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Optimizing working conditions of air blast freezers in seafood processing factories

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Abstract

At present, there are more than 633 seafood processing companies in Vietnam that consume 19.2% of the country's total industrial electrical power. The most popular freezers in these factories are air blast types. They are the most energy-consuming equipment, accounting for more than 50% of a company's total energy usage. Two key variables affecting the energy consumption of the freezers are freezing time and freezing temperature. In this study, a mathematical model was designed to determine the freezing time for food products and the refrigeration load of air blast freezers. It is presented as an objective function of total energy consumption for a given processing time. In addition, optimal freezing temperature, air velocity, and freezer efficiency were also determined. Our model can be used to operate air blast freezers with highest efficiency.

Keywords: Air blast freezer, frozen seafood, energy consumption, two-stage system of refrigeration

1. Introduction

Food safety and compliance with commercial contracts with customers from all over the world are of high priority to seafood exporters. However, competitive pricing of high-quality products is required for a company to increase its reputation and prestige. In Vietnam, exported seafood products are mainly frozen by air generated from freezers which were mostly imported in the 1980s. They consume more than 50% of total energy of the factories that utilize them. It was estimated that it takes about 68 to 188 kilogram of oil equivalents to freeze one ton of seafood products, accounting for 15 – 20% of its total production cost. Therefore, reduction of energy consumption is an attractive way for lowering costs and to enhance a company's competitiveness on the global market [1]. Currently, only a few methods are recommended to reduce energy cost, which includes optimizing intermediate pressure [2], analyzing airflow in freezers, or improving performance and improving design [3-7]. In addition, better control of the defrosting process in the cooling system was also reported to reduce energy cost [8]. However, no systematic study was performed to optimize energy reduction of air blast freezers. For those reasons, the present study was carried out to analyze the total heat load for air blast freezers using a two-stage refrigeration system at the Hue Industrial College, in Thua Thien Hue province, Vietnam. The total heat load summarizes the heat flow from many sources, including from the product during the freezing process, from cooling the packaging materials, from the fan motor, and from outside the cover structure, etc. In general, energy

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consumption always increases with decreasing temperature and increasing air velocity. Therefore, an optimal mode of operation is not obtained by considering each factor individually. By selecting the total freezing time based on product quality or practical factors, we discovered the optimal combination of velocity and air temperature for the processing time for squids. Thereby, an optimal operating mode for the squid freezing process was established.

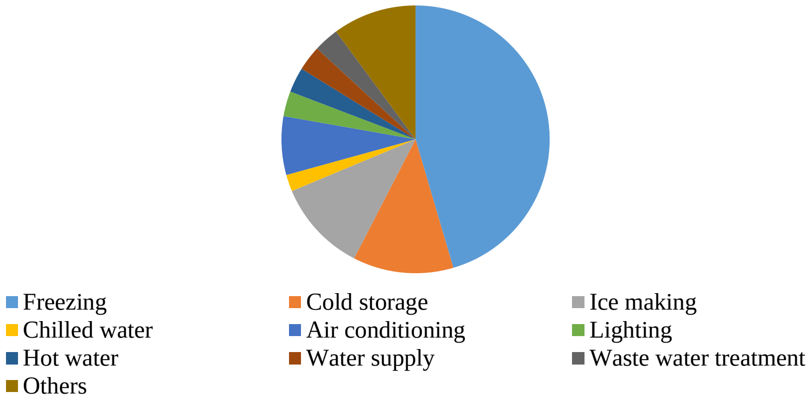


Fig. 1. The share of energy consumption in catfish processing by end-use [1].

2. Model Description

The air blast freezer dimensions were 1.2 m x 0.7 m x 0.7 m (height x width x length), with 0.7 m of polystyrene insulation in the wall, roof and floor. To reach -40^{o}C of the air temperature in the freezer, a two-stage refrigeration system with the capacity of a 4 kW compressor, an indoor fan with a capacity of 0.12 kW and volumetric flow rate of 0.5 m^3/s and pressure boost by a fan of 120 Pa was used. The air outdoor temperature was about 36 ℃ during the summer.

Fig. 2. Calculation model and experimental study

The total heat load was the sum of all the individual load components:

$$Q_{tot} = Q_p + Q_{pk} + Q_f + Q_{md} + Q_{pe} + Q_i + Q_{st} + Q_d \tag{1}$$

where Q_p was product load (W), Q_{pk} was the packaging load (W), Q_f was the fan load (W), Q_{md} was the lighting load (W), Q_{pe} was people load (W), Q_i was surface heat infiltration load (W), Q_{st} was cooling of structures load (W) and Q_d was load due to defrosting (W).

2.1. Freezer product heat load

To calculate the total heat load for the freezing process during three stages of precooling, phase change, and sub-cooling to store temperature [4], we used:

$$Q_p = \frac{M_p.\Delta h_f}{t_b} \tag{2}$$

And

$$\Delta h_f = c_l\left(T_{in} - T_f\right) + L + c_s\left(T_f - T_{av}\right) \tag{3}$$

where M_p was the mass of product (kg), Δh_f was the total enthalpy change during freezing (J/kg), t_b was the appropriate processing time (s), c_l was the specific heat capacity of unfrozen material (J/kgK), T_{in} was the initial product temperature (^oC), T_f was the initial freezing temperature (^oC), L was the enthalpy change due to freezing (J/kg), c_s was the specific heat capacity of frozen material (J/kgK) and T_{av} was final product temperature (^oC).

2.2. Packaging heat load

All packaging materials had very low moisture, and included plastics, metals and wood. Packaging heat load of freezing can be estimated [9]:

$$Q_{pk} = \frac{M_{pk} c_{pk}\left(T_{in} - T_a\right)}{t_b} \tag{4}$$

where M_{pk} was a mass of packaging material per batch (kg), c_{pk} was specific heat capacity of packaging (J/kgK) and T_a was cooling medium temperature (^oC).

Table 1. Specific heat capacity of materials [9]

Materials	c_{pk} (J/kgK)
Plastics	1600
Steel	500
Aluminum	850
Wood	2300
Fiberite	1400

2.3. Fan heat load

The best energy use estimation of the fan was given by [10]:

$$Q_f = \frac{Q\Delta P}{\eta_f} \tag{5}$$

where Q was the volumetric flow rate of air (m^3/s), ΔP was pressure drop in the facility (Pa) and η_f was fan motor efficiency. Besides, the fan energy was proportional to the fan speed cubed [9]:

$$Q_f\left(n\right) = Q_f\left(o\right)\left(\frac{v_a\left(n\right)}{v_a\left(o\right)}\right)^3 \tag{6}$$

where $Q_f(n)$ was fan energy at the new speed (W), $Q_f(o)$ was the fan energy at old speed (W), $v_a(n)$ was new air velocity (m/s) and $v_a(o)$ was old air velocity.

2.4. Surface infiltration heat load

The surface infiltration heat load depends on the insulation materials and construction of the walls, roof, floor and door of the refrigerated space, the wind conditions inside or outside, the surface area, and the temperature difference between the air inside the refrigerated space and the ambient air. The rate of heat infiltration can be calculated using [11]:

$$Q_i = UA\left(T_{am} - T_a\right) \tag{7}$$

Where U was the overall heat transfer coefficient (W/m^{2}K), A was the wall area (m^2), T_{am} was the ambient temperature (^{o}C) and T_a was the inside temperature (^{o}C).

$$U = \frac{1}{\frac{1}{h_o} + \sum \frac{x_i}{k_i E} + \frac{1}{h_i}} \tag{8}$$

where h_o was the convective heat transfer coefficient at the outer surface (W/m^{2}K), x_i was the thickness of ith layer in the wall (m), k_i was the thermal conductivity of ith layer in the wall (W/mK), E was the insulation effectiveness factor and h_i was the convective heat transfer coefficient at the inner surface (W/m^{2}K). For low air velocity < 0,4 m/s, we set h_o and h_i to about 7 W/m^{2}K. Otherwise, we used [9]:

$$h = 7,3 v_a^{0.8} \tag{9}$$

where v_a was air velocity over the surface (m/s).

Table 2. Thermal conductivity values of materials [9]

Materials	k_i (W/mK)
Fiberglass	0.040 - 0.050
Corkboard	0.035 - 0.040
Polyurethane foam board	0.022 - 0.030
Polystyrene foam board	0.026 – 0.034

Table 3. Insulation effectiveness factor [9]

Room size	Insulant type and age						
	Sandwich panel		Spray polyurethane		Other insulants		
	>10years	new	>10years	new	>30years	>10years	new
< 100m^3	2.6	1.8	2.3	1.6	3-6	2.5	2.0
100 – 500m^3	2.4	1.5	2.0	1.4	3-6	2.5	1.8
500 – 5000m^3	2.2	1.3	1.9	1.2	2-6	2.0	1.5
>5000m^3	2.0	1.2	1.8	1.2	2-6	1.8	1.4

2.5. Freezing time prediction

The total freezing time was the sum of pre-cooling, phase change, and the tempering time.

Table 4. Freezing time prediction [12]

Stages	τ (s)
Pre-cooling	$\tau_1 = \dfrac{\delta \rho_l c_l}{\left(\dfrac{\delta}{k_l}+\dfrac{1}{h}\right)^{-1}} \ln \dfrac{t_i - t_a}{t_f - t_a}$
Phase change	$\tau_2 = \dfrac{\rho_l \varphi L}{\left(t_f - t_a\right)}\left(\dfrac{\delta^2}{2k_l}+\dfrac{\delta}{h}\right)$
Tempering	$\tau_1 = \dfrac{\delta \rho_s c_s}{\left(\dfrac{\delta}{k_s}+\dfrac{1}{h}\right)^{-1}} \ln \dfrac{t_f - t_a}{t_{av} - t_a}$

2.6. Objective function

The total energy consumption (E) may be a better objective function, determined from:

$$E = P\tau \tag{10}$$

Where P was the power used by the fan (P_f) and the compressor (P_{comp}) (W) and τ was the time per freezing cycle which must be equal to or greater than the freezing time of the product (s), in which:

$$P_{comp} = \frac{Q_{tot}}{COP} \tag{11}$$

Where COP was the coefficient of performance. Typical values for actual COP's were as follows [9]:

$$COP = \frac{273+T_e}{T_c - T_e}\left(1-\alpha x\right)^n \eta_i \tag{12}$$

And

$$x = 0.00563\left(1+0.00149T_d + 0.00398T_s\right)\left(T_d - T_s\right) \tag{13}$$

in which:

$$\eta_i = 0.00476R_p^2 - 0.09238R_p + 0.89810 \text{ with } R_p = P_d / P_s \tag{14}$$

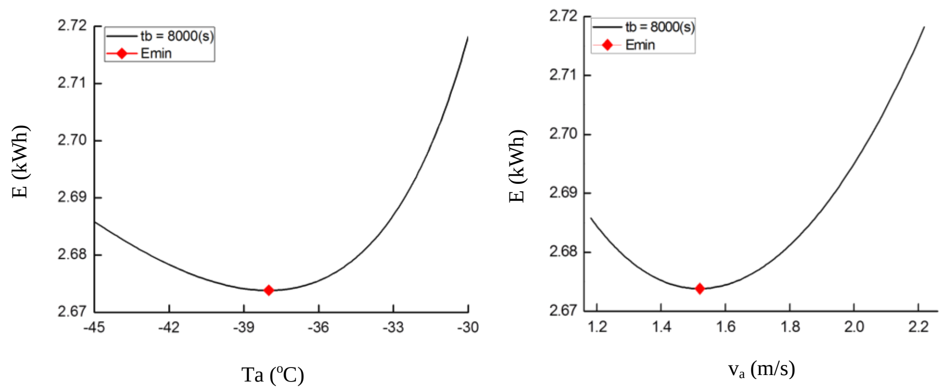
where T_e was the evaporation temperature (°C), T_c was the condensation temperature (°C), α and n were empirical constants, η_i was the isentropic efficiency, x was fractional vaporization during expansion from T_d to T_s and R_p was pressure ratio [9], [13].

3. Simulation Results

From the result of section 2, we calculated the heat load for 10 kg of squids (*Todarodes pacificus*) based on their thermal properties. The freezing time for a standard quality product was 8000 seconds ($t_b = 8000s$). Therefore, the total energy consumption can be calculated using the following objective function:

$$E = \left\{0.12 + \left[1.721 + 0.009375 v_a (n)^3 + 0.00396(36 - T_a)\right] / COP\right\} \frac{8000}{3600}$$

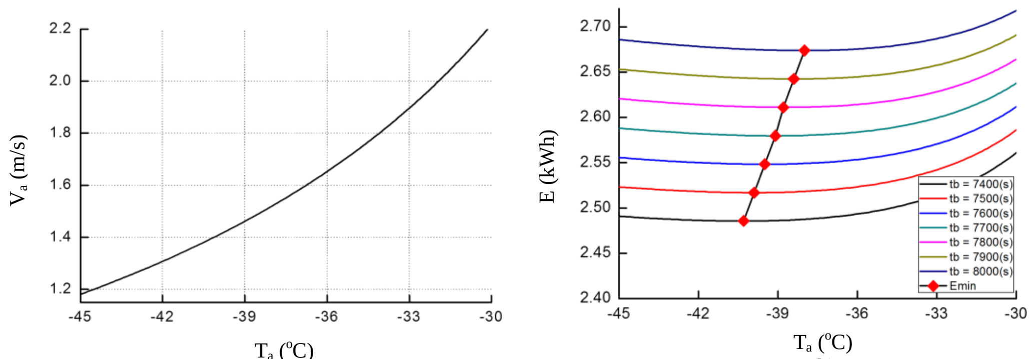
Using a Matlab program to calculate E over a certain time period and by considering the relationship between cooling medium temperature and the air velocity, we obtained the result illustrated in Fig. 3. Using function of minE, we had E_{min} = 2.6738 (kWh), with T_a = -38°C and $\upsilon_a = 1.521$ (m/s).



(a) (b)

Fig. 3. Energy consumption versus cooling medium temperature (a) and air velocity for squid (b)

It can be observed in Fig. 4(a) that if the cooling medium temperature was to be decreased, the air velocity should be also decreased. And when the product's freezing time changes, while its quality was supposedly unchanged, the minimum energy curve was displayed as Fig. 4(b).



(a) (b)

Fig. 4. Air velocity versus cooling medium temperature (a) and energy consumption versus freezing time (b)

4. Conclusion

This study provides a method for optimizing the operating mode of air blast freezers. The method was based on calculating the heat load and by building a target energy function for the system. The minimum energy consumption can theoretically be determined and predicted by the method presented here. This should help operators to operate the system optimally. Furthermore, competitiveness of the seafood processing company will be enhanced, since electricity consumption will be reduced. However, precise operating procedures may vary because our working model can be affected by many factors such as cooling temperature, air velocity, product temperature, product shapes, as well as mechanical and thermodynamic features of the freezer.

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