ANALYSIS OF THE ENERGY EFFICIENCY OF THE FAST FREEZING OF BLACKCURRANT BERRIES

I. A. Korotkiy

Kemerovo Institute of Food Science and Technology,
bul'v. Stroitelei 47, Kemerovo, 650056 Russia,
phone/fax: +7 (3842) 73-43-44, e-mail: krot69@mail.ru

(Received May 5, 2014; Accepted in revised form May 23, 2014)

Abstract: In this paper, some results of studying the energy efficiency of the fast freezing of different varieties of blackcurrant berries in a fluidized-bed fast freezer were reported. A method of calculating the energy expenditures on the fast freezing of different varieties of blackcurrant berries in an air fast freezer was proposed. The energy expenditures on the circulation of air at a rate required to create fluidization were determined depending on the air temperature. The energy consumption in the production of artificial cold for the provision of required heat-withdrawing air medium temperatures was calculated. The performed studies were used as a basis to determine the energy-efficient regimes of the low-temperature treatment of blackcurrant berries in an air fast freezer and also the types of a refrigerating machine and a refrigerant, which provided the least energy-consuming fast freezing of blackcurrant berries.

Keywords: blackcurrant, fast freezing, energy efficiency of freezing processes

UDC 664.8.037.5:634.721
DOI 10.12737/5454

INTRODUCTION

The deficit of vitamins and minerals is currently the most widespread and, at the same time, most unwholesome deviation of nutrition from the rational physiologically substantiated standards [1].

Being a natural concentrate of bioactive substances, berries manifest physiologically active properties after entering a human organism and produce an essential effect on its metabolism and vital activity. Blackcurrant is one of the most valuable vitamin-containing plants of the Russian flora. It is rich in ascorbic acid, vitamins B and P. The outstanding value of blackcurrant berries is explained by that vitamins C and P contained in them in great amounts mutually potentiate their health-promoting effects [2].

Freezing is one of the simplest and most widespread methods for the preservation of moisture-containing products. Frozen berries can be stored for many months, as the moisture in them has been brought into a solid state. A decrease in temperature and dewatering during the transition of the moisture contained in berries into a solid state create unfavorable conditions for the development of biochemical reactions in a frozen object, and their rate is abruptly decelerated [3].

The formation of ice crystals alongside with the freezing of a moisture-containing object leads to the destruction of its structure. The destruction of a product's structure is induced by both mechanical and osmotic factors. Ice crystals formed outside cells deform and destruct their membranes, growing in size during the process of freezing. Moreover, the growth of ice crystals in the intercellular space leads to the diffusion of cellular moisture through membranes and the dewatering of cells.

The intensity and character of transformations in a product under freezing depends on the conditions and parameters of the process and the qualitative characteristics of a low-temperature treatment object. The intensification of heat withdrawal in the process of freezing is accompanied by an increase in the amount of crystallization seeds, and this in turn promotes the formation of a microcrystalline structure. The more intense is heat withdrawal, the smaller will be the crystals in a frozen product [4]. In this case, the crystalline structure will be uniform, and ice crystals will form in both the intercellular space and the cells themselves.

The withdrawal of heat in the freezing of berries can be intensified either by decreasing the temperature of a heat-withdrawing medium or by accelerating its circulation. In the first case, an increase in heat withdrawal will have an extensive character due to an increase in the temperature difference between a low-temperature treatment object and a heat-withdrawing medium. In the second case, an increase in heat withdrawal will have an intensive character due to an increase in the coefficient of heat transfer between a freezable object and a refrigerating medium [5].

The intensification of heat exchange in the freezing of berries is accompanied by the growth of energy consumption in both the first and second cases. To intensify the heat withdrawal in real berry freezing processes, the cumulative effect of the two above listed factors is used.

An important question in the development of a low-temperature treatment technology is the energetic component in the primecost of the finished fresh-frozen fruit and berry products. For this reason, the ratio of the thermal and convective factors is a determinative element in the optimization of energy consumption in the low-temperature treatment of berries.

The objective of this work was to determine the low-temperature treatment regimes, which would allow the minimization of energy consumption in the production of fast-frozen blackcurrant berries.

OBJECTS AND METHODS OF STUDY

The fast freezing of berries is performed in fast freezers, which represent devices able to provide a high air circulation rate and required low-temperature treatment temperatures.

To determine the critical and optimal parameters of air motion in a freezer, it is necessary to have the data characterizing the mass and volumetric parameters of fruits and berries. Such characteristics of the studied varieties of berries are listed in Table 1.

Table 1. Mass and volumetric characteristics of the studied varieties of fruits and berries

Variety	Product unit mass, g	Product density, kg/m^3	Bulk density, kg/m^3	Bed porosity	Product unit diameter, mm
Pamyat' Lisavenko	1.4	1067	741	0.298	13–14
Seyanets Golubki	1.1	1082	751	0.293	12–13
Pamyat' Shukshina (Olimpiiskaya)	0.9	1070	743	0.290	11–12
Chernyi zhemchug	1.7	1075	746	0.306	14–15
Krasa Altaya	1.1	1059	735	0.295	12–13
Pushistaya	0.8	1063	738	0.291	11–12

The critical air velocities (w'_{cr}, w''_{cr}) are determined in compliance with the following method [6].

The critical air velocity w'_{cr} characterizes the beginning of fluidization.

$$w'_{cr} = \frac{\nu_{air}}{d_e} \times \frac{\mathrm{Ar}}{1400 + 5.22\sqrt{\mathrm{Ar}}}, \qquad (1)$$

where ν_{air} is the air kinematic viscosity, m^2/s, and d_e is the spherical product part's diameter.

The Archimedes number is determined by the formula

$$\mathrm{Ar} = \frac{g \cdot d_e \cdot \rho_{pr}}{\nu_{air}^2 \cdot \rho_{air}}, \qquad (2)$$

where g = 9.8 m/s^2 is the gravity acceleration, and ρ_{pr} and ρ_{air} are the product and air density, respectively.

The critical air velocity w''_{cr} is a velocity, at which the entrainment of a product is possible, and

$$w''_{cr} = \frac{\nu_{air}}{d_e} \times \frac{\mathrm{Ar}}{18 + 0.6\sqrt{\mathrm{Ar}}}. \qquad (3)$$

To determine the freezing time for moisture-containing food products, the Planck formula is most widely used [7]:

$$\tau_{fr} = \frac{q_{fr} \cdot \rho_{pr}}{t_{cryo} - t_{air}} \times \frac{d_e}{6}\left(\frac{d_e}{4\lambda_{fr}} + \frac{1}{\alpha}\right), \qquad (4)$$

where α is the product-to-air heat-transfer coefficient, W/(m^2·K), λ_{fr} is the heat conductivity of the product's frozen part, W/(m·K), ρ_{pr} is the product density, kg/m^3, q_{fr} is the specific heat withdrawn from a product in the process of freezing, J/kg, t_{cryo} is the cryoscopic temperature, °C, and t_{air} is the air temperature.

The heat-transfer coefficient is determined depending on the velocity and regime of the air flow, its thermodynamic parameters, and the shape and size of a product.

According to the theory of similarity, the heat-transfer coefficient can be calculated by the formula [8]

$$\alpha = \mathrm{Nu} \cdot \lambda_{air} / d_e, \qquad (5)$$

where λ_{air} is the air heat conductivity, and Nu is the Nusselt number.

The Nusselt number for the heat transfer in a fluidized bed can be determined from the empirical equation [9]

$$\mathrm{Nu} = 0.03 \cdot \mathrm{Pr}^{1/3} \cdot \mathrm{Re}, \qquad (6)$$

where $\mathrm{Pr} = c_p \cdot \mu_{air}/\lambda_{air}$ is the Prandtl number, μ_{air} is the air dynamic viscosity, Pa·s, c_p is the air specific heat capacity, J/(kg·K), $\mathrm{Re} = \omega \cdot d_e \cdot \rho_{air}/\mu_{air}$, ω is the air velocity, m/s, and ρ_{air} is the air density, kg/m^3.

The air amount m_{air} (kg) required to freeze a kilogram of berries is determined by the formula

$$m_{air} = \frac{\Delta h}{c_p \cdot \Delta t_{air}}, \qquad (7)$$

where Δh is the difference between the enthalpies of berries before and after freezing (at 10 and –18°C, J/kg), and Δt_{air} is the air heating, K.

The air heating in the freezing of berries is found as

$$\Delta t_{air} = \alpha \cdot F_{pr} \cdot \Delta t_m, \qquad (8)$$

where F_{pr} is the surface area of berries, m^2, and Δt_m is the logarithmic mean temperature difference between the air and freezable berries. In turn, Δt_m can be calculated as

$$\Delta t_m = \frac{t_{air2} - t_{air1}}{\ln \dfrac{t_{cryo} - t_{air1}}{t_{cryo} - t_{air2}}}, \qquad (9)$$

where t_{cryo} is the cryoscopic temperature of berries, t_{air1} is the initial air temperature, t_{air2} is the final air temperature, and $\Delta t_{air} = t_{air2} - t_{air1}$.

Since the initial air temperature (t_{air1}) is specified, the final air temperature (t_{air2}) can be determined by Eq. (9).

The air volume required to freeze a kilogram of berries is calculated as

$$V_{air} = m_{air}/\rho_{air}. \quad (10)$$

The energy (L_{fan}), which should be spent on the circulation of this air volume at a required rate can be found as

$$L_{fan} = V_{air}\cdot\Delta P/\eta_{fan}, \quad (11)$$

where η_{fan} is the fan efficiency coefficient, which is nearly η_{fan} = 0.76 for centrifugal fans operating under the conditions of a fast freezer, ΔP is the aerodynamic resistance to the motion of air in the circuit of a fast freezer.

The aerodynamic losses during the motion of air in a fast freezer occur in the fluidized bed, the supporting grid, the finned air cooler sections, upon the turns of the air flow at the fan's inlet, and in the fan's diffuser. The greatest aerodynamic losses take place in the fluidized bed, the supporting grid, and the air cooler sections. The other listed regions of the air circuit structurally belong to the fan, and the aerodynamic losses in them are small in comparison with the other regions, so they may be included into the fan efficiency coefficient instead of being taken into consideration in particular calculations.

The aerodynamic resistance of the fluidized bed (ΔP_{fl}, Pa) can be determined by the following method [6]:

$$\Delta P_{fl} = 1.67\left(\mathrm{Re}\frac{H_{fl}}{d_e}\right)^{0.2} \times \frac{G_{pr}}{F_{pr}}, \quad (12)$$

where H_{fl} is the fluidized product bed height, G_{pr} is the product mass, and F_{pr} is the surface area occupied by a product on the grid, m^2.

$$H_{fl} = H_0\left(\frac{1-\varepsilon_0}{1-\varepsilon}\right), \quad (13)$$

where ε_0 is the bulk bed porosity (Table 1), $\varepsilon = \left(\frac{18\,\mathrm{Re} + 0.36\,\mathrm{Re}^2}{\mathrm{Ar}}\right)^{0.21}$ is the fluidized bed porosity, and H_0 is the bulk product bed height, m.

The aerodynamic resistance of the supporting grid with meshes of 3 × 3 mm in size and the free air flow area E = 0.308 (ΔP_g, Pa) is

$$\Delta P_g = 13.72\cdot w^2 - 43.12\cdot w + 119.36, \quad (14)$$

where w is the air flow velocity, m/s.

The aerodynamic resistance of the finned air cooler section (ΔP_{ac}, Pa) is found from the dependence

$$\Delta P_{ac} = 1.35\cdot A\cdot \mathrm{Re}^{-0.24}\rho_{air}\cdot w^2, \quad (15)$$

where A is the coefficient taking into account the structural features of the air cooler.

Hence, the aerodynamic resistance to the air flow in the circulation circuit of the air cooler (ΔP, Pa) can be found from the equation

$$\Delta P = (\Delta P_{fl} + \Delta P_g + \Delta P_{ac})\alpha, \quad (16)$$

where α = 1.1 is the coefficient taking into account the friction resistances to the air flow.

The working fluid in the cycles of refrigerating machines participates in different thermodynamic processes. The efficiency of refrigerating machines depends on the fashion, in which these processes are performed. The problem of thermodynamic analysis based on the first and second laws of thermodynamics is to ascertain the possible efficiency of refrigerating machine cycles [10].

The refrigeration efficiency of one-stage, two-stage, and cascade refrigerating machines operating on freons R-134a, R-22, R-404a, R-23, and ammonia was studied. The energy expenditures on the production of artificial cold were estimated by the method described in [11].

The ambient medium is of great importance in the thermodynamic theory of refrigerating machines. The ambient medium is first of all characterized by the independence of its parameters on the operation of a considered refrigerating machine. This means that any action of a refrigerating machine produces no changes in the ambient medium. The atmospheric air with a temperature from 15 to 35°C was considered as such a medium.

An important condition for the implementation of a refrigeration engineering solution is the organization of heat exchange between a working fluid and the ambient medium, and also a cold-carrying agent, which performs the transfer of heat from a freezable object to a working fluid without appreciable expenditures. To perform the thermal analysis of the efficiency of refrigeration cycles, the temperature difference between a working fluid and the ambient medium in the evaporator was taken equal to 10°C, and the temperature difference between a working fluid and the air leaving the air cooler was also set equal to 10°C.

The working cycle of a one-stage compression refrigerating machine is shown in Fig. 1a in the *p–i* (pressure–enthalpy) coordinates. In this cycle, *1–2* is compression in the compressor, *2–3′* is cooling of a cooling agent and condensation in the condenser, and *3′–3* and *a–1* are heat regeneration in the recuperative heat exchanger of freon refrigerating machines. Heat regeneration in ammonia refrigerating machines is unreasonable. Process *3–4* is expansion in an expanding device. Process *4–a* is boiling of a cooling agent in the evaporator.

The overheating of a working fluid before the compressor is estimated by the formula $\Delta T_{oh} = T_1 - T_a$. For ammonia refrigerating machines, the overheating of a working fluid at suction onto the compressor is taken to be ΔT_{oh} = 5–10°C. Let us set ΔT_{oh} = 10°C. For freon refrigerating machines, ΔT_{oh} = 15–35°C. Let us set ΔT_{oh} = 30°C.

The overcooling before expansion is calculated as follows. For regeneration cycle refrigerating machines, it is determined from the energy balance of a recuperative heat exchanger

$$h_L^{in} - h_L^{out} = h_V^{out} - h_V^{in}. \quad (17)$$

For non-regeneration cycle (ammonia) refrigerating machines, the temperature of the liquid cooling agent before the expanding valve is taken to be 1–3°C lower than the temperature of the liquid leaving the condenser (t_C), i.e., $t_L = t_C - (1–3°C)$.

The specific refrigeration capacity of a refrigerating machine q_0, the specific adiabatic compression work in the compressor l_s, and the real mass and volumetric capacities of the compressor G_{real} and V_{real} are then determined.

The specific refrigeration capacity of a refrigerating machine will be

$$q_0 = h_a - h_4 . \quad (18)$$

The specific adiabatic work of the compressor is determined from the formula

$$l_{ad} = h_2 - h_1 . \quad (19)$$

The power of a refrigerating machine N_e (kW), which should be provided to withdraw a certain amount of heat per unit time Q_0 (kW) from a freezable object is determined by the following method:

$$N_e = N_i + N_{fr} , \quad (20)$$

where N_i is the indicated power of the compression of a working fluid in the compressor, and N_{fr} is the power spent on friction and the driver of auxiliary devices,

$$N_i = G_{real}\, l_{ad} / \eta_i , \quad (21)$$

where G_i is the mass flow rate of a cooling agent circulating in a refrigerating machine (kg/s), and η_i is the indicated efficiency coefficient of the compressor.

The friction power is determined from the empirical formula

$$N_{fr} = p_{i,fr} V_t , \quad (22)$$

where $p_{i,fr} = (40–90)\cdot 10^3$ Pa is the friction pressure, and V_t is the theoretical volumetric capacity of the compressor (m^3/s).

The mass flow rate of a cooling agent is determined as

$$G_{real} = Q_0 / q_0 . \quad (23)$$

The theoretical volumetric capacity is found by the formula

$$V_t = G_{real}\, v_1 / \lambda , \quad (24)$$

where v_1 is the specific volume of a working fluid sucked into the compressor, λ the delivery coefficient of the compressor, $\lambda = f(p_{frcd}/p_{suct})$. Here p_{frcd} is the pressure, to which a working fluid is compressed immediately in the working members of the compressor, and p_{suct} is the pressure of the working fluid entering immediately into the working space of the compressor.

The method of the estimation of λ is described in [11].

A vapor-compression refrigerating machine can be implemented in different designs. We have considered a refrigerating machine in the simplest and, consequently, most acceptable implementation for the supply of a self-contained fast freezer with cold.

The working cycle of a two-stage compression single-expansion refrigerating machine is shown in Fig. 1b in the *p–i* coordinates. In this cycle, *1–2a* is compression in the compressor of stage *I*, *2a–1a* is cooling in the interstage cooler due to the transfer of compression heat to the ambient medium, *2–3′* is compression in the compressor of stage *II* and cooling of a cooling agent and condensation in the condenser, and *3′–3* and *a–1* are regeneration of heat in the recuperative heat exchanger of freon refrigerating machines. In ammonia refrigerating machines, $t_3 = t_3 - (1–3°C)$. Process *3–4* is expansion in the expanding device. Process *4–a* is boiling of the cooling agent in the evaporator.

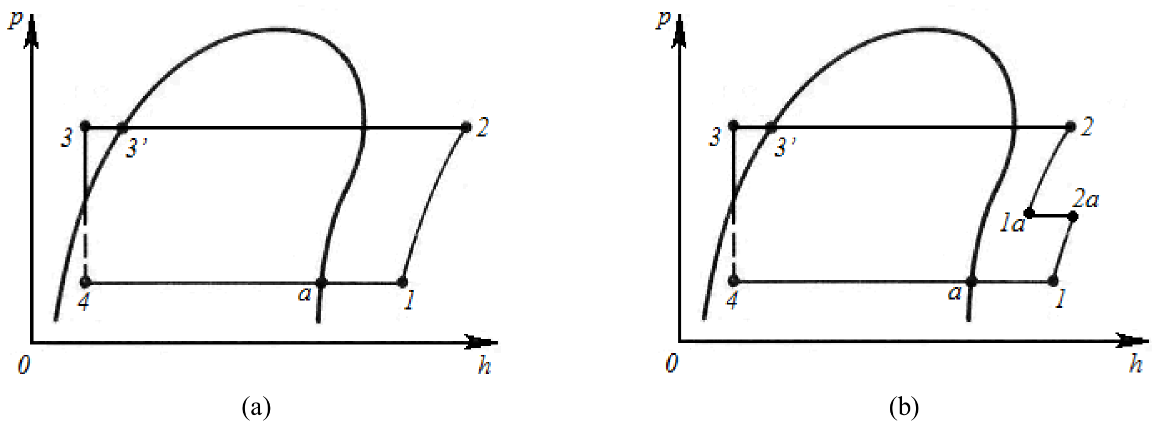


Fig. 1. Theoretical cycle of a refrigerating machine in the *p–h* (pressure–enthalpy) coordinates: (a) one-stage compression cycle, (b) two-stage compression cycle.

The method of calculating the cycle of a two-stage compression refrigerating machine principally corresponds to the method of calculating the cycle of a one-stage compression refrigerating machine, but has some peculiarities.

The interstage pressure is selected by the formula

$$p_{int} = \sqrt{p_1 p_2} , \quad (25)$$

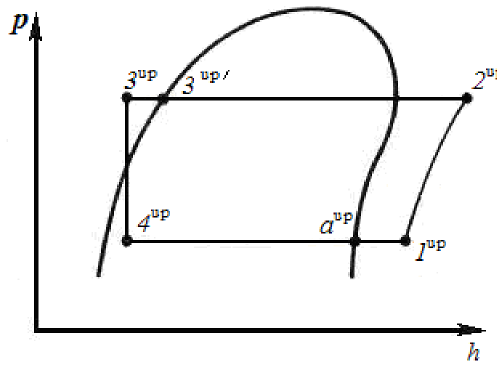
or from the condition of the maximally admissible compression end temperature for a studied cooling agent.

The temperature of point *1a* is determined from the ambient temperature and the condition of the undercooling of a cooling agent in the interstage cooler due to heat underrecuperation.

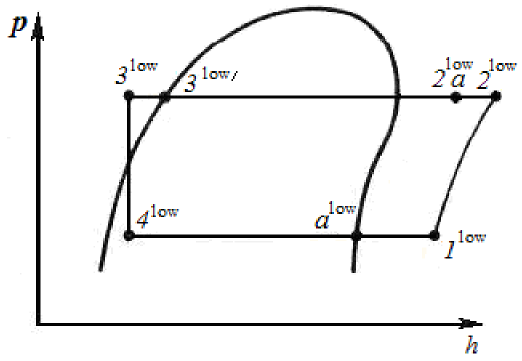
The two-stage compression work is determined by the formula

$$l_{comp} = l_I + l_{II} = (h_{2a} - h_1) + (h_2 - h_{1a}) . \qquad (26)$$

In cascade refrigerating machines, at least two working fluids are used. A two-cascade refrigerating machine consists of two independent refrigeration cycles called branches. The interaction between the branches of a cascade refrigerating machine occurs in heat exchangers. In the lower low-temperature branch, a high-pressure working fluid is used (we used freon R-23 with a normal boiling temperature of –82.14°C). The cooling agent of the upper branch was freon R-22 with a normal boiling temperature of –40.81°C. The upper branch of a cascade refrigerating machine is designed for the withdrawal of condensation heat from the cooling agent of the lower branch and its transfer to the ambient medium. The working cycle of a cascade refrigerating machine is shown in Fig. 2.



(a)



(b)

Fig. 2. Theoretical cycle of a two-cascade refrigerating machine: (a) upper (high-temperature) cascade branch, (b) lower (low-temperature) cascade branch.

Here, 1^{low}–2^{low} is compression in the compressor of the lower cascade branch, 2^{low}–$2^{low}a$ is cooling of a cooling agent in the heat exchanger due to the withdrawal of heat into the ambient medium, $2^{low}b$–$3^{low\prime}$ is condensation of the cooling agent of the lower cascade branch in the evaporative condenser due to the withdrawal of condensation heat by the upper cascade branch, $3^{low\prime}$–3^{low} is overcooling of the liquid cooling agent of the lower branch in the regenerative heat exchanger due to heat exchange with the cooling agent sucked into the compressor (a^{low}–1^{low}), 3^{low}–4^{low} is expansion in the expanding device, 4^{low}–a^{low} is boiling in the evaporator of the lower branch. The upper cascade branch works by the cycle similar to the cycle of a one-stage compression refrigerating machine, but process 4^{up}–a^{up} is boiling of the cooling agent of the upper cascade branch in the evaporative condenser due to the delivery of heat from the condensing cooling agent of the lower cascade branch. The temperature difference between the cooling agents of the branches due to underrecuperation is 3–5°C.

The refrigerating capacity of a cascade refrigerating machine is controlled by the refrigerating capacity of the lower cascade branch, and the work expenditures correspond to the total work expenditures in both cascades.

When determining the work expenditures on the freezing of fruits and berries, it was necessary not only to take into account the work expenditures on the withdrawal of crystallization heat from a product and the heat spent on the cooling of fruits and berries before and after crystallization. It was also necessary to take into consideration the heat inflow through the heat-insulating enclosures of a freezer Q_1 and the heat inflow from the air entering a freezer through the charge and discharge ports Q_4.

The heat inflow Q_1 is determined by the formula

$$Q_1 = \Sigma(k_i F_i)\Delta t , \qquad (27)$$

where k is the heat-transfer coefficient of heat-insulating enclosures and depends on the type and thickness of heat insulation, the averaged value k = 0.21 W/(m^2 k) is taken in our calculations; F is the surface area of heat-insulating enclosures and depends on the dimensions and shape of a freezer, and Δt is the temperature difference between the ambient air and the air in a fast freezer.

Since the surface area of heat-insulating enclosures can not be taken into account without the knowledge of the real dimensions of a fast freezer, the parameter $\Sigma(k_i F_i)$ was replaced by the specific heat flux through the enclosures per unit mass of a freezable product q_1. This parameter was taken to be q_1 = 90 W/(kg K) from averaged values.

The heat inflow from the air entering and leaving a freezer through the charge and discharge ports was taken from the recommendations $Q_4 = 0.4Q_1$.

RESULTS AND DISCUSSION

The experimental data given in Table 1 and Eqs. (1)–(3) were used to obtain the critical velocities of the studied fruit varieties. The results of calculations are plotted in Fig. 3.

From the performed calculations for the studied varieties of berries and fruits, it follows that the range of velocities, at which the phenomenon of fluidization takes place (from the appearance of a fluidized bed to the velocity, at which the entrainment of fruits and berries is possible), is 1.24–17.7 m/s at temperatures from –43 to –13°C for all the studied varieties. Since every variety of fruits and berries has its own range of air velocities, at which the process of fluidization is stable, we have used for studies the range of velocities, at which the process of fluidization without the entrainment of fruits and berries is ensured for all the studied varieties. These air velocities were from 2 to 11.5 m/s.

Using Eqs. (4)–(6), the freezing time was calculated for the studied varieties of blackcurrant berries at different air velocities and temperatures, and the results of calculations were plotted in Fig. 4.

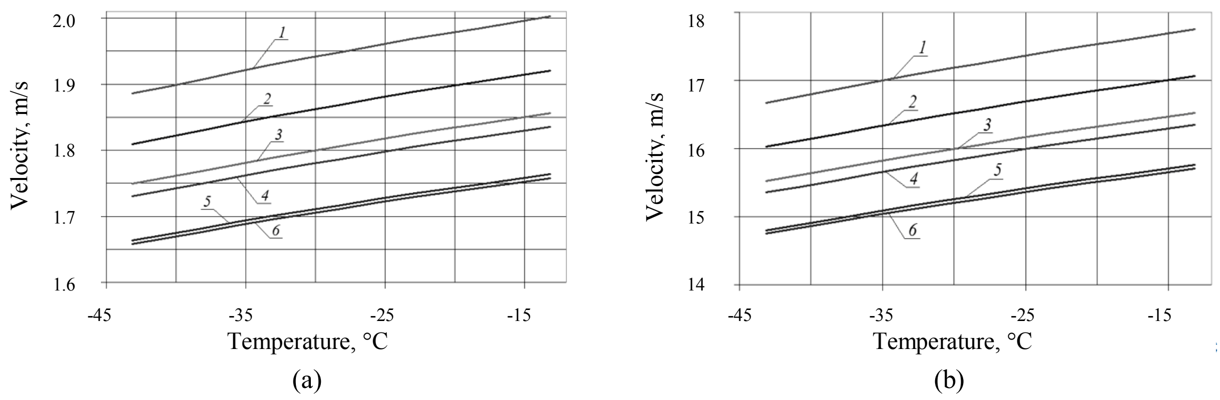


Fig. 3. Critical fluidization velocities of blackcurrant berries versus temperature: (*1*) Chernyi zhemchug, (*2*) Pamyat' Lisavenko, (*3*) Seyanets Golubki, (*4*) Krasa Altaya, (*5*) Pamyat' Shukshina, (*6*) Pushistaya, (a) fluidization beginning velocity, (b) berry entrainment velocity.

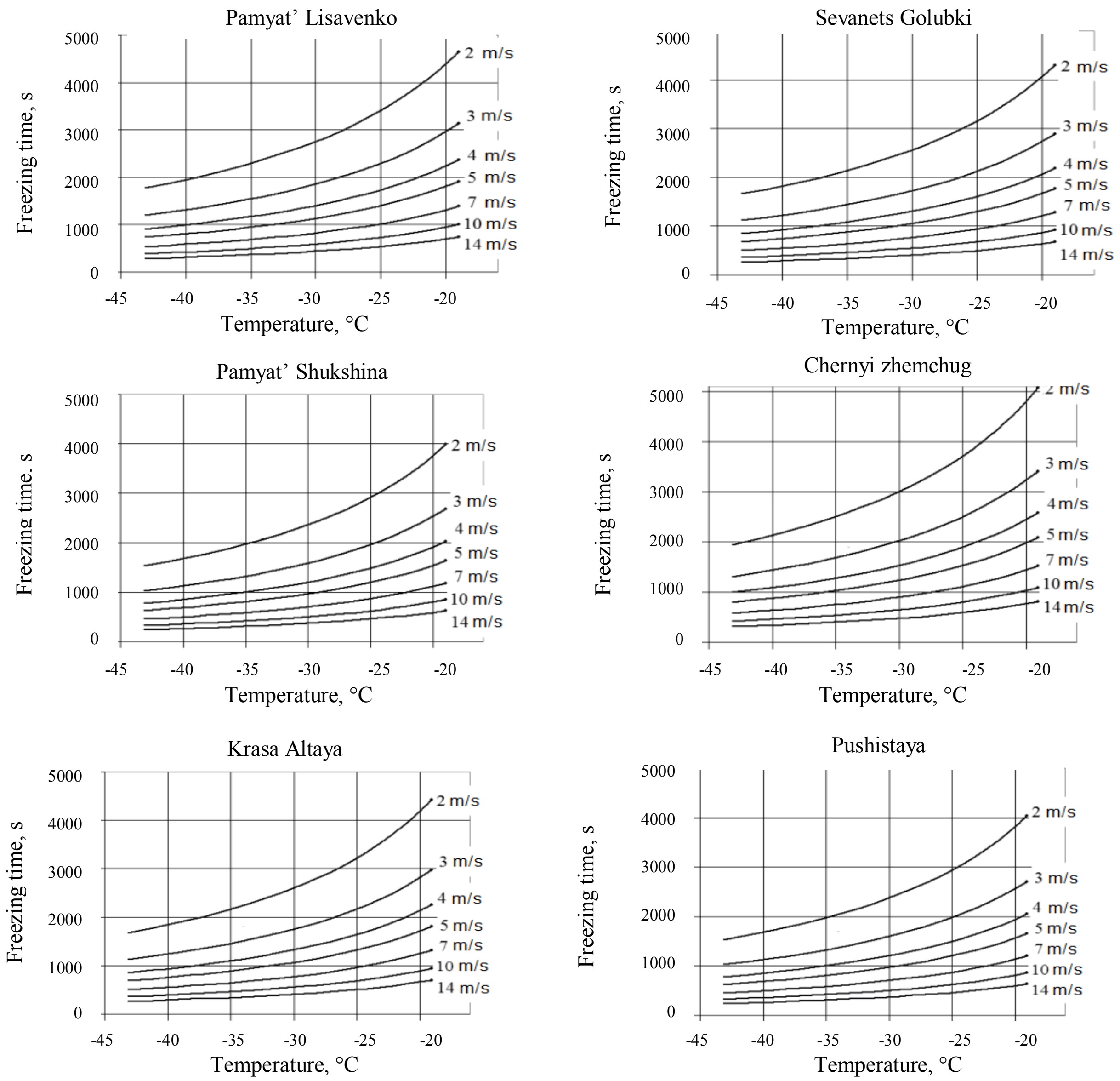


Fig. 4. Freezing time versus temperature for the freezing of the studied varieties of blackcurrant berries from the initial temperature of 10°C to the temperature of –18°C in a fluidized-bed fast freezer at different air velocities.

Using Eqs. (7)–(16), the energy expenditures on the circulation of air in a fast freezer were calculated depending on the air temperature and velocity. The energy expenditures were calculated for the freezing of blackcurrant berries from the initial temperature t_{init} = 10°C to the temperature t_{fin} = –18°C. The results of calculations are plotted in Fig. 5.

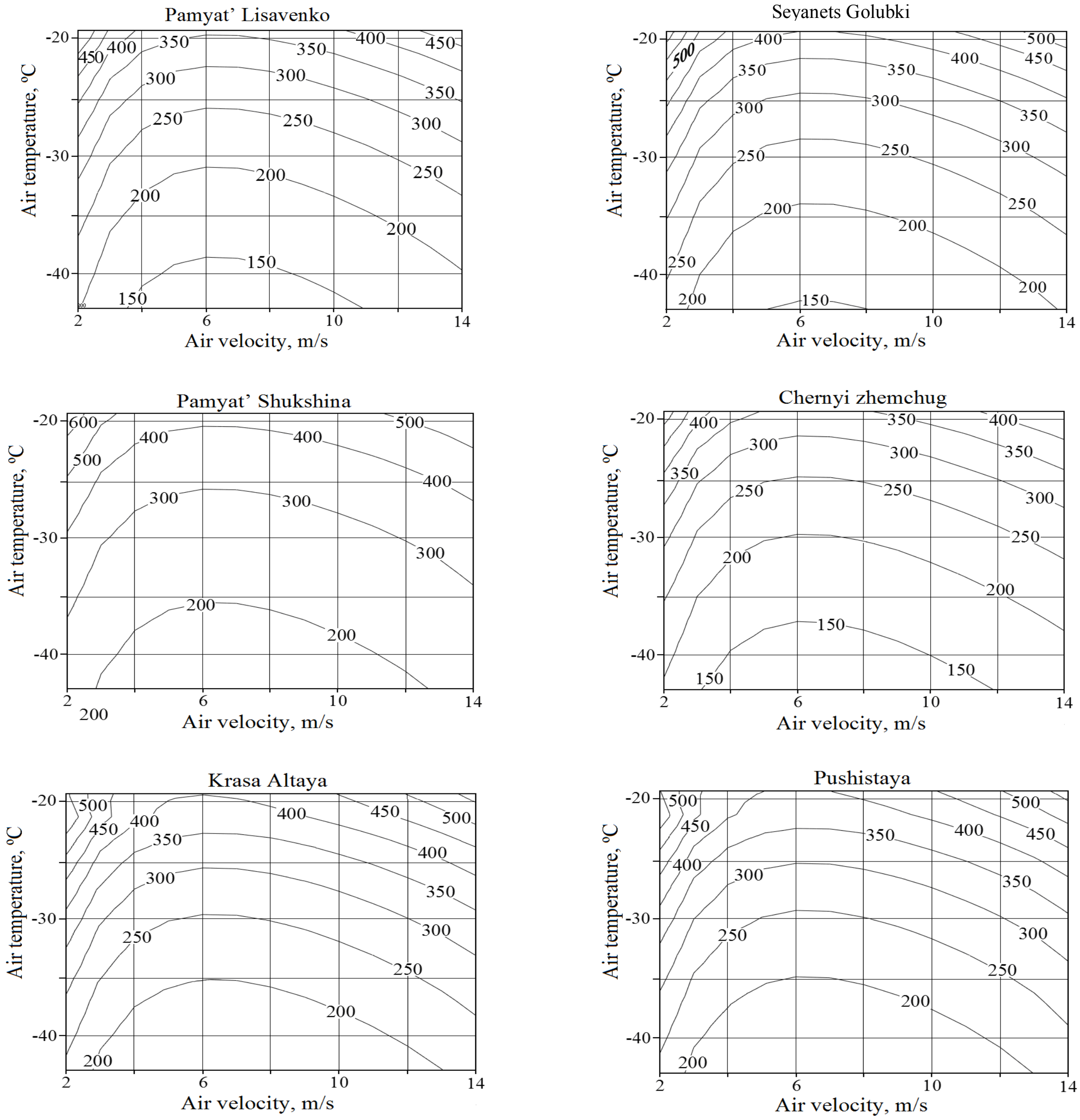


Fig. 5. Energy expenditures (kJ/kg) on the circulation of air in a fast freezer depending on the air velocity and temperature for the freezing of a kilogram of blackcurrant berries of studied varieties from the initial temperature of 10°C to the temperature of –18°C.

The results of calculations indicate the existence of a certain range of air velocities, at which the air circulation energy expenditures are minimal at a certain freezing air temperature for the studied varieties of blackcurrant berries. Thus, the optimal range of velocities for blackcurrant berries is 6–7 m/s.

An increase in the velocity of air passing through the bed of berries intensify the transfer of heat from freezable fruits and decreases the freezing time, so a smaller circulating air amount is required for the freezing of berries. However, the aerodynamic losses grow proportionally to the squared air velocity. Hence, an increase in the air velocity decreases the required air flow rate for the withdrawal of heat in the process of freezing and, consequently, the energy expenditures on the transport of a necessary air amount. At the same time, an increase in the air velocity leads to the growth of energy expenditures due to the need to overcome aerodynamic resistances. Hence, at the initial stage, a decrease in energy expenditures due to a reduction in the amount of circulating air compensates an increase in the energy spent to overcome aerodynamic resistances. When the air velocity attains the values exceeding the optimal level, the growth of the energy expenditures required to overcome aerodynamic resistances become quicker than a decrease in the energy expenditure due to a reduction in the required amount of circulating air, so the total energy expenditures on the circulation of air grow.

The calculated required specific refrigeration capacity of a refrigerating machine for the freezing of a kilogram of blackcurrant berries as a function on the air temperature in a fast freezer and the ambient air temperature is plotted in Fig. 6.

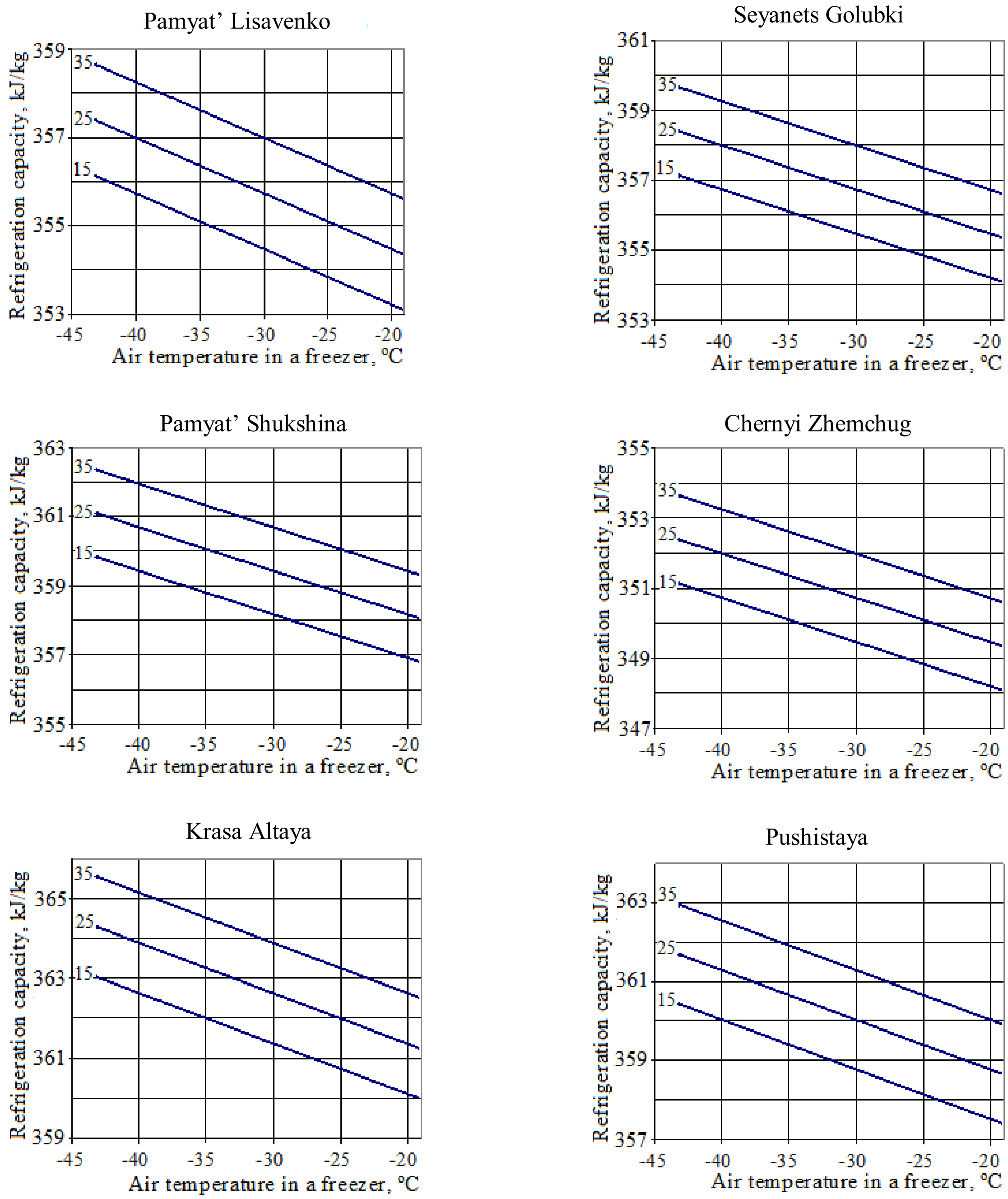


Fig. 6. Required refrigeration capacity (kJ/kg) of a refrigerating machine for the freezing of a kilogram of blackcurrant berries from the temperature of 10°C to the temperature of –18°C depending on the air temperature in a fast freezer and the ambient air temperature.

The calculation results plotted in Fig. 6 for the required refrigeration capacity of a refrigerating machine depending on the ambient medium temperature and the air temperature in a freezer indicate that the required refrigeration capacity of a refrigerating machine for the freezing of different varieties of the same species of berries and fruits differs slightly.

The required refrigeration capacity of a refrigerating machine for the freezing of a kilogram of blackcurrant berries differs by less than 1.9% depending on their variety.

The comparative analysis of the energy efficiency of different refrigeration schemes used for the freezing of fruits and berries was performed using Pamyat' Lisavenko blackcurrant berries.

The results of the comparative calculations of the energy expenditures on the freezing of berries in one-stage refrigerating machines are shown in Fig. 7.

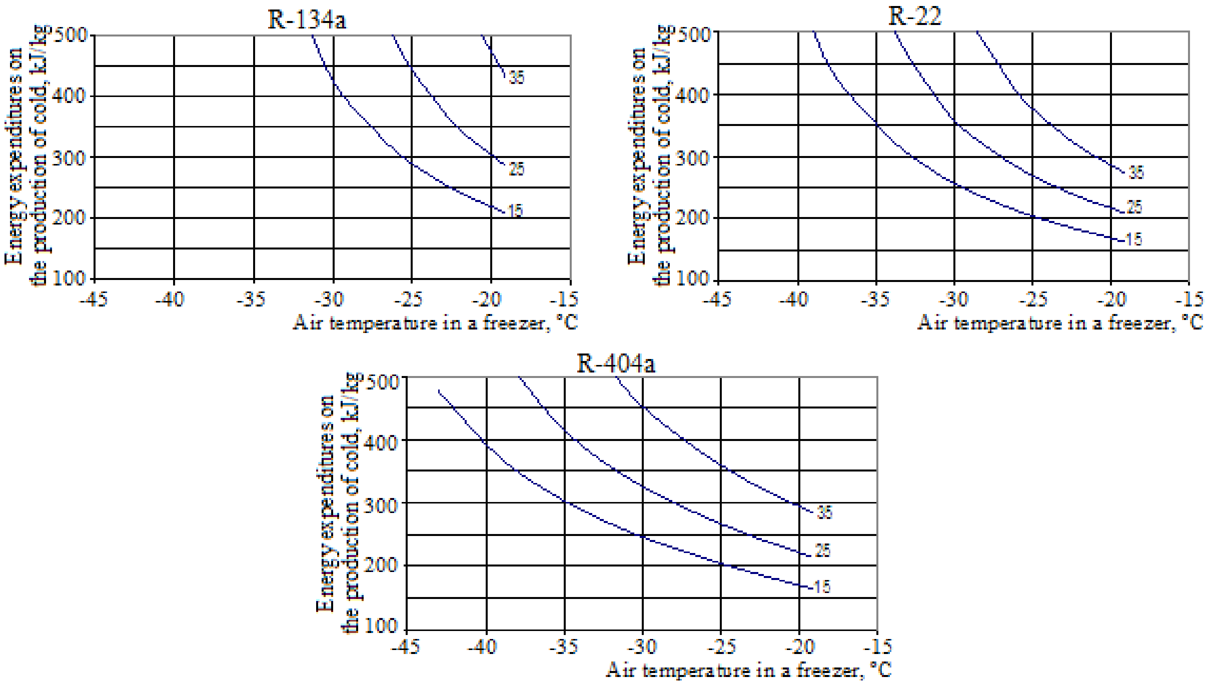


Fig. 7. Energy expenditures (kJ/kg) on the driver of a one-stage refrigerating machine for the freezing of a kilogram of Pamyat' Lisavenko blackcurrant berries from the temperature of 10ºC to the temperature of −18ºC depending on the air temperature in a fast freezer and the ambient air temperature.

The use of a one-stage refrigerating machine operating on freon R-134a is unreasonable, as the production of cold of a temperature level below –20°C in a refrigerating machine of this type is attended by unjustifiable high energy expenditures.

The energy expenditures on the production of artificial cold with a cooling source temperature above –25°C in a one-stage refrigerating machine operating on freons R-22 and R-404a are at the same level.

A decrease in the temperature of a cooling medium in a fast freezer below –25°C leads to the abrupt growth of fast freezer below –25°C leads to the abrupt growth of energy expenditures in a refrigerating machine operating on freon R-22. A one-stage refrigerating machine operating on freon R-404a can be used to obtain a temperature level above −30ºC in a fast freezer.

The results of calculating the energy expenditures on the production of cold required to freeze a kilogram of berries in a two-stage refrigerating machine depending on the air temperature in a fast freezer and the ambient medium temperature are shown in Fig. 8.

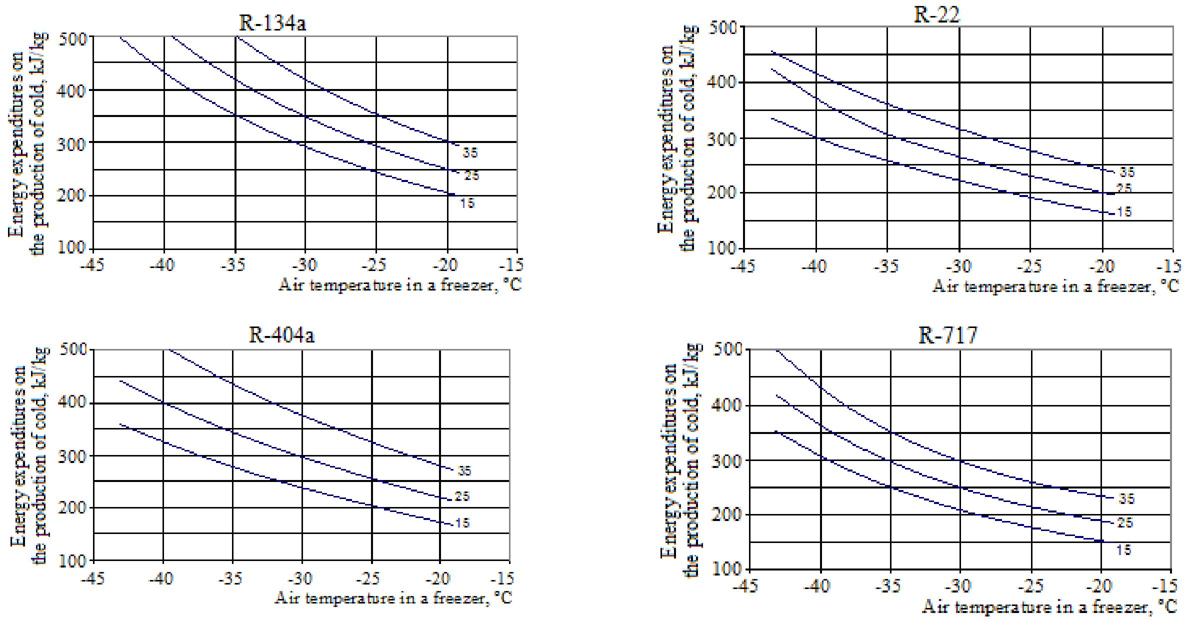


Fig. 8. Energy expenditure (kJ/kg) on the driver of a two-stage refrigerating machine for the freezing of a kilogram of Pamyat' Lisavenko blackcurrant berries from the temperature of 10°C to the temperature of –18°C depending on the air temperature in a fast freezer and the ambient air temperature.

From the preformed calculations it can be seen that the use of freon R-134a is unreasonable for the fast freezing of fruits and berries, and the use of freon R-134a in a two-stage compression refrigerating machine leads to the highest energy consumption in comparison with the other two-stage refrigerating machines.

The energetic efficiency of a two-stage refrigerating machine operating on freon R-404a is lower than for a refrigerating machine operating on freon R-22. The energy efficiencies of an ammonia refrigerating machine and a refrigerating machine operating on freon R-22 nearly correspond to each other. A two-stage refrigerating machine has insignificant advantages within a temperature range to –35°C, and a two-stage refrigerating machine operating on freon R-22 is more energe-tically profitable at temperatures below –35°C.

However, a freon refrigerating machine has better performance characteristics and a lower cost in comparison with an ammonia refrigerating machine, so the use of a two-stage refrigerating machine operating on freon R-22 is more efficient in comparison with an ammonia refrigerating machine.

The energy expenditures on the supply of a fast freezer with cold from a cascade refrigerating machine are plotted in Fig. 9.

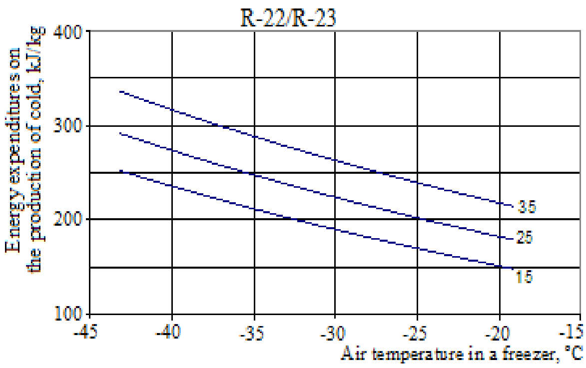


Fig. 9. Energy expenditures (kJ/kg) on the driver of a cascade refrigerating machine with freons R-23 and R-22 in the lower and upper cascade branches, respectively, for the freezing of a kilogram of Pamyat' Lisavenko blackcurrant berries from the temperature of 10°C to the temperature of –18°C depending on the air temperature in a fast freezer and the ambient air temperature.

Hence, a cascade refrigerating machine with freons R-22 and R-23 in the upper and lower cascade branches, respectively, has the best energetic characteristics among all the considered refrigeration schemes. How-ever, the practical implementation of such a scheme is most complicated, but has a number of advantages, the most important of which is lower energy expenditures on the production of artificial cold.

The energy expenditures on the attainment of temperatures below –30°C in the freezable volume of a cascade refrigerating machine and a two-stage refrigerating machine operating on freon R-22 differ slightly. As the refrigeration and operation scheme of a two-stage refrigerating machine is simpler and more reliable, it is possible to recommend the use of a two-stage refrigerating machine operating on freon R-22 for temperatures above –30°C. To attain a temperature level below –30°C, it is more preferable to use a cascade refrigerating machine.

The energy spent on the freezing of fruits and berries in a fast freezer is consumed by the driver of a refrigerating machine, i.e., for the organization of heat withdrawal from a heat-transfer agent into the ambient medium, and by the driver of fans, i.e., for the intensification of heat withdrawal from a freezable object to an intermediate cold-carrying agent—the air circulating in the freezer.

The performed studies show that the best energy efficiency can be attained at an optimal air medium velocity, which depends on the physical parameters and component composition of fruits and berries. A decrease in the air velocity, as well as an increase in this parameter with respect to an optimal value, leads to the growth of energy expenditures on the organization of heat withdrawal from a freezable object to the air.

A decrease in the air temperature in a fast freezer increases the efficiency of heat withdrawal. At the same time, a decrease in the air temperature in a fast freezer is accompanied by the growth of energy expenditures on the driver of a refrigerating machine. The cumulative effect of these two factors on the given energy expenditures on the freezing of a kilogram of berries is demonstrated in Fig. 10.

The regimes with minimum energy expenditures on the freezing of the studied varieties of fruits and berries at different ambient medium temperatures for cascade and two-stage refrigerating machines are characterized in Tables 2 and 3.

Table 2. Energetically optimal regimes of the freezing of the studied varieties of fruits and berries in a fast freezer with a cascade refrigerating machine

Variety	Air temperature in a freezer,°C			Air velocity, m/s			Energy expenditures on the freezing of a kilogram of berries, kJ			Freezing time, s		
	Ambient medium temperature, °C											
	15	25	30	15	25	35	15	25	35	15	25	35
Pamyat' Lisavenko	–39	–39	–39	6	6	6	379	417	459	695	695	695
Seyanets Golubki	–39	–39	–39	6	6	6	399	436	479	645	645	645
Pamyat' Shukshina	–43	–39	–39	6	6	6	410	448	491	527	593	593
Chernyi zhemchug	–39	–39	–35	6	6	6	369	406	447	761	761	866
Krasa Altaya	–43	–39	–39	6	6	6	410	448	491	578	651	651
Pushistaya	–43	–39	–39	6	6	6	406	444	487	529	596	596

Table 3. Energetically optimal regimes of the freezing of the studied varieties of fruits and berries in a fast freezer with a two-stage refrigerating machine

Variety	Air temperature in a freezer,°C			Air velocity, m/s			Energy expenditures on the freezing of a kilogram of berries, kJ			Freezing time, s		
	Ambient medium temperature, °C											
	15	25	30	15	25	35	15	25	35	15	25	35
Pamyat' Lisavenko	−31	−31	−31	6	6	6	429	473	524	913	913	913
Seyanets Golubki	−35	−31	−31	6	6	6	453	499	550	734	846	846
Pamyat' Shukshina	−35	−31	−31	6	6	6	466	515	567	675	778	778
Chernyi zhemchug	−31	−31	−31	6	6	6	416	460	510	998	998	998
Krasa Altaya	−35	−31	−31	6	6	6	466	515	567	742	857	857
Pushistaya	−35	−31	−31	6	6	6	462	510	562	679	784	784

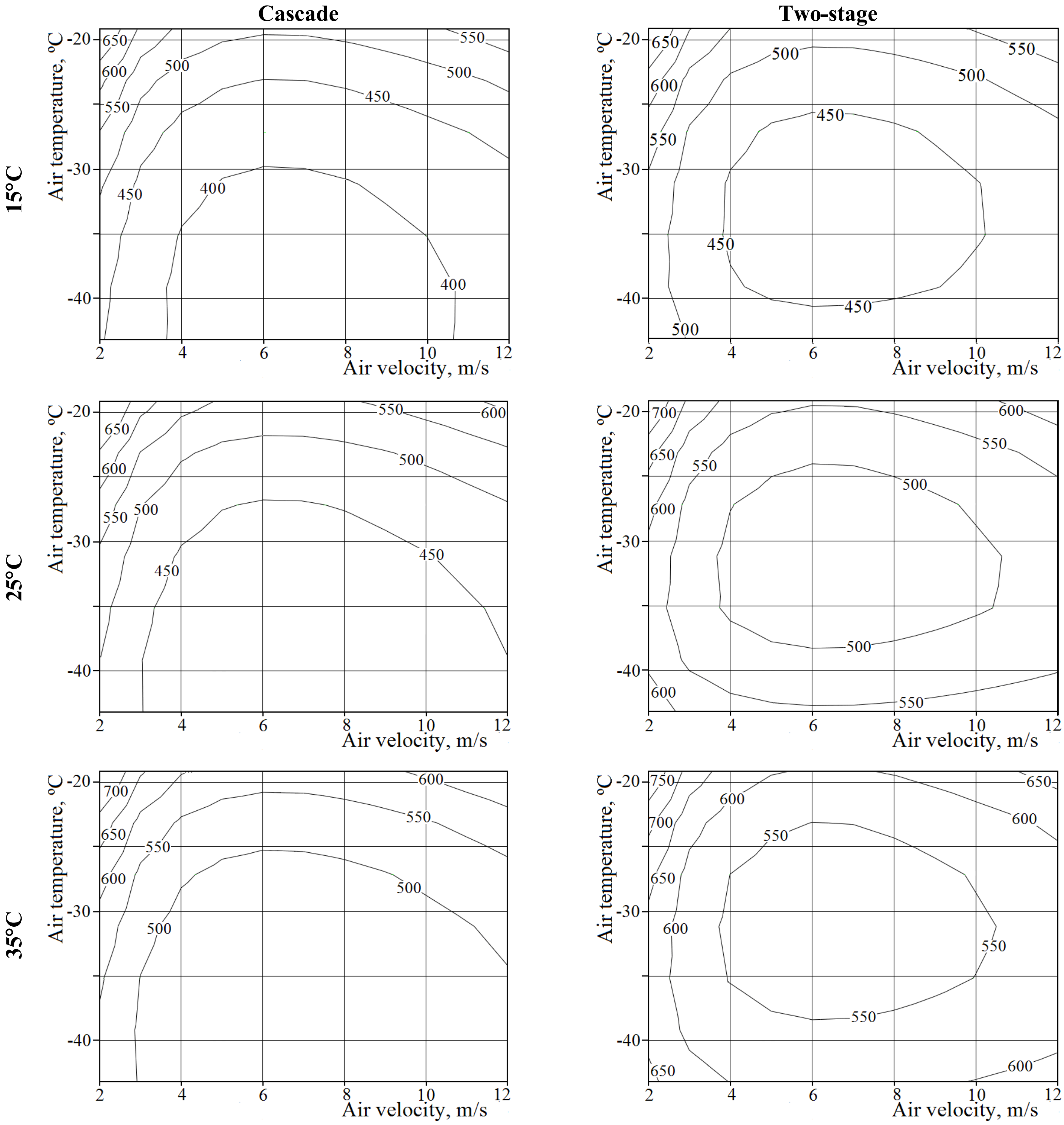


Fig. 10. Total energy expenditures (kJ/kg) on the freezing of Pamyat' Lisavenko blackcurrant berries in a fast freezer supplied with cold from cascade (R-23/R-22) and two-stage (R-22) refrigerating machines depending on the air temperature and velocity in a freezer and the ambient medium temperature.

From the presented results it can be seen that the air velocity of 6 m/s in a fluidized-bed fast freezer is least energy consuming for the freezing of all the varieties of blackcurrant berries. The air temperature in a freezer of –39ºC and –31ºC is the optimal temperature regime of fast freezing in the case of cold supply from a cascade refrigerating machine and a two-stage refrigerating machine with freon R-22 as a cooling agent, respectively.

The energy expenditures on the freezing of blackcurrant berries in a two-stage refrigerating machine are in average 14.2% higher than in a cascade refrigerating machine. The average time of the freezing of blackcurrant berries is 863 s at a temperature of –31°C and 657 s at a temperature of –39°C.

Hence, the performed calculations and the analysis of obtained results allow us to conclude that the use of cascade refrigerating machines is much less energy consuming in comparison with the other types of refrigerating machines for the fast freezing of fruits and berries. Moreover, the freezing of fruits and berries in a fast freezer supplied with cold from a cascade refrigerating machine is performed at lower temperatures, which considerably accelerate low-temperature treatment. This increases the productivity of a fast freezer and improves the qualitative characteristics of frozen fruits.

A cascade refrigerating machine is more expensive than a two-stage refrigerating machine. Its installation and maintenance are also more expensive and require a higher qualification of maintenance personnel. However, an increase in the productivity of a fast freezer with a cascade refrigerating machine due to a decrease in the time of low-temperature treatment and much lower energy expenditures on the driver of a cascade refrigerating machine makes it more economically attractive for the fast freezing of fruits and berries.

REFERENCES

1. *Dietary Reference Intakes. Applications in Dietary Assessment* (National Academy Press, Washington, DC, 2000).
2. *Microbial Hazard Identification in Fresh Fruit and Vegetables,* ed. by James, J. (John Wiley & Sons, Dublin, 2006).
3. *Handbook of Food Preservation,* ed. by Rahman, M.S. (CRC Press, Boca Raton, FL, 2007).
4. Almási, E., Erdélyi, L., and Sáray, T., *Élelmiszerek gyorsfagyasztása* (Mezőgazdasági Kiadó, Budapest, 1977).
5. Rasshchepkin, A.N., Korotkiy, I.A., and Korotkaya, E.V., Vliyanie rezhimov nizkotemperaturnoi obrabotki na kachestvennye pokazateli yagod chernoi smorodiny (Effect of low-temperature treatment regimes on the qualitative characteristics of blackcurrant berries), *Tekhnika i tekhnologiya pishchevykh proizvodstv* (Food Processing: Techniques and Technologies), 2014. №1. P. 101–105.
6. Golyand, M.M., Malevannyi, B.N., Pechatnikov, M.Z., and Plotnikov, V.T., *Sbornik primerov raschetov i laboratornykh rabot po kursu "Kholodil'noe tekhnologicheskoe oborudovanie"* (Collection of Calculation Examples and Laboratory Works on the Course in Refrigeration Process Equipment) (Legkaya i pishchevaya promyshlennost', Moscow, 1981).
7. Baranenko, A.V., Kutsakova, V.E., Borzenko, E.I., and Frolov, S.V., *Primery i zadachi po kholodil'noi tekhnologii pishchevykh produktov. Ch. 3. Teplofizicheskie osnovy* (Examples and Problems on Food Refrigeration Technology, Part 3. Thermophysical Principles) (KolosS, Moscow, 2004).
8. Lykov, A.V., *Teoriya teploprovodnosti* (Theory of Heat Conductivity) (Vysshaya shkola, Moscow, 1967).
9. Rogov, I.A., Kutsakova, V.E., Filippov, V.I., and Frolov, S.V., *Konservirovanie pishchevykh produktov kholodom* (Refrigeration Preservation of Food Products) (KolosS, Moscow, 1999).
10. Korotkiy, I.A., *Mashiny nizkotemperaturnoi tekhniki. Chast' I* (Low-Temperature Engineering Machines. Part I) (KemTIPP, Kemerovo, 2004).
11. Korotkiy, I.A., *Mashiny nizkotemperaturnoi tekhniki. Chast' II* (Low-Temperature Engineering Machines. Part II) (KemTIPP, Kemerovo, 2008).