Simulation of operating modes of the refrigerating machine and the heat pump when drying fish

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Abstract: Saving energy resources in the fishing industry is an important task. The purpose of the study is to determine rational schemes and operating modes of air conditioning systems for drying fish. The methodology for calculating the air conditioning system for drying fish and on its basis a computer program in the mathematical package MathCAD is developed. Two air conditioning schemes are considered: a scheme using an air heater and a chiller and a scheme using a chiller and a heat pump. The schemes were analyzed from the standpoint of energy indicators. The parameters of the air supplied to the drying chamber were taken for calculations on the basis of the need to maintain the parameters set by the fish processing technology. In this paper, air conditioning schemes for drying fish are considered, an analysis of the main energy costs during the operation of equipment is carried out. On the basis of calculations, the advantages and disadvantages of the systems, their energy costs for maintaining the specified technological modes are shown. The influence of temperature and flow rate of cold outdoor air and the use of heat recovery units for the exhaust air are shown. It was established that when using outside air with heat exchangers, it is possible to significantly reduce the consumption of heat energy for the technological mode in the cold season. In the absence of heat sources, the scheme with a refrigeration machine and a heat pump has an advantage. The research results were introduced at the fish enterprise of the Astrakhan region by the LLC “Technology of comfort”.

1. Introduction

Saving energy resources in the fishing industry is an important task. The development of methods for calculating and analyzing the operating modes will allow obtaining the air parameters necessary to maintain the technological modes of fish drying during the design and operation of equipment and achieving high product quality. When analyzing the operation of industrial and semi-industrial air conditioning systems, it is necessary to take into account the energy costs of its operation and the requirements of the technological process. One of the main issues of air conditioning systems is the reduction of energy costs for their operation.

The works [1-6] are devoted to the issues of drying fish and fish products, which show the importance of obtaining characteristics of fish drying for the initial design data of air conditioning systems. The works present experimental [4, 5] and calculation methods for constructing the curves of kinetics and dynamics for the dehydration of fish fillets [1, 3]. The data obtained in the works on drying kinetics are obtained for individual species and condition of fish and are not always sufficient for designing air conditioning systems for drying fish products.
There are known publications [7-17], which describe air conditioning systems using outside air, refrigeration machines, heat pumps, and internal air heat exchangers in the production of dried fish. However, calculation methods and analysis of the influence of various external and internal air parameters on the operating modes necessary for design are not given.

Designing air conditioning systems for production is a complex and important task, including the synthesis of engineering solutions and modern design methods. The analysis of schemes and modes of air conditioning systems in the production of dried fish at the design stage will increase its energy efficiency and improve environmental and energy safety, as well as product availability for consumers.

The purpose of the study is to determine energy-efficient schemes and operating modes of air conditioning systems for fish drying. The task of the work is to analyze the schemes and operating modes of air conditioning systems for drying fish and to develop the method for their calculation and analysis.

Currently, air conditioning schemes may be classified using electrical and thermal energy or using electrical energy only. In this case, external cold air, heat utilizers, refrigerators and heat pumps can be used [7-17].

Depending on air parameters at the entrance to the fish drying chamber, the operation modes of the air conditioning system during drying may be divided into the following:
- modes with constant air parameters during the whole drying process;
- modes with air parameters varying during drying (consecutive or continuous) [8].

Variable modes allow combining heating of the material with its humidity and properties better than constant modes.

To obtain high-quality products after fish drying, it is necessary to observe drying modes, which shall take into account the specific features of individual types of products [1-6].

2. Materials and methods
Let us consider two air conditioning diagrams.
1 – a scheme using a refrigerating machine and an air heater heated by hot water or steam, with the supply of external air and a heat utilizer for the removed air (Fig. 1);
2 – a scheme using a refrigerating machine and a heat pump with the possibility of supplying external cold air with a heat utilizer for the removed air and using the heat of the refrigeration condenser (Fig. 2).

Figure 1. Schemes of air conditioning systems for fish drying: a – a diagram with a refrigerating machine and an air heater, b – a diagram with a refrigerating machine and a heat pump: 1 – fish drying chamber, 2 – air cooler of the refrigerating machine, 3 – exhaust fan, 4 – supply fan, 5 – heat exchanger, 6 – circulation fan, 7 – air heater, 8 – heat pump air cooler, 9 – condensation heat exchanger of a refrigerating machine, 10 – heat pump condenser.
The first scheme is used with the available source of electricity and thermal energy, while the second – in the absence of thermal energy.

For this type of raw material and its processing, the equation of the consumption of moisture to be removed during drying is written as (1)

\[ G_m = f(G_{fish}, G_1, t_1, \varphi_1, \tau_{dr}) \]  

where \( G_m \) – amount of evaporating moisture in the drying chamber, kg/s; \( G_{fish} \) – amount of fish in the chamber, kg; \( G_1 \) – air flow rate at the inlet to the drying chamber, kg/s; \( t_1 \) – air temperature at the inlet to the drying chamber, °C; \( \varphi_1 \) – relative air humidity at the inlet to the drying chamber, %; \( \tau_{dr} \) – current time of fish drying, s.

In the general case of an air conditioning system, the dependence of the removed moisture flow rate of fish drying depends on many parameters (2)

\[ G_m = f(V_p, t_1, \varphi_1, \tau_{dr}, G_{fish}, Q_{e_r}, N_{fr}, Q_{e_u}, Q_{e_h}, N_{hp}, G_{out}, t_{out_in}, \varphi_{out_in}, E, N_{sup}, N_{ex}, N_{cir}) \]

where \( Q_{e_r} \) – heat load on the refrigerator evaporator, kW; \( N_{fr} \) – power consumed by the electric motor of the refrigerating machine, kW; \( Q_{e_u} \) – thermal power of refrigerating machine condenser-utilizer, kW; \( Q_{e_h} \) – heat load on heat pump evaporator, kW; \( N_{hp} \) – power consumed by heat pump motor, kW; \( G_{out} \) – outside air flow rate, kg/s; \( t_{out_in} \) – outside air temperature at fan inlet, °C; \( \varphi_{out_in} \) – relative humidity of outside air at fan inlet, %; \( E \) – efficiency factor of the heat exchanger; \( N_{sup} \) – power consumed by the supply fan, kW; \( N_{ex} \) – power consumed by exhaust fan, kW; \( N_{cir} \) – power consumed by circulation fan, kW.

By setting the temperature, relative humidity of the air, the flow rate of the supplied air to the drying chamber and the current drying time, we solve the system of equations (3-7) and determine the parameters of the moist air, the air flow rate at the outlet of the drying chamber, the moisture flow rate and thermal loads. The total heat load during drying is zero.

\[ G_m = f(G_{fish}, G_1, t_1, \varphi_1, \tau_{dr}) \]  

\[ Q_{dr_tot} = G_2 \cdot t_2 (t_2, \varphi_2) - G_1 \cdot t_1 (t_1, \varphi_1) \]  

\[ Q_{dr_exp} = C_{1-2} \cdot t_2 (t_2, \varphi_2) \cdot (t_1 - t_2) \]  

\[ Q_{dr_hid} = Q_{dr_tot} - Q_{dr_exp} \]

\[ G_m = (G_2 \cdot d_2 (t_2, \varphi_2) - G_1 \cdot d_1 (t_1, \varphi_1)) \cdot 3.6 \]

where \( G_m \) – amount of evaporated moisture in the drying chamber depending on the drying time, kg/s; \( G_{fish} \) – amount of fish in the chamber, kg; \( G_1, \ G_2 \) – air flow in the inlet and outlet of the drying chamber, kg/s; \( t_1, \ t_2 \) – air temperature at the inlet and outlet of the drying chamber, °C; \( \varphi_1, \ \varphi_2 \) – relative air humidity at the inlet and outlet of the drying chamber, %; \( Q_{dr_tot}, \ Q_{dr_exp}, \ Q_{dr_hid} \) – total, explicit and hidden heat load for air in the drying chamber, kW; \( C_{1-2} \) – average specific heat capacity of moist air during drying, kJ/(kg*K).

By setting the temperature, relative air humidity, ambient air flow rate, efficiency of the heat exchanger, power of fans, we solve the system of equations (8-13) and determine the parameters of the supply air at the outlet of the heat exchanger and the thermal load. The power consumed by the fan is determined by the model obtained from the fan characteristics depending on the network resistance and air flow rate, taking into account the efficiency of the electric motor, drive and efficiency of the fan. The efficiency factor of the heat exchanger \( E \) is accepted based on the manufacturer’s catalogs.
\[
d(t_{\text{out}, \text{out}}, \varphi_{\text{out}, \text{out}}) = d(t_{\text{out}, \text{in}}, \varphi_{\text{out}, \text{in}})
\]

(8)

\[
N_{\text{sup}} = G_{\text{out}} * (I(t_{\text{out}, \text{out}}, \varphi_{\text{out}, \text{out}}) - I(t_{\text{out}, \text{in}}, \varphi_{\text{out}, \text{in}}))
\]

(9)

\[
N_{\text{ex}} = G_{\text{out}} * (I(t_{\text{r}, \text{out}}, \varphi_{\text{r}, \text{out}}) - I(t_{\text{r}, \text{in}}, \varphi_{\text{r}, \text{in}}))
\]

(10)

\[
t_{\text{h}, \text{out}} = E * (t_{\text{r}, \text{out}} - t_{\text{h}, \text{in}}) + t_{\text{r}, \text{in}}
\]

(11)

\[
d(t_{\text{h}, \text{out}}, \varphi_{\text{h}, \text{out}}) = d(t_{\text{h}, \text{in}}, \varphi_{\text{h}, \text{in}})
\]

(12)

\[
Q_{\text{ther}} = G_{\text{out}} * (I(t_{\text{h}, \text{out}}, \varphi_{\text{h}, \text{out}}) - I(t_{\text{h}, \text{in}}, \varphi_{\text{h}, \text{in}}))
\]

(13)

where \(t_{\text{out}, \text{in}}, t_{\text{out}, \text{out}}\) – outside air temperature at fan inlet and outlet, °C; \(t_{\text{h}, \text{in}}, t_{\text{h}, \text{out}}\) – outside air temperature at the inlet and outlet of the heat exchanger, °C; \(t_{\text{r}, \text{in}}, t_{\text{r}, \text{out}}\) – temperature of removed air at the inlet and outlet of the exhaust fan, °C; \(\varphi_{\text{out}, \text{in}}, \varphi_{\text{out}, \text{out}}\) – relative humidity of the outside air at the inlet and outlet of the fan, %; \(\varphi_{\text{h}, \text{in}}, \varphi_{\text{h}, \text{out}}\) – relative humidity of the supply air at the inlet and outlet of the heat exchanger, %; \(G_{\text{out}}\) – outside air flow rate, kg/s; \(Q_{\text{ther}}\) – thermal load on heat recovery unit, kW; \(N_{\text{sup}}\) – power consumed by the supply fan, kW; \(N_{\text{ex}}\) – power consumed by the exhaust fan, kW.

By setting the temperature, relative air humidity at the inlet to the evaporator of the refrigerating machine and adjusting the capacity of the refrigerating machine, and solving the system of equations (14-20), we determine the air parameters at the outlet of the evaporator, the amount of moisture and thermal loads

\[
Q_{\text{e}, \text{rm}} = Q_{\text{eo}, \text{rm}} * n_{\text{rm}}
\]

(14)

\[
Q_{\text{e}, \text{rm}} = G_{\text{e}, \text{rm}} * (I(t_{\text{e}, \text{in}}, \varphi_{\text{e}, \text{in}}) - I(t_{\text{e}, \text{rm}, \text{out}}, \varphi_{\text{e}, \text{rm}, \text{out}}))
\]

(15)

\[
G_{\text{m}, \text{rm}} = G_{\text{e}, \text{rm}} * (d(t_{\text{e}, \text{in}}, \varphi_{\text{e}, \text{in}}) - d(t_{\text{e}, \text{rm}, \text{out}}, \varphi_{\text{e}, \text{out}})) * 3.6
\]

(16)

\[
N_{\text{rm}} = N_{\text{e}, \text{rm}} * n_{\text{rm}}
\]

(17)

\[
Q_{\text{c}, \text{rm}} = Q_{\text{e}, \text{rm}} + N_{\text{rm}}
\]

(18)

\[
Q_{\text{c-u}, \text{rm}} = Q_{\text{c}, \text{rm}} * n_{\text{c-u}}
\]

(19)

where \(t_{\text{e}, \text{in}}, t_{\text{e}, \text{out}}\) – air temperature at the inlet and outlet of the evaporator of the refrigerating machine, °C; \(t_{\text{e}, \text{rm}, \text{w}}\) – evaporator wall temperature; \(\varphi_{\text{e}, \text{in}}, \varphi_{\text{e}, \text{out}}\) – relative air humidity at the inlet and outlet of the refrigerator evaporator, %; \(G_{\text{e}, \text{rm}}\) – air flow through the evaporator, kg/s; \(G_{\text{m}, \text{rm}}\) – amount of moisture removed, kg/h; \(Q_{\text{e}, \text{rm}}\) – heat load on the refrigerator evaporator at 100% load, kW; \(Q_{\text{e}, \text{rm}}\) – thermal load on the refrigerator evaporator at partial load \(n_{\text{rm}}\), kW; \(N_{\text{e}, \text{rm}}\) – power consumed by the refrigerator electric motor at 100% loading, kW; \(N_{\text{rm}}\) – power consumed by the refrigerator motor at partial load \(n_{\text{rm}}\), kW; \(Q_{\text{c-u}, \text{rm}}\) – heat power of the condenser-utilizing unit, kW.

The cooling capacity and capacity of the refrigerator \(Q_{\text{eo}, \text{rm}}\) and \(N_{\text{eo}, \text{rm}}\) are obtained from the catalog of the selected refrigerator based on the calculation depending on external parameters.

By setting the temperature, relative air humidity at the inlet to the evaporator of the heat pump, and solving the system of equations (21-26), we determine the air parameters at the outlet of the evaporator of the heat pump, the amount of moisture, cooling capacity, thermal power and air heating in the condenser of the heat pump

\[
Q_{\text{e}, \text{hp}} = Q_{\text{eo}, \text{hp}} * n_{\text{hp}}
\]

(20)
\[ Q_{e\text{-hp}} = G_{e\text{-hp}} \times \left( I(t_{e\text{-hp\_in}}, \varphi_{e\text{-hp\_in}}) - I(t_{e\text{-hp\_out}}, \varphi_{e\text{-hp\_out}}) \right) \]  
\[ \frac{l(t_{e\text{-hp\_in}, \varphi_{e\text{-hp\_in}}}) - l(t_{e\text{-hp\_out}, \varphi_{e\text{-hp\_out}}})}{l(t_{e\text{-hp\_in}, \varphi_{e\text{-hp\_in}}}) - l(t_{e\text{-hp\_w}, 100\%})} = \frac{d(t_{e\text{-hp\_in}, \varphi_{e\text{-hp\_in}}}) - d(t_{e\text{-hp\_out}, \varphi_{e\text{-hp\_out}}})}{d(t_{e\text{-hp\_in}, \varphi_{e\text{-hp\_in}}}) - d(t_{e\text{-hp\_w}, 100\%})} \]  
\[ G_{m\text{-hp}} = G_{e\text{-hp}} \times \left( d(t_{e\text{-hp\_in}, \varphi_{e\text{-hp\_in}}}) - d(t_{e\text{-hp\_out}, \varphi_{e\text{-hp\_out}}}) \right) \times 3.6 \]  
\[ N_{hp} = N_{o\text{-hp}} \times n_{hp} \]  
\[ Q_{c\text{-hp}} = Q_{e\text{-hp}} + N_{hp} \]

where \( t_{e\text{-hp\_in}}, t_{e\text{-hp\_out}} \) – air temperature at the inlet and outlet of the heat pump evaporator, °C; \( t_{e\text{-hp\_w}} \) – temperature of the heat pump evaporator wall; \( \varphi_{e\text{-hp\_in}}, \varphi_{e\text{-hp\_out}} \) – relative air humidity at the inlet and outlet of the heat pump evaporator,%; \( G_{e\text{-hp}} \) – air flow rate through heat pump evaporator, kg/s; \( G_{m\text{-hp}} \) – amount of removed moisture, kg/h; \( Q_{e\text{-hp}} \) – heat load on the heat pump evaporator at 100% load, kW; \( Q_{c\text{-hp}} \) – heat load on the heat pump evaporator at partial load \( n_{hp} \), kW; \( N_{hp} \) – power consumed by the heat pump motor at 100% load, kW; \( N_{o\text{-hp}} \) – power consumed by the heat pump motor at partial load \( n_{hp} \), kW.

The cooling capacity and the capacity of the heat pump \( Q_{e\text{-hp}} \) and \( N_{hp} \) are obtained from the catalog of the selected heat pump and calculation depending on external parameters.

Next, we define the point parameters in the air conditioning system diagrams taking into account the air heating in the circulation fan depending on \( n_{hp} \).

A program in the mathematical package MathCAD was developed for calculation. The calculation of operation modes of air conditioning systems may be carried out with different equipment composition. The program allows quickly correcting the calculation methodology and the composition of the equipment included in the unit diagrams.

The program uses models of moist air properties, which allow determining enthalpy, moisture content and calculating the processes of moist air treatment. For calculation, the models of the refrigerating machine and the heat pump are given depending on the boiling point and condensation, and for the model of fans their power is given depending on the air flow rate and network resistance.

The program takes into account the technological processes in the drying chamber. The consumption of removed moisture in the drying chamber is calculated by the kinetic curve of fish drying. The program takes into account the heat generated by the fans. The program allows calculating the heat utilization unit of the removed air by the coefficient of temperature efficiency of the plate recuperator. The program calculates the operating parameters of the refrigerating machine and the heat pump using the method of solving the systems of equations of thermal and humidity balance taking into account the correction when the parameters get into the moist area of the moist air diagram. The condensation heat utilization unit of the refrigerating agent of the refrigerating machine is calculated according to the specified air temperature at the outlet of the heat utilization unit. The program checks mass and heat balances by convergence of the specified parameters at the inlet to the drying chamber.

The program allows conducting calculations under various operating modes, determining the parameters of air flows, thermal and humidity loads and analyzing the effect of various parameters on energy costs. The program makes it possible to conduct a multifactorial experiment and determine the operating modes for a given area, the time of the year and the operating conditions of the fish drying facility.

Let us consider the results of the calculation according to the developed program for two schemes of the air conditioning system and two modes 1) at a constant drying speed and 2) with a varying drying speed.
3. Results and Discussion
Design data: Calculations were carried out at a given air temperature of 25 °C and a relative humidity of 55% at the input to the drying chamber, the air velocity in the chamber is 1.5 m/s. For mode 1, the drying rate \( G_m = 30 \) kg/h, and for mode 2, the drying rate varied from 75 kg/h to 5 kg/h. The amount of fish loaded was 1000 kg. The air flow to the drying chamber was supplied in the amount of 40,000 m\(^3\)/h, and the efficiency of the recuperator is accepted as 0.6. The refrigerating machine and the heat pump with spiral compressors operate on R410A refrigerating agent.

**Calculation according to mode 1.** The analysis of scheme 1 shows that without the supply of external air, the air conditioning system operates at a thermal energy consumption of 29.8 kW and electric energy of 12.0 kW.

Figure 2 shows the results of calculation of scheme 1 with external air supply and non-operating refrigerating machine with heat utilization of removed air and without utilization.

![Figure 2](image)

**Figure 2.** Results of the calculation of scheme 1 with the supply of external air and an air heater. a – external air consumption, b – electric power, c – thermal power depending on the external air temperature when external air is supplied
In the process of removing air, the moisture in this flow is also removed, the flow rate of which is determined as the difference in the moisture content of the warm removed air and cold supply air multiplied by the air flow rate.

In the cold period of the year, air supply leads to a reduction in electricity consumption by 44.2÷58.5%, while thermal energy consumption decreases by 22% (at a temperature of 15 °C), and starting from a temperature of +5 °C it increases by 0÷42.2%. The air flow rate for maintaining the drying mode and parameters at the inlet to the chamber depending on the external air temperature ranges from 1875 m³/h to 3550 m³/h. When using a utilization system with a recuperator efficiency of 60%, the electric power consumption when supplying cold air does not change, and the thermal energy consumption decreases with an increase in external air temperature by 13.5÷42.4%.

The analysis of scheme 2 shows that without the supply of external air, the air conditioning system operates at an electric energy consumption of 12.92 kW, and the thermal energy of the air heater is not used (Fig. 3):
In the cold period of the year, the air supply for scheme 2 leads to a reduction in electricity consumption, for example, at an external air temperature of 10 °C by 42.9%, and the refrigerating machine is disconnected at a flow rate of 2000 m³/h. External air flow rate to maintain the drying mode and parameters at chamber inlet is 3000 m³/h at external air temperature $t_{ex}=0$ °C, 2000 m³/h at $t_{ex}=-10$ °C, 1000 m³/h at $t_{ex}=-15$ °C.

**Calculation according to mode 2.** The analysis of operating modes at a variable drying speed in accordance with the drying kinetics of a given type of fish was carried out. The dependence of relative moisture in fish on the drying time is shown in Fig. 4. At the same time, the drying speed in the first period is high, and then significantly decreases.

![Figure 4. Dependence of the removed water on the drying time of the fish (whiting fillet) [4]](image)

We accept the data based on the calculation in the MathCAD program and the actual data of the enterprise for fish drying. The amount of moisture in the fish is 80.66%, then the dry matter content in the fish is $\xi=19.34\%$. With the amount of the fish at the beginning of drying $G_{fish}=1000$ kg, the amount of dry matter is determined by equation (27):

$$G_{dr} = \xi \times G_{fish} \times 0.01 = 193.4 \text{ kg}$$

(27)

We take fish drying time for 20 hours (which corresponds to real technology). The relative amount of moisture in the fish is calculated by equation (28):

$$W = \frac{G_m}{G_{dr}} \times 100$$

(28)

The mathematical model of fish drying kinetics, dependence of relative amount of moisture on the dry part of fish in percentage in the range of drying time $\tau$ from 0 to 25 h has the form of the regression dependence (29) (graphical dependence given in paper [4] is used):

$$W = -0.0291 \times \tau^3 + 1.7296 \times \tau^2 - 36.831 \times \tau + 417.05$$

(29)

The validity of the approximation $R^2=0.9985$.

The amount of moisture in fish depending on the drying time (30), kg

$$G_m = W \times G_{dr} \times 0.01$$

(30)

The relative amount of moisture in the fish at the beginning of the process is 417.1% or 806.6 kg, and at the end of the process 139.5% or 269.7 kg. The amount of moisture removed in 20 hours was 536.8 kg. The average amount of moisture removed was 26.84 kg/h.

Figure 5 shows the results of the calculation of scheme 1 at a variable drying speed and a disconnected refrigerating machine when cold air is supplied without heat utilization and with heat utilization.

Heat utilization of the removed air reduces the heat load of the air heater by 20÷36.8%.
The dependencies shown in Figure 5 allow calculating the external air flow rate from the drying time at the specified external air temperature and control the drying process depending on the time of the external air flow rate and heating in the air heater by the air temperature when supplied to the drying chamber.

**Figure 5.** Results of the calculation of scheme 1 when the outside air is supplied when the refrigeration machine is not working and the air heater is operating at a variable drying speed: a – the amount of moisture removed, b – the outdoor air consumption, c – the consumed electric power, d – thermal power of the air heater e – heat output of an air heater with a heat recovery unit from the time of drying and the temperature of the outside air.
Figure 6 shows the results of calculation of scheme 1 and 2 without external air supply in the warm period of the year.

![Graphs showing the results of calculation of scheme 1 and 2](image)

Figure 6. Results of the calculation of scheme 1 with a refrigerating machine and an air heater at a variable drying speed: a) amount of moisture removed, b) electrical power consumed, c) cooling power, d) thermal power of the air heater on the drying time

The dependencies obtained in Figure 6 provide data on the consumption of thermal power and electricity. Energy consumption at the beginning of the process is significant and exceeds energy consumption by 4 times, and thermal energy – by 20 times. Therefore, when designing, it is necessary to give preference to economical methods for regulating the performance of thermal and especially refrigeration equipment, where inverter systems can be used.

As a result of calculating scheme 2 with a refrigerating machine and a heat pump at a variable drying speed, it is obtained that the consumed electric power on the drying time remains the same as in scheme 1. The combined refrigerating power of the refrigerating machine and the heat pump on the drying time corresponds to the refrigerating power of the refrigerating machine of scheme 1. Heat capacity of the heat pump corresponds to the heat capacity of the air heater. Thus, only electrical energy is consumed in scheme 2, so this system may be recommended especially in the absence of thermal energy in the area of production of dried fish.
The calculation results for scheme 2 were used during the design and implementation by Comfort Technology LLC at the Astrahan fish processing plant. In this case, three split systems of Daikin company with an external block RZQ250C7Y1B and an internal block FDQ250B8V3B were used [18, 19]. Two split systems and an internal unit of the refrigerating machine are installed inside the room, and a condenser of the refrigerating machine is installed outside the building. Air circulation is performed by an axial fan VO-11.2-5.5/1500 of series 1 [20], air supply and removal – by centrifugal fans VC-4-70-6.3/1500 of series 1 [21].

4. Conclusion
The calculations of air conditioning systems for fish drying for various schemes and operating modes, as well as the analysis of the results, show that the external air flow in the cold season shall be carried out depending on the temperature of the external air and the amount of moisture removed when the refrigerating machine is not working. The supply of external air for the scheme with the refrigerating machine and air heater reduces the consumption of electric energy to 60%, and the use of the heat exchanger of the removed air reduces the consumption of thermal energy to 37%. When using a scheme with a refrigerating machine and a heat pump, the supply of external air and the disposal of the heat of the removed air reduce electricity consumption to 43%, while there is no thermal energy consumption. The calculated drying in accordance with the drying kinetics requires more powerful equipment with regulation of its productivity.

The developed methodology allows calculating air parameters, thermal and humidity loads in the air conditioning system during drying and analyzing its operating modes. The program developed using this methodology allows designing air conditioning systems for fish drying with less energy costs and reducing the cost of the product for the end user.

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