CONSERVING ENERGY IN BLAST FREEZERS USING VARIABLE FREQUENCY DRIVES

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ABSTRACT

A stationary blast freezer processing 22-lb cartons of sardines in 19,000 pound lots was modified to improve efficiency and to conserve energy. Baffles were first added to produce uniform air flow. Maximum measured freeze times of 12.6 hours fell to 10.5 hours; total electrical energy savings was estimated to be 12%. A Variable Frequency Drive (VFD) was then installed to slow evaporator fans during the freeze cycle. In one example run when fans were slowed to 75% speed after an initial freezing period, freezing times then increased by about an hour while overall energy consumed fell by an additional 10%. Based on the effect of the VFD alone and $0.05/kWh power in this case, payback period would be in the range of 850 freezing cycles. Mathematical models, calibrated by measurements, analyzed further design and fan control options. Further improvements in freezing efficiency were predicted.

INTRODUCTION

Stationary blast freezers, when well-designed, can uniformly batch-freeze food products at a rate that preserves quality and keeps pace with production. They can also be the major power-user in the operation, a financial concern as energy costs escalate. This project demonstrated mechanisms to improve performance in a production blast freezer cell. Application of a Variable Frequency Drive (VFD) to control evaporator fan speed showed the potential for additional energy conservation by slowing fans as freezing proceeded.

Food products in stationary batch freezers typically have freezing times of several hours. Partway into the cycle, when surface layers are frozen, a diminishing heat flow rate is increasingly controlled by the internal conduction resistance – that is, the Biot number becomes large. As this occurs, we can reduce fan speed and so, air velocity, with only a minor effect on freezing time. Air velocity will vary directly with fan speed; but power driving the fan varies with $(\text{fan speed})^3$. So a minor slowing of fans can effect a major decrease in fan power and resulting heat. During the summer of 2002, we performed experiments on a production freezer, one of 12 operated by Columbia Colstor Co., Woodland, WA and leased to Pacific Seafood Group for freezing of 22-lb cartons of sardines (*Sardinops sagax*).

The work was supported by the National Sea Grant program with contribution both from Columbia Colstor and the local Utility, Cowlitz PUD.
Objectives were to:
- Demonstrate measures to maximize performance in an existing stationary freezer;
- Demonstrate the use of a VFD to conserve energy in a production freezer;
- Develop analyses to simulate and optimize freezer performance under conditions not tested.

BASELINE CONDITIONS AND PRACTICES

Blast Cell

Twelve cells in the plant, configured in a line, were said to be identical, with the exception of two in which false ceilings had been installed. Figure 1 shows one of these two cells, subsequently used for our controlled experiments. Air flowed from right to left through racks loaded with product. Horizontal floor width between curbs in the air-flow direction, was 90"; length between the steel 10" angle-iron cart stops which fold down to protect doors at each end of the cell, was 24-1/2'. The height of the initial ceiling (shown) was 84".

Each cell held 12 racks of product, wheeled into the room with electric pallet jacks. Base dimensions of the racks were 42" x 48" with the former dimension lying in the air-flow direction. Overall height when loaded was 74-1/2". A drop-down plastic curtain slung from the air-supply-side ceiling contacted the upper row of cartons, limiting air flow between the top row of cartons and the ceiling. For the 22-lb sardine cartons used, the loading capacity of the cell was 19,200 lb [(12 racks) x (12 rows/rack) x (6 cartons/row) x (22-lb/carton)]. An upper ceiling platform in each cell supported a 3-fan evaporator unit. A vent in the upper wall of each cell enabled pressure-equalization (preventing damage to walls and ceiling) when the room temperature initially fell. These vents were said to be undersized, thus start-up guidelines called for fans to come to full-speed over a period to exceed 15 minutes (1).

Figure 1. Blast air in the original (unbaffled) cell flows from right-to-left; air by-passes over the top and around the near and far ends.

Evaporator Unit

One Frigidcoil unit (York International) in each cell used ammonia refrigerant in a liquid overfeed mode, with a design-pumping ratio of around 4. Three axial-flow fans (Airfoil Impellors, Inc.) supplied a total no-load flow rate specified as 84,280 cfm. This might fall by 25% in a fully loaded room. Fan motors were nominal 7-1/2 HP inverter-duty models manufactured by Baldor.

Refrigeration System

An engine room dedicated to freezing operations had a total capacity intended to meet the needs of 12 blast freezer cells, 12 vertical plate freezers, and two tunnel freezers, operated at various sequences. This two-stage ammonia system operated under the following typical conditions:

- Low-stage: 14” Hg vacuum (-50°F);
- Intermediate stage controlled to 14 psig (-2.5°F) and fluctuating some with load (increasing with higher load);
- High-stage operates between 157 and 167 psig (87° and 91°F).

An automatic monitoring and control system, installed by TechniSystems Inc. (2), enabled optimized performance of the refrigeration system, as well as continual monitoring of all components. All parameters were routinely recorded at 15 minute intervals and stored in memory.

A separate computer screen designed for our experiment monitored and recorded the following at 1 minute intervals, storing in a separate file:

- Temperature of air entering the evaporator;
- Temperature of air leaving the evaporator and downstream fans;
- Saturated refrigerant suction temperature;
- Status (open vs. closed) of the liquid refrigerant supply valve;
- Speed of fans;
- Electrical power drawn by fan motors;
- Time since last room defrost.

This screen also enabled one to control fan speed by controlling frequency of the VFD that would be installed for later experiments. Electric power readings resulted from a Veris Industries power meter installed on the three-phase power supply to the VFD.

The overall performance of this system – installed in 2001 -- was monitored by Cascade Energy Engineering (3) who estimated the following: a) For low loads in the off-season, performance was recorded as 6.43 kW/TR (tons of refrigeration). This included electric power (kW) driving compressors, condensers, and ammonia recirculation pumps. b) During the 2400 hour sardine season (7/1-10/8, 2001), this figure was 2.73 kW/TR. By the time of our 2002 tests, the refrigeration system had changed, adding a booster compressor with economizer.

The Product

This project concentrated on a single product: corrugated cartons holding 22 lb of whole sardines. Outer dimensions were 3-1/4” x 13” x 20-1/4”. There were small holes in the sides, plus six 3”-diameter holes in the top. However, air was unable to penetrate because of a plastic wrap used to line the box. The approximate 5/32” corrugated cardboard wall acted as an insulator, keeping nominal freezing times fairly high, 12-15 hr.

Operation

In a typical freeze cycle, workers loaded the blast cell with product using electric pallet jacks. Once loaded, the operator closed both doors and started the refrigeration sequence using a key which opens a liquid-line solenoid valve to admit liquid to the evaporator; the fans started automatically by timer soon after.

To prevent an initial over-load of the refrigeration machinery (and unnecessary start-up of extra compressors), the start-up sequence actually took place over a 1-hour period as follows (4): An operator manually started the cycle with the turn of a key located at the loading door. This opened a valve supplying liquid ammonia to the evaporator coils. A few minutes later, the center (of three) fan switched on to full speed; one half-hour later, the center fan switched off as the two outer fans switched on to full speed; one half-hour after that, the center fan switched on. If a loading door were to be opened during the freeze cycle, the liquid line automatically closed and the fans shut down. The one-hour start sequence would begin again when the door is then closed. [Note that one problem with this start sequence is that idle fans began to rotate in reverse. Sudden start-up then caused a high spike in the starting current, risking motor burnout]

Although thermal load in the cell was highest at the start of the freeze cycle, balanced use of freezer units enabled air and evaporator temperatures to drop quickly. One initial experiment recorded conditions when freezing cartons of chicken parts. Although freezing time was about 24 hours, suction temperature fell to a near-steady value of -30°F almost immediately. [The refrigeration was operating on a single stage at the time]. Supply air fell to -20°F within an hour, then declined slowly to about -30°F.

Another early experiment with sardines (coded by date: 0710) and application of the full, two-stage refrigeration capacity, showed suction temperature dropping immediately to about -48°F; supply air fell quickly within the first hour to around -35°F, then slowly decreased to -40° over the freezing period. The completion of freezing was routinely confirmed by pulling a carton or two from areas of the freezer in which poorest freezing rates might be expected. The worker could then drill a hole and measure core temperature with a stem thermometer. Longest measured freezing time in the original cell using slightly thicker cartons was about 15 hours.

Every few cycles, production schedule permitting, the operator would initiate a manual defrost which sent hot gas through the coils for about 30-40 minutes, while fans are switched off. On occasion, the room itself would need to be cleared of built-up ice, particularly on the floor. This was accomplished by running fans at full speed (doors closed, refrigerant flow off) for a few hours.

METHODS AND MATERIALS

Instrumentation

The first project activity was to select and install instrumentation needed to measure freezer performance and results of various modifications to be made to structure and operation.

Airflow

A project objective was to modify flow paths in an attempt to achieve maximum velocities with the highest uniformity through the pack. To measure velocities under different configurations and fan speeds, we used a hand-held anemometer with a digital readout and measurement range of 60-7830 fpm (feet per minute; Extech Pocket Thermo Anemometer). An attached streamer (a piece of cassette-tape) enabled the operator to note the air velocity direction and ensure a correct orientation of the impellor. Calibration conducted in a wind tunnel at the University of Portland showed readings to agree within 1%.

The procedure for measuring velocity profiles was to position the operator in the blast-cell with doors closed, downstream of the pack, prior to opening the refrigeration valve. We thus assumed the operator's body was sufficiently small (compared to the space) that it would not affect the flow stream. It can be shown that temperature has very little effect on the volumetric flow rate or velocity profile.

Temperature Measurement

These experiments employed two methods of continuous temperature measurement. In the first, product temperatures resulted from a series of 14 Type-T thermocouples (AWG 20; Special Limit of Error, $\pm$0.5°C) connected to a 14-channel temperature data logger (Databook/260 by Iotech) and a laptop PC. These sensed temperatures of air and of sardine cartons, typically near the core, at 1-minute intervals over the course of each freezing cycle. Deployment of these thermocouples with 12' leads had to be fast to avoid disrupting the loading/unloading of the cell. Thus two circuit strips and embedded sockets (Omega Model RSJ-T-R, round-hole panel jack) were "permanently" mounted to the ceiling: 7 on the supply air side; 7 on the return air side. Bundled thermocouple wires from these sockets exited the room at the lower corner of one of the doors and terminated in a locked cabinet, immediately outside the room. This setup remained undamaged and workable throughout the experiment period. Moisture entering these

sockets disrupted electrical connections on occasion. However with the use of plastic plugs in unused sockets, and drying with a hot-air gun from time to time, this was not a problem.
At the start of each run, we would attach the data logger and laptop computer in the cabinet, then insert thermocouples as the room was loaded, identifying each as to location and data logger circuit.

To enable thermocouples to be quickly withdrawn from the frozen carton at the conclusion of experiments, we fabricated stainless probes from tubing (L = 7.25"; OD = 3/16"; ID = 0.118") having a right-angled handle at the end . A twist of this handle would break the probe free of the frozen block, allowing us to withdraw it from the carton. Calibration of probes in an ice-bath at the beginning and end of the experimental period verified them all to be within 1°F of 32°F.

The second temperature data collection method used thermistor sensors (plastic bulbs of approximate ¼" diameter) mounted immediately upstream of the evaporator and downstream of the fans. These systems data were saved in the system computer memory at 1-minute intervals. Another temperature of significance to refrigeration performance was that of the saturated ammonia in the evaporator. This was continually recorded in system memory from a thermistor strapped with insulation to the suction line feeding the evaporator (4). Also recorded were the saturated ammonia temperatures in the intercooler and condenser.

Power

Because a major effect of fan speed control would be electrical power savings, we installed a power meter on the 3-phase circuit supplying the three fan motors. It was a Hawkeye "Enercept" Model H8044-0100-2 (Veris Industries, Portland, OR) operating at 480 volts. Its accuracy is said to be $\pm$1% and arrives factory-calibrated. Experience with past use of these units indicated that no further field calibration was necessary (3). Output from the meter was continually monitored on our dedicated screen and saved on the system file.

Optimize air flow patterns.

After installing instrumentation, the next project phase demonstrated how various baffles could minimize the by-passing of air and give the greatest velocity and uniformity of flow over the freezing product. Because this was a freezer in full production, constraining factors were that no change could be made to the sardine package, rack design, quantity frozen per batch, room configuration and refrigeration equipment. A sequence of early experiments (ending with experiment 0809) established an "optimized" air flow design. Experiments and results are reported below. Baffling was in place for the remaining experiments which demonstrated the results of fan speed control.

Variable Frequency Drive (VFD)

The final project phase followed installation of a VFD and demonstrated effects of its use while freezing 22-lb cartons of sardines. The VFD selected by Munter Electric (Moses Lake, WA) was a Baldor Model ID15H430-EO, rated for a 40 HP variable torque load. Other significant components were:

- The enclosure;
- A "choke" or line reactor on the incoming line to the VFD. This serves two purposes: it protects the VFD electronics from incoming power spikes; it also greatly reduces the harmonics that the drive wants to put back into the utility power system (5,3);
- A second choke, or load reactor, served to protect windings of the motor from any spikes (up to 2 KV) caused by the high frequency switching of the transistors;
- A bypass switch and related components.

RESULTS AND DISCUSSION – PHASE I: OPTIMIZE EXISTING FREEZER PERFORMANCE

The option we had to optimize the blast cell performance was placement of baffles to better direct air flow through the product. To first establish the baseline performance, we measured airflow on the downstream side of the empty room. As expected, an uneven profile showed velocities ranging from 300 fpm on the edges to over 2,500 fpm near the center where fan streams overlapped. Making the same measurements on the room when fully loaded showed a reduced total flow rate, as expected. Figure 2 shows the original system curve (solid line), assumed to represent the resistance of the evaporator coils alone. We've sketched in an estimated new system curve (dashed line) representing the resistance of the coils plus the racks of product. The equilibrium operating condition – the intersection of the fan curve with the new systems curve -- moves to a lower volumetric flow rate and higher pressure drop.

A look back at Figure 1, however reveals that not all flow went through the product. Some flowed over the top; by-pass around the ends was significant, given that the clearance between racks and doors was on the order of 15". A curtain hanging from the ceiling on the supply-air (right) side tended to pin against the top row of cartons, restricting flow immediately downstream of it.

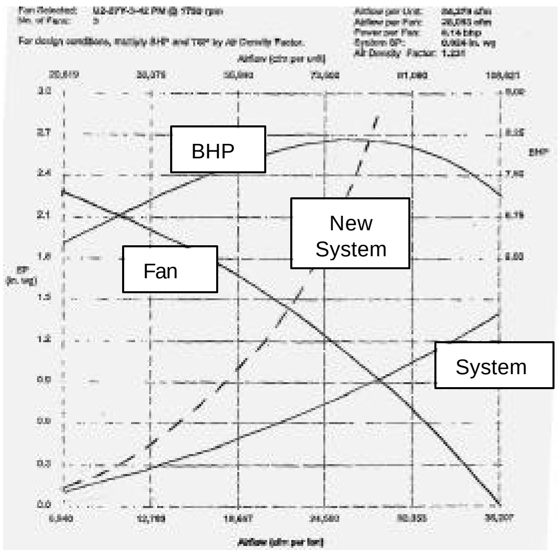


Figure 2. Pressure Drop vs. Flow rate. Operating Point is the Intersection of Fan and System Curves.

An airflow distribution in the loaded room showed velocities between cartons to range roughly from 300-800 fpm. Blast freezer designers typically look for velocities approaching 1,000 fpm to gain the fastest freezing. Further increase gains little improvement in freezing rate and adds much to the freezer's fan heat load.

Measurements also showed high velocities around the ends and over-top of the pack. By multiplying velocities by cross-sectional flow area, we can estimate volumetric flow rates in this original unmodified room, as follows:

35% flowed through the racks of cartons
15% flowed over the top of the racks
50% by-passed around the two ends.

Measured freezing times for three runs in this 'unbaffled" room appear in Table 1. The "early experiment" (0710) used cartons that were slightly thicker (3-1/2" vs. 3-1/4"); it is shown here for general information but was not used for subsequent comparisons. The measured fan energy refers to the electric motors; differences were due to the different periods of operation. Note that "freeze time" is defined here as the time required for the core to reach 0°F. Also note that results of these experiments are given here in terms of maximum and minimum freezing times *among those cartons measured*. Only 14 sensors were available to monitor temperatures of 864 cartons. Our accuracy of results depended upon a judicial selection of cartons on the inlet- and exit-air sides of the room. Although we're comfortable that results represent real and comparative performance, the low numbers of sensors and repetitive experiments prevent use of statistics to describe these results.

Prior to the next two experiments, we redirected and channeled the air-flow with a series of baffles:

- Plywood on upper supply-side corners prevented air from sweeping around the upper end of the ceiling structure;
- The ceiling was lowered to a distance of about 3" from the upper row of cartons. Although there was some concern that pallet-jack operators would damage the ceiling with excessive lifts, that was not observed to be a problem.
- Plastic sheeting on the supply-air side sealed the horizontal corner where the near-vertical and horizontal ceilings meet, and at the vertical corners between wall and cartons.
- Floor-to-ceiling plywood sheets, installed at the start of each freezing cycle, covered the ends of the racks and prevented end-around by-pass.

These modifications effectively shut off all bypass around the ends. Flow in narrow channels above and below the pack provided good convection heat transfer on surfaces of the top and bottom rows. Velocities ranged from 600 to 800 fpm – higher than for the un-baffled case, although still shy of the 1000 fpm target.

Of the two experiments run in the "full-baffled" room, one had to be discarded when workers continually opened doors, an action which restarted the one-hour fan start-up sequence each time. The other (Exp. 0809) appears in Table 1. Full-baffling reduced maximum freezing time by 15%, and fan energy use by 6%. Uniformity, the difference between measured maximum and minimum freezing times, improved appreciably. This case, then became the new baseline performance of the room, as experiments then shifted to those using fan speed control

.RESULTS AND DISCUSSION – PHASE II, APPLICATION OF THE VFD

After "optimizing" air flow in the cell, phase II of the project demonstrated effects of fan speed control on both freezing time and energy use. Because we conducted these experiments during production runs, conservative steps were taken to minimize disruption.

Table 1. Effect of Baffling in the Freezer Cell[1]

Room	Experiment Number	Max Time (hr)	Min Time (hr)	Time Difference (hr)	Meas. Fan Energy (kWh)	Total Electrical Energy (kWh)
Early Experiment	0710[2]	15.1	8.7	6.4	331[3]	1225
Base-Line Unbaffled	0717	12.3	8.7	3.6	306[4]	1082
	0725	12.7	8.6	4.1	257[5]	1025
	Averages:	12.5	8.7	3.8	282	1054
Full Baffled	0809	10.6	9.0	1.6	266[6]	924

Notes:
1 Evaporator coils defrosted immediately before each experiment
2 Product in this experiment was packed in slightly larger boxes than in subsequent experiments.
3 Used a ½ hour staggered start. Fan energy measured over 16+ hours.
4 Fan start was at 20%; going to 100% in 1 hour. Fan energy measured over 14+ hours
5 Fan start was at 20%; going to 100% in 1 hour.
6 Fan start was at 50%; going to 100% in ½ hour.

Table 2 gives results for 3 fan schedules. For each, measured fan power fell roughly according to the theoretical fan law which states that power varies with the cube of speed. Total electrical energy for fans and compressors was estimated from a system COP (Coefficient of Performance) as described in the next section.

Trial 1. In the first trial, fans came to full speed within 45 minutes. After 3-1/2 hours, the fans then slowed to 75% speed for the remainder of the freezing period.

Trial 2. A second trial employed a slightly faster startup schedule, then slowed fans to half speed after 7 hours. Of these two trials, the first saved more energy and would be the preferred option, assuming that a freeze-time increase could be tolerated.

Trial 3. The final trial in Table 2 resulted from a request from Columbia Colstor to investigate energy savings that could result from using existing on-off fan controls, i.e. without the expense of a VFD. So for Trial 3, three fans started at half-hour intervals. After 4 hours, the center fan was switched off, reducing fan power by 1/3. Full-speed operation of the two outside fans produced a flow field similar to the case in which all fans are turning at 50% speed. [The pattern for a single center fan at full speed showed poor uniformity and areas with velocities as low as 100 fpm.]

These experimental results showed that Trial 1 shortened maximum freezing time by an hour (8%) and reduced total energy use by 22%. This is based on the room performance prior to modifications. Using the full-baffled room as the baseline, maximum freezing time increased by 8%; total energy use decreased by 11%. With half of the installed VFD cost of $9,000 covered by an energy grant, and electrical rates of $0.05/kWh, the pay-back period would be about 850 freezing cycles.

RESULTS AND DISCUSSION – ANALYSIS AND SIMULATION

The above data and results enabled first the validation of freezing models. These could then simulate the effects of other variables on freezing time. The use of Excel tables to analyze data also enabled us to calculate other energy saving strategies.

Freezing Times

We used the software PDEase, a finite element package developed by Macsyma, Inc. (Arlington, MA). As described by Zhao et al (6), this package enabled relatively accurate predictions using temperature-dependent properties of the product and time-dependent boundary conditions of air temperature and velocity.

Calculations below show how this model can be used to support package and freezer design. Three examples show how freezing time is expected to vary with: a) local air velocity; b) carton thickness; c) cutout patterns in carton. Unless noted, simulations assume the carton to have the standard hole pattern (six 3-inch diameter holes) cut into the lid only; thus the last point to freeze will be slightly below-center.

Table 2. Effect of Varying Fan Speed in Fully-Baffled Room

Experiment (Fan Speed)	Max Time (hr)	Min Time (hr)	?Time (hr)	Meas. Fan Energy (kWh)	Total Elect. Energy (kWh)	Fan Startup Schedule
Trial 1 Full to 75% at 3.5 hours						
0815	11.5	8.1	3.4	151	819	55-100%, 45 min.
0819	11.3	7.7	3.6	150	819	55-100%, 40 min.
Average	11.4	7.9	3.5	151	819	
Trial 2 Full to 50% at 7 hours						
0904	11.7	7.9	3.8	168	846	50-100%, 35 min.
0905	11.2	6.9	4.3	166	828	50-100%, 35 min.
Average	11.4	7.4	4.0	167	837	
Trial 3 Center fan off at 4 hours						
0906	11.9	7.4	4.5	197	847	Staggered over 1 hr

Other assumptions made in these calculations:

Carton thickness = 3-1/4”.
Initial fish temperature= 38°F.
Fish block voids ratio=31%.
Carton walls are 3.5 mm corrugated cardboard.
Holes in carton lid represent 16% opening.
Velocity = 800 fpm (equal on top and bottom).
Initial air velocity=295 fpm; this jumps to design velocity after 30 minutes.
Freeze time is defined as the time it takes the last-point-to-freeze (about ¼” below center) to reach 0°F.
Air temperature decreases linearly by 12°F over 10 hours. Unless noted, initial air temperature is -30°F.

Results, validated with experiment, appear in Figures 3a-c. Figure 3c shows a strong effect of corrugated cardboard as packaging material. Note that the circle datapoint indicates the existing carton having 16% cutout on the top surface only.

<u>Analysis of Data</u>

All temperature and systems data, collected at one-minute intervals, were analyzed within two Excel files.

<u>Temperature Information</u>

With reference to the Systems data (described below), the following events were recorded from the temperature logs for each experiment:

- *Start*. The start of the experiment: the moment the refrigeration valve was opened;

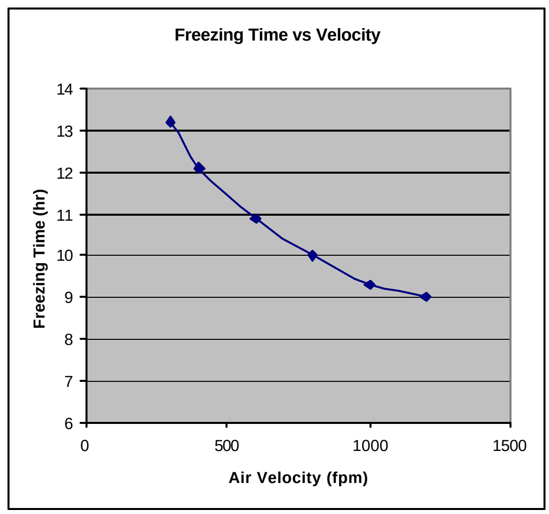
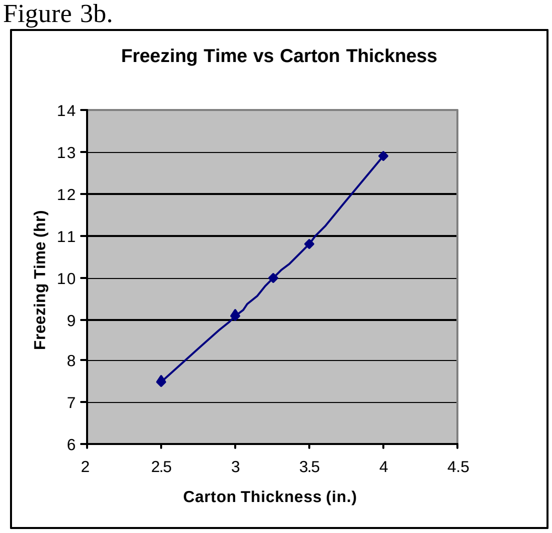


Figure 3a.

Figure 3b.



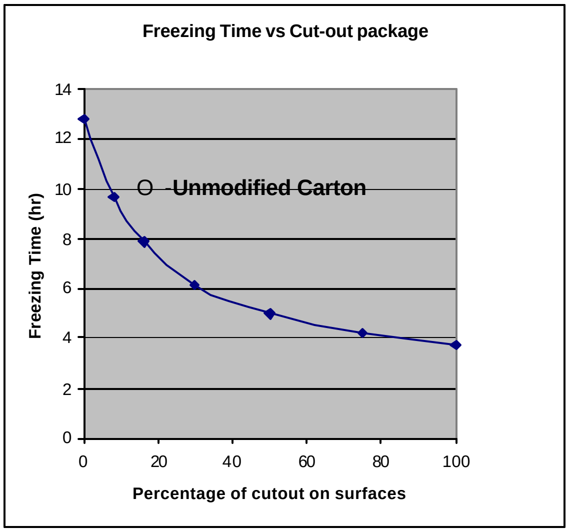


Figure 3c.

- *Minimum*. The time when the first measured product core temperature reaches 0°F.
- *Average*. The time when the average of all measured product core temperatures reaches 0°F. Note that achieving this goal would require reversing fan direction periodically, with appropriate baffling.
- *Maximum*. The time when the last measured core temperature reaches 0°F. The freezer could be stopped at this point automatically using a measured product core temperatures or other indicators (as, a dummy load).
- *End*. The time when the refrigeration valve is closed and fans are turned off -- often longer than the maximum freezing time.

System Information

The TechniSystems electronic monitoring and control system recorded information at one-minute intervals: time, air and refrigerant temperatures, refrigerant valve status, fan speed and power.

Calculations

These occur in the Systems file and include the following:

- *Product Energy, kWh*. Energy required to reduce the product from its initial to its final temperature of 0°F, calculated using moisture content and empirical thermal properties;
- *Fan Air*. Estimated from calculations and from intersection of the fan and system curves;
- *Evaporator Energy, kWh*. Calculated from the air flow rate, air specific heat and density, and temperature drop across the coil.
- *Refrigeration System Electrical Energy, kWh.* Because this could not be measured directly, we calculated it from the evaporator loads and an assumed system efficiency, expressed as Coefficient of Performance (COP).
- *Total Energy, kWh*. Sum of fan electrical energy and refrigeration system energy use;
- *Blast Freezer Efficiency*. Assumed to be the ratio of "useful" heat energy removed in the freezer (fan + product energy) to the total heat energy extracted from the evaporators. The average freezer efficiency for the 8 experiments we conducted was 87%.

Results

These analyses enable calculation of a range of cases. Some examples:

1) Controlling to last-frozen product. If we could immediately terminate freezing when the last product reached 0 °F, savings, based on our 8 experiments, averaged 1.2% of total energy.
2) Controlling to average product temperature. More-uniform freezing could be accomplished with additional baffling and reversing fan direction, similar to common practices in lumber-drying kilns. If freezing were halted when the average core temperature of all cartons reached 0°F, the total energy savings of 8% would be significant; and this would come with a 14% reduction in freeze time.
3) Controlling to evaporator load. We modeled a case for which the fan speed is continually reduced to maintain a constant temperature drop across the evaporator -- 4 °F in our example. Results: an 11% reduction in total energy, a 2% increase in freezing time.

ANCILLARY FINDINGS

Our experiments provided additional information that may be of some value to others designing and managing blast freezing of food.

A. Local flow conditions can create freezing delays. Unless reversible flow fans are used, downstream cartons will freeze more slowly than those upstream. We found, however, some additional and controllable factors that can be detrimental to freezing time.

1) Improperly-taped carton lids. Lids that are not adequately taped down will block the narrow air passage when they blow open.
2) Broken shelf support. On a couple of occasions, we noticed a rack with a failed corner weld, reducing flow in that immediate area. For two of these situations, our measurements showed 1-2 hour freezing delays as a result.
3) Placement of racks. Racks pushed hard against either curb will have inadequate flow under the bottom row of product.

B. VFD selection depends on a range of factors.
1) Sizing for cold-air fan operation. When air temperature dropped to -30°F, density, and thus required fan power increased by nearly 25% over rated power. The VFD must be sized to handle that increased load.
2) Environment. Cottonwood seeds filled the air in the summer months at our VFD location. These plugged the cooling fan intake on the drive and caused overheating.
3) VFD Bypass Switch. For our experiments, a bypass switch was installed for an additional $1,000 or so. It proved to be necessary and therefore cost effective when the VFD overheated and malfunctioned during a heavy processing schedule.

C. Package design can significantly affect freezing time.
Both calculations (Fig. 4c) and experiments showed that freezing time will vary dramatically with package design.

D. Defrost schedule should be reviewed for potential savings. Frost formation on the finned-tube evaporator can diminish performance primarily by restricting air flow. Based on detailed review of two experiments, the defrost frequency of once-every-3-cycles may have been more often than necessary.

SUMMARY OF SOME POTENTIAL VFD BENEFITS

A. Energy savings are significant for blast freezing products such as cartons of chicken or sardines, and whole salmon which take multiple hours to freeze. Application would require that some freezing delay is tolerable.

B. A diminished heat load in the room due to reduced fan heat, frees up refrigeration capacity for other loads in the plant.

C. Programmed, slow starting speeds for the three parallel fans, avoid high loads on electrical circuits and refrigeration, while providing good uniform cold air flow over the warm product.

D. Potential problems due to back-winding of staggered-start fans are avoided.

E. Weight-loss of wet product would be lower.

F. Less frost would form on the finned coils, improving air-flow and heat exchanger performance, and lengthening time between defrosts.

G. Heat exchanger effectiveness is slightly better with lower flow rates.

H. Noise level in the freeze cell is dramatically lower, providing a safer environment for workers.

I. The process of modifying a blast cell and evaluating its behavior *prior* to VFD use, characteristically leads to better air balance, decreased freezing times, and increased production rates.

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