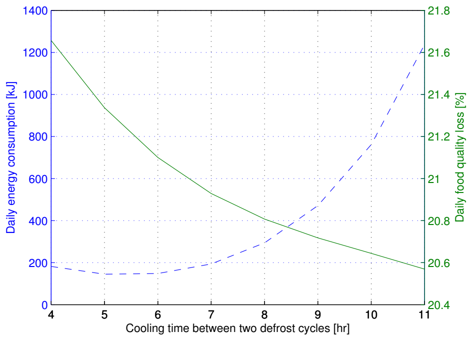

Fig. 9. Daily Energy consumption and Food quality loss as a function of cooling time between defrosting

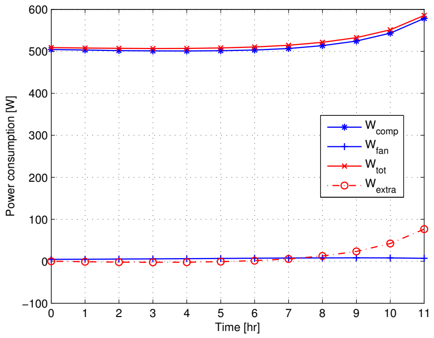

Fig. 6. Power consumption for compressor, fan, total and extra as a function of time as frost build-up

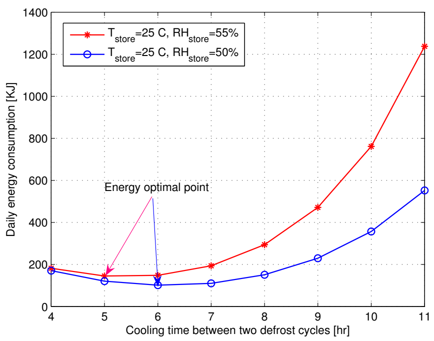

Fig. 10. Energy optimal time under different store RH

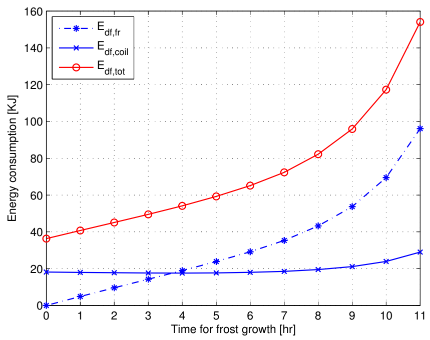

Fig. 7. Energy consumption for melting frost, warming coil and the total as a function of time

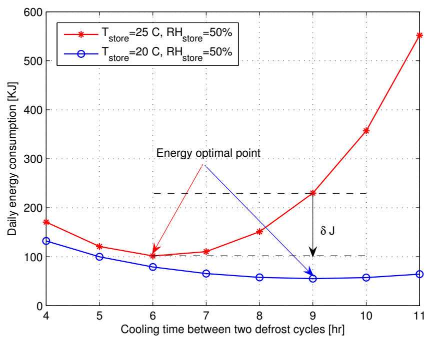

Fig. 11. Potential gains on energy by the new defrost-on-demand control scheme