Mathematical modeling of the refrigerating chamber of a combined solar dryer-refrigerator with a heat pump

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Abstract. Mathematical modeling appears to be a valuable tool for controlling the cooling and storage process of agricultural products. Therefore, mathematical modeling of the balance of heat and mass in refrigeration chambers is relevant. Mathematical modeling of the thermal regime of the thermal regime of a refrigeration chamber with a heat pump and determining temperature changes depending on time. Mathematical modeling is carried out based on the thermal balance of the refrigerating chamber by the method of conservation of mass and energy. It has been established that cooling and long-term storage of agricultural products in refrigeration chambers require a huge amount of electrical energy. At an outside air temperature of 30℃, it takes 30-40 minutes to cool products to the required temperature; for long-term storage, 5.5-6.0 kWh/day of electrical energy is consumed. A combined dryer-refrigerator with a heat pump for storing agricultural products is proposed and a mathematical model of the refrigeration chamber is given, which makes it possible to analyze the temperature regime in the refrigeration chamber.

1 Introduction

Currently, for high-quality and safe storage of agricultural products in the refrigeration chamber, low (usually from -1 to 5 ℃) or negative (below -15 ℃) temperatures are used. To create optimal temperatures, vapor compression refrigeration units are used. According to the International Institute of Refrigeration, energy consumption in refrigeration processes accounts for 15% of global energy consumption [1]. In 2012, there were 1.6 million refrigeration units in Europe, 67% of which were small refrigeration units with a volume of up to 400 m^3 [2]. It is known that refrigerators consume large amounts of energy. According to Duiven and Binar [3], the relative energy consumption in refrigeration chambers is 30÷50 kW/m^3 per year. The lowest value in this study corresponds to the Netherlands. According to research by Bosma [4], the average energy consumption of refrigeration chambers is 35 kW/m^3 per year. According to Singh's research [5], in the

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USA, relative energy consumption ranged from 19 to 88 kW/m^{3}year and from 15 to 132 kW/m^{3}year.

An in-depth study conducted by Werner, Wein, Merz, and Cleland in New Zealand [6] compared the performance of 34 cold storage rooms. According to the results, the relative energy consumption ranged from 26 to 379 kW/m^3 per year, and savings of 15 to 26% were achieved through the use of energy-efficient devices.

To solve issues of energy saving in refrigeration chambers with heat pumps and select optimal technical solutions, a mathematical modeling method is used using modern computer calculation programs. A large number of studies have proposed various models for increasing the efficiency of refrigeration chambers and solving problems of energy saving in the refrigeration chamber. Most of them are related to the study of refrigeration chambers in various modes, including the Getu&Bansal LT-case model, EKS, RETScreen, EnergyPlus, and Cybermart programs. Special programs CoolPack and PackCalculation II are also available for cold rooms and cycles.

However, many researchers focus on increasing the efficiency of refrigeration chambers and improving the refrigeration cycle. Refrigerator efficiency can be improved by improving operating conditions, reducing refrigeration losses, and reusing heat rejected from the refrigeration compartment. The purpose of this work is to mathematically model the thermal regime of a refrigeration chamber with a heat pump and determine temperature changes depending on time [9-21].

2 Materials and methods

Taking into account the above, an underground refrigeration chamber with a vapor-compression heat pump was developed, and a prototype was manufactured and tested. The main thermal parameters of the refrigeration chamber are presented in Table 1. For the numerical study of the cooling chamber, energy balance equations were used; the calculation diagram for this balance is shown in Figure 1.

Table 1. Basic thermal parameters of the refrigeration chamber.

I. Specifications				
No.	**Parameters**	**Unit measurements**	**Designation**	**Quantity**
1	Cooling capacity of the refrigerator compartment	kW	N	1.1
2	Refrigerantquantity (freon-R22)	kg	m	4.2
3	Refrigeratorvolume	m^3	V_{cool}	5
4	Refrigerator capacity (by product type)	t	G_{cool}	1.5-2.0
II. Temperature regime				
1	Inside the refrigerator compartment	℃	$t_{c.ch.}$	-5...+10
III. Humidityconditions				
1	Inside the refrigerator compartment		φ	90…95

All the indicated values were taken into account in the heat balance of the refrigerating chamber (Figure 1). A mathematical model of the heat balance of the refrigeration chamber was developed based on the following assumptions:

- One-dimensional heat flow, perpendicular to the airflow, flows between the components of the cooling chamber.
- The temperature of the earth is equal to the ambient temperature.
- The flow in each part of the chamber is directed in one direction.
- Distribution of products on the rack inside the chamber is uniform.

- The temperature and humidity of the product and air remain unchanged within a given temperature range.
- The volume of the product remains unchanged during the storage period.
- There is no convective heat exchange between the outer wall of the cooling chamber and the soil mass.
- Take the temperature of the internal air of the refrigeration chamber equal to 0°C.

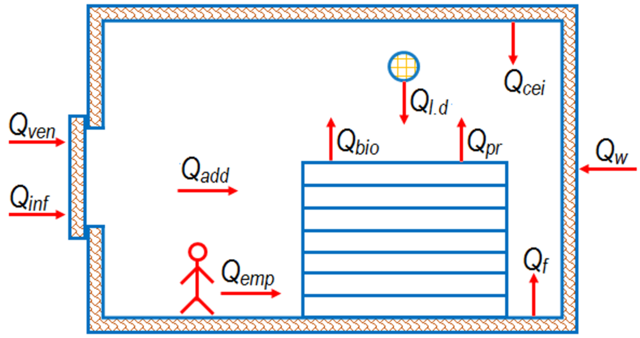

Fig. 1. Scheme for calculating the heat balance of the refrigeration chamber.

Q_{inf}-thermal load transferred to infiltration account; Q_{ven}-thermal load arising due to ventilation;Q_{emp}- heat load brought by the employee;Q_f- thermal load transmitted through the floor;Q_{cei}- thermal load transmitted through the ceiling;Q_w- thermal load transmitted through the outer wall;$Q_{l.d}$- thermal load transmitted through the lighting device;Q_{pr} and Q_{bio}- thermal load introduced with the original product and released as a result of a biochemical reaction;Q_{add}- the thermal load is separated from additional devices

The cooling capacity of a refrigerator is the heat load that must be removed from the interior of the refrigerator. The thermal load of the refrigeration chamber is divided into two groups: external and internal.

Externalloadsinclude:

- Loads Q_w, $Q_f$andQ_{cei} caused by heat transfer through the wall, floor and ceiling parts of the refrigeration chamber.
- Load introduced into the cooling chamber by the flow of outside air (ventilation), Q_{ven}.
- Load included due to the infiltration of outside air when opening and closing the door of the refrigeration chamber, Q_{inf}.

Internal loads include:

- Load from product cooling, Q_{pr}.
- Load caused by the "breathing" of the stored product, Q_{bio}.
- Load transmitted from the lighting device,$Q_{l.d.}$.
- Heat load separated from the working load, Q_w.
- Heat load separated from additional devices inside the refrigeration chamber, Q_{add}.

In general, the balance equation has the following form:

$$Q_f = \sum_{i=1}^{n} Q_i \tag{1}$$

According to the calculation scheme presented in Fig. 1, equation (1) has the following form:

$$Q_f = Q_w + Q_f + Q_{cei} + Q_{ven} + Q_{inf} + Q_{pr} + +Q_{bio} + Q_{l.d.} + Q_w + Q_{add} \tag{2}$$

Let’s look separately at each of the components of the energy balance equation:
Heat flow during heat transfer through the outer wall:

$$Q_w = \frac{F_w}{R_w}(t_{soil} - t_{in.air.}) \tag{3}$$

Where $F_w = 11.4\, m^2$ - total surface area of the refrigeration chamber wall; R_w -thermal resistance of the refrigeration chamber design:

$$R_w = \frac{1}{\alpha_{in}} + \frac{\delta_{pl}}{\lambda_{pl}} + \frac{\delta_{br}}{\lambda_{br}} + \frac{\delta_{cs}}{\lambda_{cs}} + \frac{\delta_{pl}}{\lambda_{pl}} + \frac{\delta_{m.sh.}}{\lambda_{m.sh.}} + \frac{1}{\alpha_н} = 2{,}66 \tag{4}$$

Where α_{in} is the heat transfer coefficient from the inner surface of the wall, with natural air circulation inside the chamber $\alpha_{in} = 9.4\, W/(m^2 \cdot°C)$, with forced circulation $\alpha_{in} = 22.7 \div 34.1\, Vt/(m^2 \cdot°C)$, where air speed $w = 3.3 \div 6.7\, m/s$; δ_{pl} =50+50=100 mm, δ_{br}=18 mm, δ_{cs}=15 mm, $\delta_{m.sh.}$=1 mm - thickness of layers of foam plastic, brick, cement-sand plaster and metal sheet; λ_{pl}=0.04 W/(m·°C), λ_{br}=0.46 W/(m·°C), λ_{cs}=1.2 W/(m·°C), $\lambda_{m.sh.}$=64 W/(m·°C); α_{ex} - convective heat exchange between the outside air and the wall, where $1/\alpha_{dir}$=0, since the walls of the refrigeration chamber are located directly underground, there is no convective heat exchange; F_w is the surface area of the wall; t_{soil}=17°C - average soil temperature; $t_{in.a.}$=0°C– internal air temperature.
Heat flow during heat transfer through the floor:

$$Q_f = \frac{F_f}{R_f}(t_{soil} - t_{in.a.}) \tag{5}$$

Where F_f=2.65 m^2 - total floor area of the refrigerator compartment; R_f-thermal resistance of the design of the floor part of the refrigeration chamber:

$$R_f = \frac{1}{\alpha_{in}} + \frac{\delta_{cs}}{\lambda_{cs}} + \frac{\delta_{pl}}{\lambda_{pl}} + \frac{1}{\alpha_{out}} = 2.61 \tag{6}$$

Where δ_{cs}=0.01 mandδ_{pl}=0.1 m - thickness of cement-sand layer and foamplastic; λ_{cs}=1.75 W/(m·°C) andλ_{pl}=0.04 W/(m·°C) - heat transfer coefficients of the cement-sand layer and foam plastic.
Heat flow during heat transfer through the ceiling:

$$Q_{cei} = \frac{F_{cei}}{R_{cei}}(t_{d.cham.} - t_{in.a.}) \tag{7}$$

Where F_{cei}=2.65 m^2 - total surface area of the refrigerator compartment ceiling; R_{cei}-thermal resistance of the ceiling part of the refrigeration chamber:

$$R_{cei} = \frac{1}{\alpha_{in}} + \frac{\delta_{m.sh.}}{\lambda_{m.sh.}} + \frac{\delta_{pl}}{\lambda_{pl}} + \frac{\delta_{wl}}{\lambda_{wl}} + \frac{1}{\alpha_{out}} = 1.54 \tag{8}$$

Where$t_{d.cham.}$=30°C – average temperature of the drying chamber.
The heat flow introduced by ventilation occurs as a result of periodic changes of air inside the refrigerating chamber. The outside air entering the refrigeration compartment must be cooled to the internal temperature of the refrigeration compartment, which represents an additional heat load and is defined as:

$$Q_{ven} = G_{air} c_{p\,air} (t_{out.a.} - t_{in.a.}) \tag{9}$$

Where G_{air} -outside air flow into the cooling chamber, kg/s:

$$G_{air} = V_{out.a.} \rho_{out.a.} \tag{10}$$

$V_{out.a.}$- volumetric flow rate of outside air entering the cooling chamber, m^3/s; $\rho_{out.a.}$-air density inside the chamber:

$$\rho_{air} = \frac{\rho_0}{\left(1+\frac{t_{chamber}}{273{,}15}\right)} = 1.293\ kg/m^3 \tag{11}$$

ρ_0-air density at temperature 0°C, ρ_0=1.293 kg/m^3;$t_{chamber}$=0°C-air temperature inside the refrigerator compartment;$t_{out.a.}$=30°C – outside air temperature.

Heat load included in outdoor air infiltration calculation:

$$Q_{inf} = q D_{in.a.} D_{en} (1 - E) \tag{12}$$

n=5 - number of entrances and exits per day; $\tau_{1\,open}$ =30 sek- time of keeping the door open/closed for each entrance and exit; $\tau_{2\,open}$ =20 min - time of keeping the door open during the day; 86400 - number of seconds per day; $D_{in.a.}$ - coefficient characterizing the flow of air entering through the door, which depends on the difference in temperature of the external and internal air, i.e. up to 1.1 at a temperature difference of 7-10°C, 0.8 at a temperature difference of 16°C and above, in our case D_{en}=0.8; E is the level of effectiveness of the protective device when entering through the door. E=0, when there is no protective device, E=0.2, when it is made in the form of a curtain, E=0.7 in other designs, in our case E=0.

Thermal load generated when the product temperature drops to the storage temperature in the chamber:

$$Q_{pr} = \frac{m_{pr} c_1 (t_1 - t_2) + m_{pr} r + m_{pr} c_2 (t_2 - t_3)}{86400} \tag{13}$$

Where m_{pr}– daily product turnover, kg/day; c_1 - average heat capacity of each type of product in the temperature range from $t_1$tot_2, kDJ/kg; t_1=18÷20°C – initial temperature of the stored product; t_2-high freezing temperature of the stored product; r-latent heat during freezing of the stored product, kDJ/kg; c_2 - average heat capacity of each type of product in the temperature range from $t_2$tot_3kDJ/kg; t_3-product storage temperature.

If the refrigerating chamber is intended only for cooling without freezing the product, then expression (13) has the following form:

If the refrigerating chamber is intended only for cooling without freezing the product, then expression (13) has the following form:

$$Q_{pr} = \frac{m_{pr} c_1 (t_1 - t_3)}{86400} \tag{14}$$

Thermal load caused by the "breathing" of the stored product:

$$Q_{bio} = \frac{M_{pr} q_{bio}}{86400} \tag{15}$$

WhereM_{pr} – weight of stored product, kg; q_{bio} – released during the respiration of a biostored product, kDJ/(kg·day).

Thermal load transferred from a lighting fixture. The lighting device installed in product storage rooms must be resistant to cold, moisture, and dust, waterproof, and shockproof. The nominal illumination of refrigeration chambers is from 60 to 100 lux, and the thermal load is assumed to be 3÷6 W per square meter of surface. In general, the thermal load transferred from a lighting fixture is determined as follows:

$$Q_{l.d.} = \frac{n_{l.d.}P_{l.d.}\tau_{l.d.}}{24} \tag{16}$$

Thermal load separated from the worker:

$$Q_w = \frac{n_w q_w \tau_w}{24} \tag{17}$$

Where n_w=1is the number of employees working in the refrigerating chamber; q_w is the amount of heat released per unit time with average worker activity, when the temperature inside the refrigerating chamber is 0℃, q_w=270 W; τ_w=1 hour/day – time spent by one employee in the refrigeration chamber for one day.

Additional heat loads in the refrigeration compartment include heat loads separated from equipment for transporting, lifting, unloading, and loading products. In this case, the allocated thermal load is determined as follows:

$$Q_{add} = \frac{nP\tau}{24} \tag{18}$$

Where n is the number of transport devices; P-power of each mechanism; t is the operating time of each mechanism in one day, in our case Q_{add}=0.

Substituting into expression (1) expressions (2-18), which form the heat balance, we obtain (in our case, the additional heat load is not taken into account):

$$\begin{aligned}
\rho_{in.a.}V_{in.a}c_{in.a.}\frac{dt_{in.a.}}{d\tau} &= \frac{F_w}{R_w}\left(t_{out.a.} - t_{in.a.}(\tau)\right) + \frac{F_f}{R_f}\left(t_{out.a.} - t_{in.a.}(\tau)\right) \\
&+ \frac{F_{cei}}{R_{cei}}\left(t_{out.a.} - t_{in.a.}(\tau)\right) + \\
&+\rho_{in.a.}V_{in.a.}c_{p\ air}\left(t_{out.a.} - t_{in.a.}(\tau)\right) + 0.076c_{p\ air}\left(t_{out.a.} - t_{in.a.}(\tau)\right) \\
&+ \frac{m_{pr}c_1(t_1 - t_3)}{86400} + \\
&+ \frac{M_{pr}q_{bio}}{86400} + \frac{n_{add}P_{add}\tau_{add}}{24} + \frac{n_w q_w \tau_w}{24}
\end{aligned} \tag{19}$$

When solving the heat balance equation (19) using the Euler method, equation (20) was obtained based on the condition t=0, $t_{in.a.}(\tau) = t_{out.a.}$ and a block diagram was constructed in Matlab/Simulink software (Figure 2).

$$\frac{dt_{in.a.}}{d\tau} = -\frac{\frac{F_w}{R_w} + \frac{F_f}{R_f} + \frac{F_{cei}}{R_{cei}} + \rho_{in.a.}V_{out.a.}c_{p\ air} + 0{,}076c_{p\ air}}{\rho_{in.a.}V_{in.a.}c_{in.a.}} t_{in.a.}(\tau) \tag{20}$$

$$\frac{\frac{F_w}{R_w}+\frac{F_f}{R_f}+\frac{F_{cei}}{R_{cei}}+\rho_{in.a.}V_{out.a.}c_{pair}+0{,}076c_{p\,air}}{\rho_{in.a.}V_{in.a.}c_{p\,air}}t_{out.a.}+$$
$$\frac{\frac{m_{pr}c_1(t_1-t_3)+M_{pr}q_{bio}}{86400}+\frac{n_{add}P_{add}\tau_{add}+n_w q_w \tau_w}{24}}{\rho_{in.a.}V_{in.a.}c_{in.a.}}$$

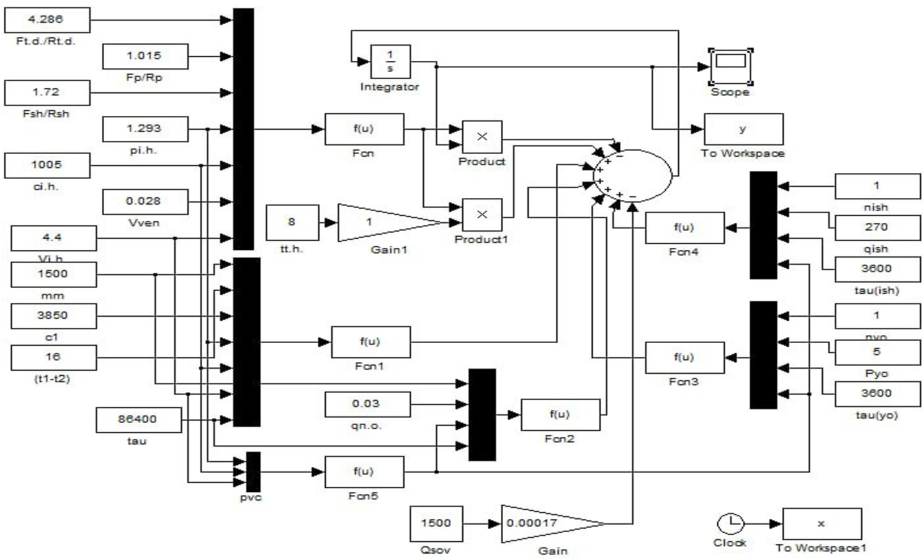

Fig.2. Block diagram of the mathematical model for calculating the internal air temperature of the refrigeration chamber in the Matlab/Simulink program.

3 Results and Discussions

The calculation results for modeling the internal air temperature of the refrigeration chamber are presented in Figure 3.

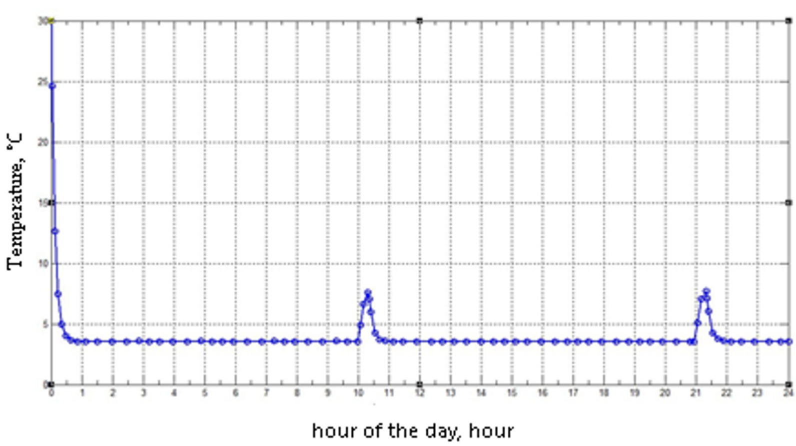

Fig. 3. Change in internal air temperature in the refrigerator compartment.

The results of modeling the temperature of the internal air of the refrigerating chamber show that at an external air temperature of 30°C, 30-40 minutes are enough to reduce the temperature of the product to the required storage temperature. As a result of an employee entering the refrigerating chamber to monitor the refrigerating chamber at any time of the day, the load of the refrigerating chamber changes significantly; this load can be eliminated in 10-15 minutes. At an outside temperature of 30℃, daily electricity consumption is 5.5-6 kWh. Based on the simulation results, it is possible to determine the required cold load to ensure the required temperature conditions inside the refrigeration chamber at different outside temperatures. Thus, the mathematical model of the cooling chamber allows you to regulate the temperature of the internal air of the cooling chamber depending on the temperature of the external air.

4 Conclusions

A mathematical model has been developed based on the thermal balance of the refrigeration chamber using the energy conservation method. The obtained modeling results make it possible to predict and control the optimal temperature regime of the refrigeration chamber depending on changes in ambient temperature and the thermophysical properties of storage objects. The results of modeling the temperature of the internal air of the refrigerating chamber show that at an external air temperature of 30℃, 30-40 minutes are enough to reduce the temperature of the product to the required storage temperature.

Based on the simulation results, it is also possible to determine the required refrigeration load to ensure the required temperature conditions inside the refrigeration chamber at different outside temperatures.

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