# Automatic control of active ventilation systems in agricultural products storage facilities

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**Abstract.** The article recommends automatic control of active ventilation systems in agricultural product storage facilities. The search for rational solutions for ventilation, heating, and automatic control systems (ACS) that improve product storage quality and methods and means of storage is considered.
As a result of the analysis of the requirements for air-cooling storage units that operate for no more than 25 days per annual storage cycle, it was proved that refrigerating arrangements that are based on the principle of operation of vortex energy separators - vortex tubes (VT) meet these requirements most fully. In terms of temperature effect, the VT is inferior to the expander by 1.3 ... 1.5 times, in terms of cooling capacity by 3.0 ... 3.5 times. Compared to the throttling process, vortex tubes, when operating in air, are about 30 times better in terms of temperature effect and 15 times better in terms of cooling capacity.

## 1 Introduction

An analysis of studies in the field of designing storage facilities for agricultural products conducted in recent years has shown that they cover a wide range of issues: structural, layout, and space-planning solutions; development of special chemical compositions and ambient that improve the quality of product storage; improvement of methods and means of storage, and the search for rational solutions for ventilation, heating and automatic control systems (ACS). Currently, up to 70% of all agricultural products are stored in bulk in storage facilities equipped with active ventilation systems [1-6].

Insufficient knowledge of heat-air processes in the upper zone of the storage is confirmed by the presence of the problem of condensation forming on the inner surfaces of the external enclosing structures (EES) and the fill, which leads to sweating and freezing of the upper layers of the stored products. Losses of produce, in this case, can be 2/3 of the total loss. This situation is due to the design of external enclosing structures of the storage according to the existing method of thermal engineering calculation, in which the value of the heat transfer coefficient of the internal surface of the enclosure is normalized in the

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same way as for public or industrial buildings $(\alpha = 8.7Bm/(м^2 \cdot {}^0C))$ not considering the specifics of heat-air processes occurring in storage facilities [3, 6-14].

## 2 Research methods and principles

Considering the processes of heat and mass transfer in the fill as processes of heating and moistening the air, we conclude that in the first layer in the direction of air movement (a corrective layer) of the fill, the air increases the relative moisture from $\varphi_{rh}$ to $\varphi_p$ and is moistened from $d_{rh}$ to $d_{k\min}$, and heated from $t_{rh}$ to $t_{k\min}$ (AB process). Heat and moisture treatment in the main layer occurs along the beam BC, equidistant to the boundary curve $(\varphi_{rh} \approx \varphi_p = cont)$ Quantitative substantiation of moisture loss in the fill at $\varphi_p = cont$ obeys the following dependence (V.Z.Zhadan): $(w \approx Q/\varepsilon_1 = B(1-\varphi) = cont)$. Air is removed from the fill with the parameters of point C. When passing through a colder surface layer, the air can cool to saturation (CD process) or condensate formation (CDj process).

The amount of outside air during the main storage period for the assimilation of moisture released in storage from respiration $w_b$ and decay $w_{de}$, and the total amount and part of recirculated air are, respectively:

$$L_{Hd} = \frac{w_b + w_{de}}{(d_{tk\max} + d_H)p_B}; \; L_{od} = \frac{w_b + w_{de}}{(d_{tk\max} - d_H)p_B}; \; L_{pd} = L_{od} - L_{Hd}; \quad (1)$$

Outside air consumption to remove sensible heat from the unheated storage facility, the total amount of air to remove excess heat from the fill and the part of recirculated air are, respectively:

$$L_{Hl} = (Q_B + Q_{de} + Q_{tex} - Q_H)/\,p_H c_B (t_{k\max} - t_H + \Delta t_{vent}); \quad (2)$$

$$L_{ot} = \frac{Q_B + Q_{de}}{p_H c_B (t_{k\max} - t_{k\max})}; \; L_{pd} = L_{od} - L_{Hd}; \quad (3)$$

The minimum natural loss (the amount of evaporated moisture) is observed when the points are aligned, which is possible under strictly defined ratios of outdoor air $L_H$ and recirculation air $L_p$, when the air is heated or cooled. The latter is accompanied by an increase in the energy intensity of the MCS. With the general exchange mechanical and natural ventilation of storage and storage pits, it is often impossible to achieve such air parameters [17-25].

## 3 Results and Discussion

The methods of supply air treatment are analyzed using the I-d-diagram, on which the area of change in the average monthly parameters of the outdoor air is plotted. The resulting area is divided into several sections characteristic of air processing algorithms, within which it is possible to process outdoor air according to unified algorithms with minimum energy consumption. Within section I, the pre-treatment of outside air is not required if its parameters lie on the $K_1C$ line. Air mixed in proportion $\frac{L_{Hd}}{L_p} = CB/B_1^{'}$ with the parameters of point B enters the fill. In all other cases, before mixing with recirculation air, the minimum amount of outside air is heated $Q_{B1} = c_B p_B L_{Hd}(t_{k1} - t_{H1})$. Additional evaporation of moisture from the SVRM within the parameters of the outside air of section P is prevented during artificial (process $H_2K_2$) or adiabatic (process $H_2K_2^{'}$) cooling of the supply air: $Q_{X2} = p_B L_{H2}(I_{H2} - I_{k2})$. It is possible to bring the SVRM losses to the biologically inevitable ones $\Delta d_x L_{od} p_B$ (section III) either by pre-cooling a part of the outside air using chillers process (process $H_3K_3$) or by adiabatic cooling ($H_3K_3^{'}$) followed by mixing with recirculating air.

The cold consumption is determined by the difference in enthalpies $I_{H^3} - I_{k^3}$ In the zone of section III, which lies above the beam $I_B = const$ (for example, point $H_2^{'}$), with adiabatic air humidification, it is impossible to achieve the parameters of the supply air necessary to minimize the SVRM losses ($t_{k\min,}$ $\varphi_p$ ). In the outdoor climate section IV area, the supply of untreated outdoor air ($t_{нч}$ $\varphi_p$) into the fill leads to its heating. To stabilize and maintain the temperature and humidity regime of the SVRM fill, artificial cold is required $Q_{x4} = p_B L_{H2}(I_{H4} - I_{k4})$. The use of artificial cold in section I is mandatory. To save cold, a first recirculation is possible, in which air with the parameters of point $C_5$ enters the air cooler. The part of the outside air is $L_{Hd} = (\frac{BC_5}{BC_5})L_{od}$. If the supply air parameters do not match the parameters of point B, then a second recirculation is necessary. The amount of air of the second recirculation is $L_{p2} = (\frac{BC_5}{BC_5})L_{od}$. The required parameters of the air entering the fill (section VI) are reached only under machine cooling with the surface temperature of the air coolers below the dew point temperature $t_{mp}$. The construction of air treatment processes and calculating their energy intensity are similar to section V.

Previously known dependences on the normalization of thermal resistance $R_o^{TP}$ of external enclosures of vegetable and potato storage facilities in conjunction with the power of heating systems and biological heat releases $R_o^{TP} = (t_B - t_O)F/[Q_B(1-m) + Q_{he}]$, the capacity of heating systems: $Q_{he} = (t_B - t_O)F/R_o - Q_B(1-m)$ and outdoor air temperature, starting from which air heating systems need to be turned on $t_o^{'} = t_B - [Q_B(1-m) + Q_{he}]/F/R_o + c_B G_{o\min})$ were obtained for the full (calculated) loading of storage capacities. In practice, part of the raw material is sold, or the containers are not completely filled at the loading time, which leads to a shortage of heat due to a decrease in the amount of biological heat release.

The power of heating systems $Q_{he}^{dd}$ , in this case, increases inversely with the amount of loading capacity $\alpha = \frac{G_d}{G_p}$;

$$Q_{he}^{dd} = (G_p - G_d)q_v = G(1-\alpha)q_v \tag{4}$$

Considering the value of *a*, the outdoor air temperature $t'_н$ takes the following values:

$$t_o^{'} = t_B - \alpha G_p q_v /(F / R_o^{TP} + \alpha c_B G_{o\min}), \tag{5}$$

Required capacity of heating systems is:

$$Q_{he}^{T} = (t_B - t_H)F / R_o^{TP} = G_p(1-\alpha)q_v(1-m). \tag{6}$$

As a result, quantitative characteristics of the coefficient of products safety $K_{se}$ were experimentally and analytically determined: $K_{se.s} = 0.95\ K_{se.k}\ K_{se.e}\ K_{se.ref}$.

It is recommended to take the coefficient of quality assurance for storage of medium-sized fills of potatoes and vegetables $К_{se.к}$ within the following limits: for potatoes $K_{se.k} = 0.90..0.92$; for table beet $K_{se.k} = 0.93..0.95$; for carrots $K_{se.k} = 0.88..0.90$; for cabbage $K_{se.k} = 0.95..0.97$.

A quantitative definition of the storage microclimate security factor can be provided in the form $K_{se.m} = f(K_{se.o}, K_{se.b}, K_{se.p}, K_{se.з}, K_{ose.\varphi}, K_{se.L}, K_{se.ref})$. Storage security factor of SVRM when using air heating systems $К_{se.o}$ of power N are characterized by the values: for potatoes $K_{se.o} = 1 - 2.07N / G_p$: for cabbage $K_{se.o} = 1 - 1.3N / G_p$; for table beet $K_{se.o} = 1 - 1.07N / G_p$; for carrots $K_{se.o} = 1 - 1.24N / G_p$.

The heating of ventilation air in fans and air ducts increases the natural loss of production, which corresponds to the coefficients of supply for radial fans $K_{se.B} = 0.999$, for axial fans $K_{об.В} = 0.9995$. The coefficient of microclimate security with uniform sales of products $K_{se.p}$ is determined by the same dependencies used for air heating systems (the value of $G_p$ is replaced by $aG_p$). The same dependences determine the coefficients of microclimate security under incomplete initial loading of storage facilities $K_{se.з}(G_d = aG_p)$.

When storing different agricultural products in one room, the humidity regime is determined by the products with the highest value $\varepsilon_u$. We classified other agricultural products according to their compatibility during storage in the same room [17, 18, 19, 26, 27, 28, 29].

The developed systems of cold and heat supply storage facilities using the VT consider their most rational geometric dimensions, operating conditions, and prospects for use. The required consumption of cold $Q_{x1}$, generated by the VT to maintain the technological microclimate, is calculated for a storage facility with a capacity of $G_p$ =500 tons.

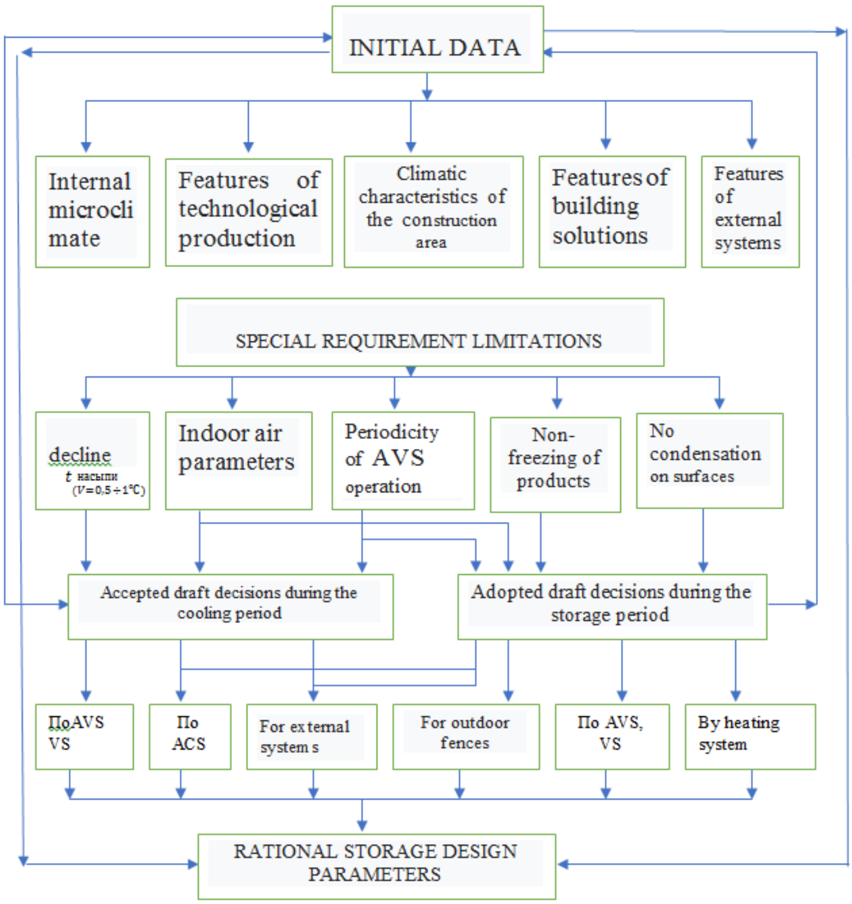


**Fig. 1.** The structure of initial data for setting the optimization problem in the storage of agricultural products

Cold and heat performance of the studied VT at $\mu$=0.55…0.65 are given in Table. 1. The efficiency of the VT studied by the authors with the results of tests of VT with large diameters performed by other authors is in good agreement.

**Table 1.** Cold and heat performance of the studied VT

| Pressure $\pi$ | G, kg/h | $q_x$ kJ/kg | $Q_x$ W | $Q_r$ W |
|---|---|---|---|---|
| 5 | 30.86 | 15.0 | 128.6 | 385.8 |
| 4 | 22. 63 | 13.0 | 81.9 | 164.5 |
| 3 | 16.02 | 10.5 | 46.7 | 120.2 |

A model that takes into account the specific features of storage, in the form of a storage facility with minimal heat transfer properties of the EES $R^{\max}$ (complete thermal insulation of the upper storage area) and, as a result, the absence of heat consumption for heating the upper storage area ($Q_{co} = 0$), as well as when consuming in the AVS, the minimum inevitable amount of outside air $(G_{p_{H\min}})$ in combination ensures the maximum safety of products ( $M_{\max}$ (Fig. 1). The state of any system is characterized by the corresponding tuple of parameters. The tuple of the ideal model is represented by the following values: $R^{\max}, Q_{co} = 0, Q_{H\min, M_{\max}}$ .

The main system of equations was derived according to the methodology for constructing mathematical models of heat-air processes in rooms; it consists of the heat and humidity balances of the day for all characteristic volumes and surfaces. Radiant, convective, and mass transfer flows are replaced by the corresponding parameters reflecting the physical meaning of the ongoing processes [8].

Here, the main system of equations is not fully presented; an example of the calculated form of heat balance equations is given:

- for the upper layer of the fill:

$$M_{BC} \times g_t = c_B \times M_{BC} \times \frac{t_M^k - t_M^H}{2h} + 1.69 \times m \times \sqrt[3]{|\tau_1 - t_B|} \times (\tau_1 - t_B) \times F_{mou} + 5.77 \times \varepsilon_{np}^{I-2} \times \left[\left(\frac{\tau_1 + 273}{100}\right)^4 - \left(\frac{\tau_2 + 273}{100}\right)^4\right] \times \varphi^{I-2} \times F_i - c_{np}^{H-1} \times \varphi^{H-1} \times \left[\left(\frac{\tau_1 + 273}{100}\right)^4 - \left(\frac{\tau_2 + 273}{100}\right)^4\right] \times F_H + r \times \beta \times \left[P_{\tau_1^n} - \varphi_B \times P_{t_B^n}\right] \times F_{mou} \tag{7}$$

- for the upper area of the storage:

$$Q_{BO} = G_{Bn}^{BH} \times c_B \times (t_B - t_{Bn}^{BH}) \times i - 1.69 \times m \times \sqrt[3]{|t_{Bn}^I - \tau_4^I|} \times (t_{Bn}^I - \tau_4^I) \times F_4^I - 1.69 \times m \times \sqrt[3]{|t_{Bn}^{II} - \tau_4^{II}|} \times (t_{Bn}^{II} - \tau_4^{II}) \times F_4^{II} + 1.69 \times m \times \sqrt[3]{|t_{Bn}^I - \tau_3^I|} \times (t_{Bn}^I - \tau_3^I) \times F_3^I - 1.69 \times m \times \sqrt[3]{|t_{Bn}^{II} - \tau_3^{II}|} \times (t_{Bn}^{II} - \tau_3^{II}) \times F_3^{II} + 1.69 \times m \times \sqrt[3]{|\tau_i - t_B|} \times (\tau_i - t_B) \times F_{Hac} + 1.69 \times m \times \sqrt[3]{|t_i - \tau_2|} \times (t_i - \tau_2) \times F_2 - A_2 \times \left[\frac{|t_B - \tau_n|}{d}\right]^{1/4} \times (\tau_n - t_B) \times F_n; \tag{8}$$

- for the inner surface of the ceiling floor:

$$5.77 \times \varepsilon_{np}^{i-2} \times \left[\left(\frac{\tau_1 + 273}{100}\right)^4 - \left(\frac{\tau_2 + 273}{100}\right)^4\right] \times \varphi^{i-2} \times F_i + 1.69 \times m \times \sqrt[3]{|t_B - \tau_2|} \times (t_B - \tau_2) \times F_2 + \left[\left(\frac{\tau_n + 273}{100}\right)^4 - \left(\frac{\tau_2 + 273}{100}\right)^4\right] \times 5.77 \times \varepsilon_n \times \varepsilon_2 \times \varphi^{n-2} \times F_i = \frac{\tau_2 - t_n}{R_2 + \frac{1}{a_n}} \times F_2 \tag{9}$$

The system of balance equations for characteristic volumes and surfaces, as well as special relationships that reflect the specific conditions of heat and mass transfer (no condensation on the surfaces of enclosures and fills and no freezing of products), present the sought-for mathematical model that describes the heat-air processes occurring in the agricultural products storage facilities, in their interconnection and interaction.

Using the engineering methodology developed, calculations were conducted on a computer for ground-based potato storage equipped with AVS, with a wide variation in the determining factors: indoor air parameters $t_B = 1 \div 4^0 C$ and $\varphi_B = 70 \div 95\%$ space-planning

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solutions $m\frac{F_1}{F_2}=0.6;$ 0.8; 1; different climatic zones with outdoor temperature $t_0$ = -20; -30; -40 $^0C$.

For these options, an assessment was made of the microclimate parameters in storage facilities designed according to the existing method of thermal engineering calculation, i.e., at $\alpha = 8.7Bm/(м^2 \cdot {}^0C)$. The results of the calculations made it possible to conclude that at design outside temperatures below -20°C, the temperatures on the surfaces of the fills ($\tau_1$), floors ($\tau_2$), walls ($\tau_3$), and bins ($\tau_4$) are below 0°C and less than the dew point temperature ($t_p$ The required parameters of the internal microclimate are not maintained in the storage rooms; this leads to condensation on the enclosures and fills, as well as sweating and freezing of products. Thus, in the first stage of using the developed mathematical model, the shortcoming of the existing design methodology is confirmed [26, 30].

In the mode between switching on AVS, there is a continuous flow of moisture into the air of the storage's upper zone from the product's upper layer.

To study the processes of moisture exchange in the upper zone of the storage in the mode between the operation of the AVS, a differential equation was compiled under the boundary condition $d_{B\ |h=0} = d_{ini}$, the solution of which made it possible to estimate the change in the relative air humidity $\varphi_B$ depending on the time between the switching on AVS (*h*, hour):

$$\varphi_B = \frac{P_{\tau_i^H} - (P_{\tau_i^H} - \varphi_{ini} \times P_{t_i^H}) \times \exp\left(-\frac{\beta \times F_{mou} \times P_б}{0.623 \times P_{t_i^H} \times V_B \times p_B} \times P_{t_B^H} \times h\right)}{P_{t_B^H}}. \quad (10)$$

The calculation results showed that in the mode between the switching on AVS, even the minimum volume of the upper zone (for $h_{uz}$ = 0.5 m) allows accumulating excess moisture from the upper layer of the products (the relative air humidity in the upper zone does not exceed the value of the normalized value given in ONTP-6-88) [3, 4, 5,19].

The duration of the upper zone $(h^{add})$ ventilation is determined from a differential equation with initial conditions $d_{B\ |h^{add}=0} = d_{mou}$:

$$h^{add} = -\frac{V_B}{L_{EES}} \ln\left(\frac{d_{ini} - \varphi_B d_{mou}}{d_{ini} - d_{mou}}\right). \quad (11)$$

Outside air temperatures during the experiment varied within $t_H = -8 \div 22^0C$. The shortcoming of the existing method of thermal engineering calculation of the EES of storage facilities was confirmed since even at outdoor temperatures above the calculated ones $(t_o > t_{np} = -37^0C)$ (there occurred the condensation formation and sweating of surfaces.

## 4 Conclusion

As a result of the requirement analysis for air-cooling units in storage facilities that are operated for no more than 25 days per annual storage cycle proved that refrigeration units based on the principle of operation of vortex energy separators - vortex tubes (VT) meet these requirements most fully. In terms of temperature effect, the VT is inferior to the expander by 1.3 ... 1.5 times, and in terms of cooling capacity - by 3.0 ... 3.5 times. Compared to the throttling process, vortex tubes, when operating in air, are about 30 times better in terms of temperature effect and 15 times better in terms of cooling capacity.

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