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Studying the air flow heating process in the vertical type ground heat exchanger

***Abstract**. The article presents the results of analytical studies of air losses of vertical ground heat exchangers for two proposed schemes (concentric and U-shaped). The distribution of the temperature field during the summer and winter periods was obtained through numerical simulation of the air heating process in concentric and U-shaped vertical ground heat exchangers using the Simcenter Star-CCM+ software package.*

Streszczenie.** W artykule przedstawiono wyniki badań analitycznych strat powietrza w pionowych gruntowych wymiennikach ciepła dla dwóch zaproponowanych schematów (koncentrycznego i U-kształtnego). Rozkład pola temperatur w okresie letnim i zimowym uzyskano poprzez symulację numeryczną procesu nagrzewania powietrza w koncentrycznych i pionowych gruntowych wymiennikach ciepła w kształcie litery U z wykorzystaniem pakietu oprogramowania Simcenter Star-CCM+.* (Badanie procesu nagrzewania przepływu powietrza w pionowym gruntowym wymienniku ciepła***)

Keywords: soil heat exchanger, numerical modeling, experimental studies, pneumatic losses, effective heat capacity.
Słowa kluczowe: gruntowy wymiennik ciepła, modelowanie numeryczne, badania eksperymentalne, straty pneumatyczne.

Introduction

Global trends of growing prices for traditional fuel resources used for electricity generation require agricultural enterprises to diversify electricity supply sources and increase the level of power autonomy. Despite the widespread present-day opinion regarding the low efficiency of wind and solar energy utilization in Ukraine’s natural and climatic conditions, the experience of highly developed countries testifies to the contrary:

- implementation of projects related to introduction of autonomous power supply systems for agro-industrial enterprises based on SPPs and WPPs has environmental and economic advantages over traditional power supply [1];
- today’s development of technologies makes it possible to convert of solar and wind energy into electric power in the territories that were previously (20-30 years ago) considered unsuitable for this type of energy [2].

Development of the organic sector is particularly significant and promising for domestic farmers, consumers and the state as a whole, especially in the context of ensuring food supply security, healthy nutrition and preservation of natural environment. In accordance with strategic goal 1 “Ensuring a stimulating and reasonable agro-policy” being part of the Strategic policy course in the field of agro-industrial sector development, one of the ways to achieve the strategic goal lies in support of organic production. Another way to achieve the goal is to ensure the development of sustainable production, in which respect the Government’s task is defined as encouragement of sustainable agricultural production, protection of environment and fauna, spreading the use of organic production methods and the use of biotechnology, “climate-smart” agriculture and forestry along with reduction of greenhouse gas emissions and adaptation to climate change, sustainable management of natural resources, as well as biodiversity preservation and augmentation [3].

The livestock breeding sector of agro-industrial production has the greatest potential for increase of power use efficiency. It can be seen that the power used for air cooling makes a significant part of total energy consumption, which is constantly increasing due to increased requirements for maintenance of optimal microclimate in livestock premises [4, 5].

The largest share of power consumption in livestock breeding premises falls on generation of standard microclimate parameters, particularly those required for heating of supplied ventilation air. According to various estimates, during the heating period, these premises’ heat-generating devices consume from 40 to 90% of the total cost of fuel and power resources [5]. Hence, even partial reduction of these expenses will lead to a significant reduction in the livestock products’ cost.

An effective way to reduce power consumption in livestock premises is to utilize ventilation emissions’ heat to warm up supplied ventilation air. The difficulty of utilizing the ventilation emissions’ heat lies in exhaust air’s being a low-potential source of thermal energy [6].

To ensure air removal from piggery premises, an automatic ventilation system for polluted air intake from livestock premises was created [6]. As a result of analytical studies of this system, the conditions for its effective operation have been mathematically presented.

In the larger part of Ukraine, soil is the most accessible source of low-potential heat. At the depths of over 10 meters, it maintains constant its temperature ranging from +9 to 12°C throughout the year. This creates favorable conditions for an efficient use of heat pumps. Ukraine has a significant potential for using the heat of soil and groundwater [7]. The temperature of soil and rocks on the top of the ground depends on the balance of thermal energy coming from the Sun and thermal radiation from the earth surface. Thermal energy coming from the Sun gets accumulated in the layer of sedimentary strata and rocks at the depths up to the isothermal surface. This soil layer may be considered as a natural seasonal accumulator of thermal energy, since the energy withdrawn in winter period is restored in summer. This also applies to groundwaters, which saturate the upper layer of soil and sedimentary strata [8].

Analysis of technical means of thermal energy extraction from surface layers of soil [9–11] made it possible to determine that vertical-type ground heat exchangers are the most effective ones among considered options used to achieve required microclimate parameters in livestock premises. However, their optimal parameters, location and energy efficiency limits have not been sufficiently covered in the literature.

Relying on the analysis of previous studies of structural and process diagram of ground-based air heat exchanger, let us consider the two options being most cost-effective in terms of their manufacture. The first option is based on studies [12–14] – a concentric-type vertical ground heat exchanger. For this purpose, the structural and process parameters were chosen that were obtained as a result of experimental research (Fig. 1, a). The second option is a more classical one – the U-shaped vertical-type ground heat exchanger [15–16] (Fig. 1, b)

According to research [17] (in which water was the heat transfer agent), a concentric-type heat exchanger is the most effective one for short-term operation, but for long-term operation, its efficiency drops by 18–20%. In addition, our own experience in research of concentric-type heat exchangers [18] shows that a concentric-type pipe-in-pipe arrangement causes secondary temperature change through internal walls. This negatively affects the process of air flow heating or cooling the within soil. It should be noted that under internal walls' thermal insulation, sufficiently large non-operating zone is generated, this only leading to pneumatic losses.

Analytical studies of ground heat exchanger's pneumatic losses

The first stage in determining the most rational arrangement of ground-based air heat exchanger is calculation of air pumping power.

Let's divide the air path into five sections (Fig. 1). According to [19], pressure losses in each section are summarized in table. 1.

Table 1 – Pneumatic pressure losses in all sections of vertical-type ground heat exchangers

Area	Concentric-type	U-shaped
I	(1) $\Delta p_{CI} = 0{,}11\frac{273\rho_{\text{н.у.}}\,L_C}{2TD_{C1}}\left(\frac{4q_{in}}{\pi D_{C1}^2}\right)^2\left(\sqrt[4]{\frac{17\mu T\pi D_{C1}}{273q_{in}\rho_{\text{н.у.}}}}+\frac{\psi}{D_{C1}}\right)$	(6) $\Delta p_{UI} = 0{,}11\frac{273\rho_{\text{н.у.}}\,L_U}{2TD_{U1}}\left(\frac{4q_{in}}{\pi D_{U1}^2}\right)^2\left(\sqrt[4]{\frac{17\mu T\pi D_{U1}}{273q_{in}\rho_{\text{н.у.}}}}+\frac{\psi}{D_{U1}}\right)$
II	(2) $\Delta p_{CII} = \zeta_{CII}\frac{273\rho_{\text{н.у.}}}{2T}\left(\frac{4q_{in}}{\pi D_{C1}^2}\right)^2$	(7) $\Delta p_{UII} = 2\frac{273\alpha\rho_{\text{н.у.}}}{T}\left(\frac{4q_{in}}{\pi D_{U1}^2}\right)^2$
III	(3) $\Delta p_{CIII} = 0{,}11\frac{273\rho_{\text{н.у.}}\left(L_C - L_{C1}\right)}{2T\sqrt{D_{C2}^2-D_{C1}^2}}\left(\frac{4q_{in}}{\pi D_{C1}^2}\right)^2\times \left(\sqrt[4]{\frac{17\mu T\pi D_{C1}^2}{273q_{in}\sqrt{D_{C2}^2-D_{C1}^2}\,\rho_{\text{н.у.}}}}+\frac{\psi}{\sqrt{D_{C2}^2-D_{C1}^2}}\right)$	(8) $\Delta p_{UIII} = 0{,}11\frac{273\rho_{\text{н.у.}}\left(L_U - L_{U1}\right)}{2TD_{U1}}\times \left(\frac{4q_{in}}{\pi D_{U1}^2}\right)^2\left(\sqrt[4]{\frac{17\mu T\pi D_{U1}}{273q_{in}\rho_{\text{н.у.}}}}+\frac{\psi}{D_{U1}}\right)$
IV	(4) $\Delta p_{CIV} = \frac{273\alpha\rho_{\text{н.у.}}}{T}\left(\frac{4q_{in}}{\pi D_{C1}^2}\right)^2$	(9) $\Delta p_{UIV} = \frac{273\alpha\rho_{\text{н.у.}}}{T}\left(\frac{4q_{in}}{\pi D_{U1}^2}\right)^2$
V	(5) $\Delta p_{CV} = 0{,}11\frac{273\rho_{\text{н.у.}}\,L_{C2}}{2TD_{C1}}\left(\frac{4q_{in}}{\pi D_{C1}^2}\right)^2\left(\sqrt[4]{\frac{17\pi D_{C1}\mu T}{273q_{in}\rho_{\text{н.у.}}}}+\frac{\psi}{D_{C1}}\right)$	(10) $\Delta p_{UV} = 0{,}11\frac{273\rho_{\text{н.у.}}\,L_{U2}}{2TD_{U1}}\left(\frac{4q_{in}}{\pi D_{U1}^2}\right)^2\left(\sqrt[4]{\frac{17\pi D_{U1}\mu T}{273q_{in}\rho_{\text{н.у.}}}}+\frac{\psi}{D_{U1}}\right)$

where $\rho_{\text{н.у.}}$ = 1.293 kg/m^3 – air density under normal conditions; L – air duct length, m; T – air flow temperature, K; D – air duct diameter, m; q_{in} – input air flow rate, m^3/s; $\mu = 18.27\cdot10^{-6}$ N·s/m^2 – dynamic air viscosity; ψ = 0.1 mm (for polyethylene) – equivalent roughness of air duct walls; ζ= 2 – local resistance coefficient for spatial (circular) rotation by 180° during injection; α= 0.55 – impact mitigation factor for a constant cross-section knee.

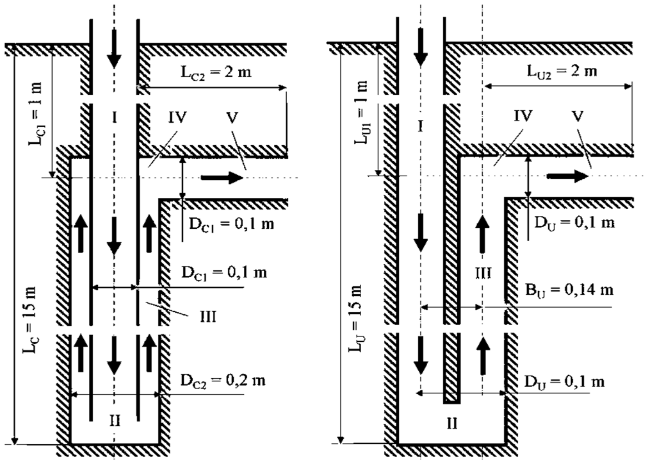

Fig. 1. Calculation model of vertical-type ground heat exchangers, a – concentric-type; b – U-shaped

Pressure losses in the heat exchanger can be calculated as the sum of all pressure losses in each section

(11) $$\Delta p_{ST} = \Delta p_I + \Delta p_{II} + \Delta p_{III} + \Delta p_{IV} + \Delta p_V .$$

Also, the power required for air pumping through a vertical heat exchanger is determined using the formula:

(12) $$N_{ST} = \frac{q_{in}\,\Delta p_{ST}}{\eta_п}$$

where η_n is the fan's overall efficiency, η_n = 0.85.

Having performed a joint calculation of dependencies (1)–(12) using Wolfram Cloud software, we obtained the dependencies between pneumatic losses' power change N_{ST} of U-shaped vertical-type ground heat exchanger and its length L_{U1} , diameter D_{U1} and air flow injection q_{in} shown in Fig. 2.

Comparing the power values of pneumatic losses in concentric-type and U-shaped heat exchangers, it was determined that the power of the latter is higher by 0.9–1.7%. That is, air losses in both options of vertical-type ground heat exchangers are almost the same. Therefore, to evaluate the efficiency, one should investigate their heat capacity.

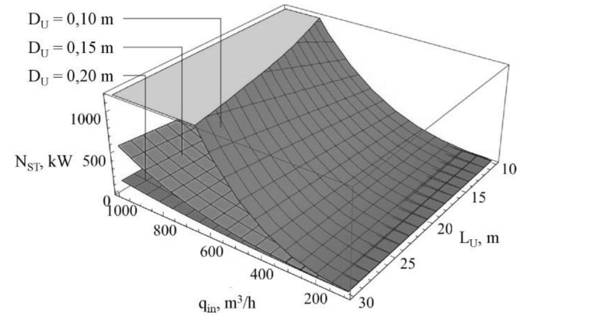

Fig. 2. Dependencies between pneumatic losses' power change N_{ST} of U-shaped vertical-type ground heat exchanger and its length L_{U1}, diameter D_{U1} and air flow injection q_{in}

Physical-and-mathematical apparatus of air flow heating process in a vertical-type ground heat exchanger

The second stage involves the physical-and-mathematical apparatus of air flow heating process in a vertical-type ground heat exchanger.

Calculation models of two options of vertical-type ground heat exchangers housed in the ventilation system for clean air injection [20–21] are shown in fig. 3.

A rectangular coordinate system is chosen in such a way that axis OX is directed along the horizon and parallel to the axis of the heat exchanger's horizontal section, with axis OY being directed perpendicular to the plane of the figure, and OZ axis directed vertically down.

We take the following assumptions [22]:

– the soil is homogeneous and isotropic, and its thermophysical properties remain unchanged when temperature changes;

– the thermal contact between the ventilation system's wall and surrounding soil is ideal;

- due to a slight change in the air flow pressure during its movement in the ventilation system, air is deemed to be an incompressible liquid.

The heat flowing through the ventilation system's surface S at a given moment of time τ can be represented by formula [23-24]

$$\frac{dq}{d\tau} = -\lambda_S \int_S \frac{\partial T_S}{\partial n} dS \tag{13}$$

where T_S (x,y,z,τ) is the temperature at the soil point, which has coordinates (x, y, z) at time moment τ, °C .

The continuity equation that reflects the fact that there are no voids and gaps in the field occupied by air M_{air}, in the accepted rectangular coordinate system, has the following form

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0, \ (x,y,z) \in M_{air} \tag{14}$$

where u, v, w are velocity components in x, y, z directions.

Air movement in such a case is described by Navier-Stokes equations, which take the form for a rectangular coordinate

$$\begin{gathered} \rho\frac{Du}{\partial\tau} = \frac{\partial p}{\partial x} + \mu\nabla^2 u, \qquad \rho\frac{Dv}{\partial\tau} = \frac{\partial p}{\partial y} + \mu\nabla^2 v, \\ \rho\frac{Dw}{\partial\tau} = \frac{\partial p}{\partial y} + \mu\nabla^2 w, \qquad (x,y,z) \in M_{air}. \end{gathered} \tag{15}$$

Substantial derivatives making part of the previous equations are expressed by the following dependencies:

$$\begin{aligned} \frac{Du}{\partial\tau} &= \frac{\partial u}{\partial\tau} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}, \\ \frac{Dv}{\partial\tau} &= \frac{\partial v}{\partial\tau} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}, \\ \frac{Dw}{\partial\tau} &= \frac{\partial w}{\partial\tau} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}. \end{aligned} \tag{16}$$

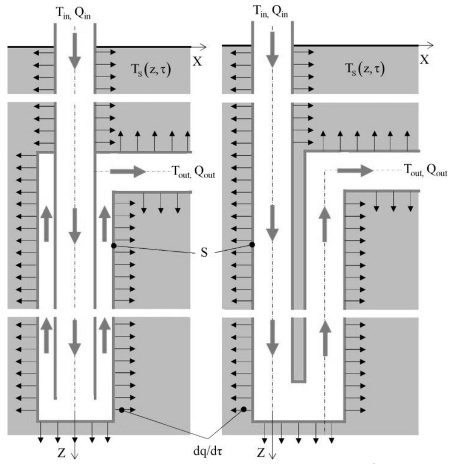

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Fig. 3. Calculation models of concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers of the clean air injection system

The Laplace operator in a rectangular coordinate system has the following form^

$$\nabla^2 = \left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}\right). \tag{17}$$

The temperature field in a moving air flow is described by energy equation [25–26]

$$\frac{DT_a}{\partial\tau} = a_a\nabla^2 T_a, \quad (x,y,z) \in M_a, \tag{18}$$

where T_a (x,y,z,τ) is the air temperature at the air point, which has coordinates (x,y,z) at time moment τ, °C.

The derivative included in previous equation (18) is written as follows

$$\frac{DT_a}{\partial\tau} = \frac{\partial T_a}{\partial\tau} + u\frac{\partial T_a}{\partial x} + v\frac{\partial T_a}{\partial y} + w\frac{\partial T_a}{\partial z}. \tag{19}$$

The temperature field in the soil stratum is described by the following thermal conductivity equation [27–28]

$$\frac{\partial T_s}{\partial\tau} = a_s\nabla^2 T_s, \quad (x,y,z) \in M_s, \tag{20}$$

Initial conditions [29]

$$\begin{cases} T_S(x,y,z,0) = T_{S0}(z), & (x,y,z) \in M_S, \\ T_a(x,y,z,0) = T_{a0}(z), & (x,y,z) \in M_a. \end{cases} \tag{21}$$

Boundary conditions

$$\begin{cases} T_a(x,y,0,\tau) = T_1(x,y), & (x,y,z) \in M_a, \\ T_S(x,y,0,\tau) = f(\tau), & (x,y,z) \in M_a. \end{cases} \tag{22}$$

where T_S (x,y,0,τ) is the function that determines the soil surface temperature and depends on natural and climatic conditions [30–33]:

$$T_S(z,\tau) = T_m - A_s e^{-z\sqrt{\frac{\pi}{365\alpha_s}}} \cos\left(\frac{2\pi}{365}\left(\tau - \tau_0 - \frac{z}{2}\sqrt{\frac{365}{\alpha_s\pi}}\right)\right) \tag{23}$$

where T_S (z, τ) is the soil temperature at time τ and depth z, °C; T_m – the average soil surface temperature, °C; A_s – the amplitude of soil surface change, °C; α_s – thermal conductivity coefficient of the soil, m^2 /day; τ – the time elapsed since the beginning of the calendar year, day; τ_0 is the phase constant of the soil surface, day.

The condition of heat flow density equality on the ground heat exchanger's wall

$$\alpha = -\frac{\lambda_a}{T_a - T_s}\frac{\partial T_a}{\partial n}. \tag{24}$$

By approximating the ten-year data on the temperature on the surface of Vinnytsia region soils [34] in the form of a cosine function, we got the graph shown in Fig. 4.

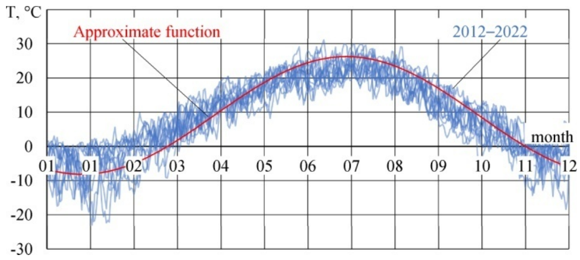

Fig. 4. Soil surface temperature according to long-term observations in Vinnytsia region

Taking into account the approximated data, the values of T_m = 9.79°C, A_s = 22.005°C, τ_0 = 23 days were determined. The coefficient of soil thermal conductivity α_s = 0.0253-0.089 m^2 /day depends on soil density, humidity and type. For further calculations we assume the largest value of soil thermal conductivity coefficient α_s = 0.089 m^2 /day. Then, by substituting the obtained values into equation (23), we obtain graphs of soil temperature distribution by depth (Fig. 5)

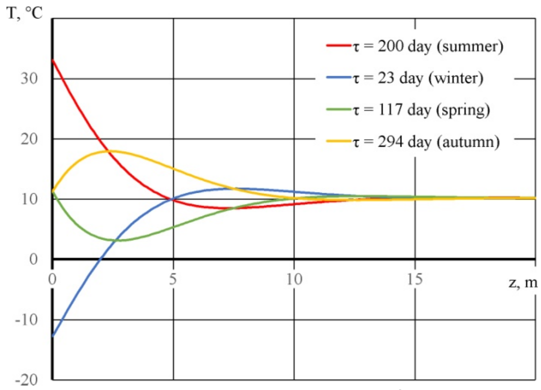

Fig. 5. Temperature distribution by soil depth for Vinnytsia region

It was determined from Figure 5 that for natural and climatic conditions of Vinnytsia region, soil temperature fluctuations are the smallest ones (0.5°C) starting from depth z = 12.3 m, which can be taken as the smallest effective depth for the ground heat exchanger.

The obtained results will be used for numerical modeling of vertical-type ground heat exchangers.

Results of numerical modeling of air flow heating process in vertical-type ground heat exchangers

At the third stage, provision is made for a numerical modeling of the air flow heating process in vertical-type ground heat exchangers.

Numerical modeling was performed using Simcenter Star-CCM+ software package, which employs the spatial discretization method to use the finite volume method with calculation of unknown cells in the centers. In order to reduce the number of finite elements' grid elements and to save computing resources, the symmetry area along plane XOZ was used.

The general appearance of the calculated grid, as well as structural-and-technological parameters of two options of vertical-type ground heat exchangers are shown in fig. 6.

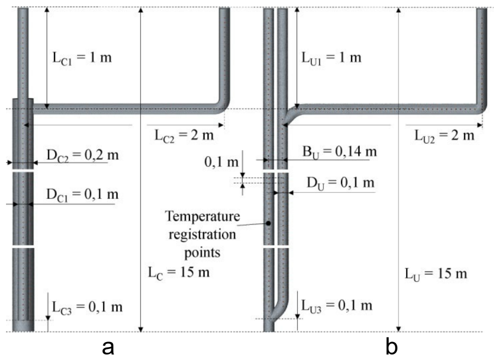

Fig. 6. General view of the calculated grid, as well as structural-and-technological parameters of concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers

The following were selected as physical models of air: the three-dimensional one, the Eulerian multiphase model, VOF method of separated flow and volume liquid, the phase interaction model and the model of separated multiphase temperature. Current flow is subject to the Navier-Stokes equation and k-ε model of turbulence. Euler phases were air and water. The air phase was subjected to MASVP-PR97 real gas (vapor) and turbulent flow models. The water phase obeyed the van der Waals real gas and turbulent flow models [35]

Physical models of the walls of the ventilation system for clean air injection are as follows: three-dimensional model of solid body material, constant density and the model of separated solid body's energy.

The following models were selected as physical models of soil: three-dimensional model of solid body material, constant density and the model of separated solid body's energy.

Temperature distribution in soil is subject to equation (23), i.e. fig. 5.

A non-stationary implicit solver was chosen. The number of internal inertias was equal to 10. Total simulation time – 10^5 s.

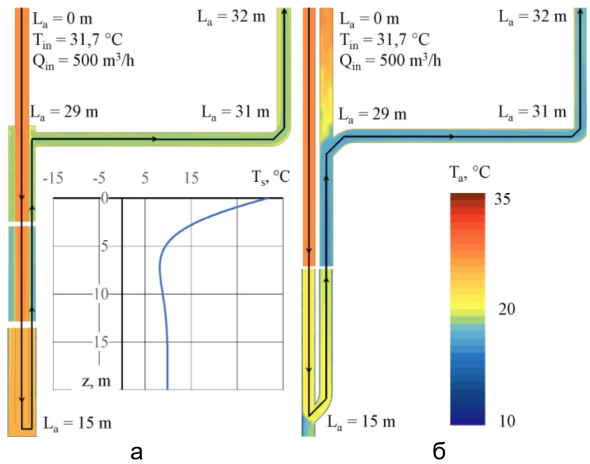

Fig. 7. Temperature field distribution in concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers in summer period with the air injection of 500 m^3 /h

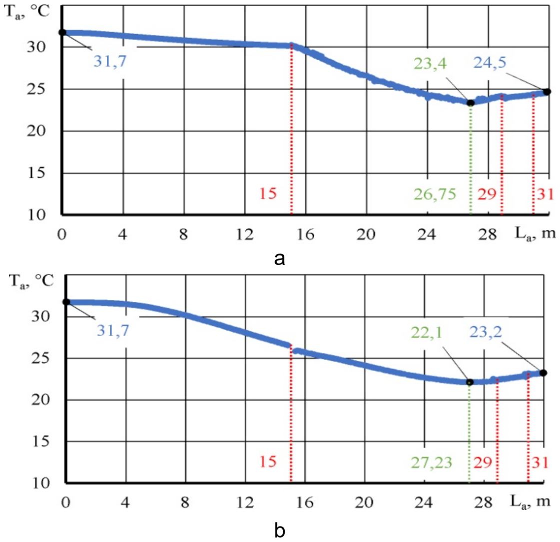

Fig. 8. Graph of dependence between air flow temperature T_a and its path L_a in concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers in summer period with the air injection of 500 m^3/h

According to the simulation results, the distribution of the temperature field in vertical-type ground heat exchangers in summer period was obtained (Fig. 7).

For more accurate display of simulation results, the graph of dependence between air flow temperature T_a and its path L_a was plotted , which is shown in Fig. 8.

It follows from fig. 7–8 that summer temperature of the air flow, which moves along the pipes with the total length of 32 m, decreases from 31.7°C to 24.5°C in concentric-type heat exchangers and to 23.2°C in U-shaped heat exchangers.

That is, the largest temperature difference is observed for the U-shaped heat exchanger (8.5°C) as opposed tot the value of the concentric-type heat exchanger (7.2°C).

By analyzing the graphs in fig. 8, it was determined that the lowest temperature observed for the air flow path is 26.75 m (concentric-type heat exchanger) and 27.23 m (U-shaped heat exchanger), being 23.4°C and 22.1°C, respectively. This testifies to the need for the air duct's thermal insulation at the depth of approximately 2.77–3.25 m, this to ensure preservation of the lowest temperature right up to the exit from the heat exchanger.

Let us conduct a similar analysis with respect to the results of the numerical simulation in winter period. Temperature distributions and graphs of temperature changes are shown in fig. 9–10.

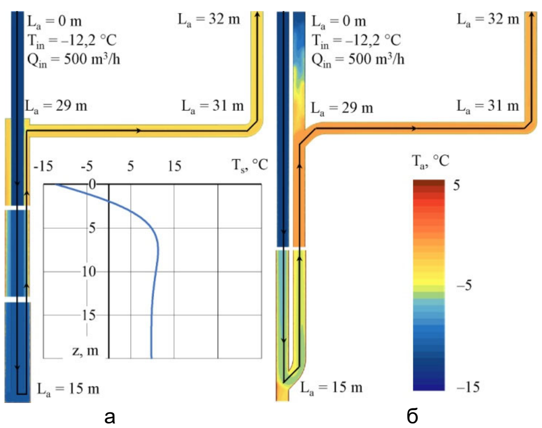

Fig. 9. Temperature field distribution in concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers in winter time period with the air injection of 500 m^3/h

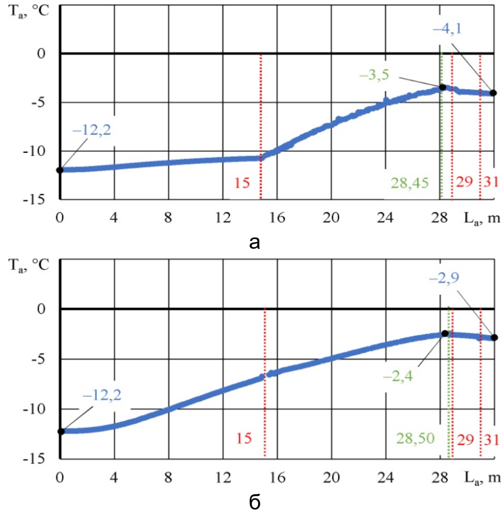

Fig. 10. Graph of dependence between air flow temperature T_a and the path of its movement L_a for concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers in winter period with the air injection of 500 m^3/h

It was determined from fig. 9–10 that winter-period temperature of the air flow that moves along pipes with the total length of 32 m, increases from –12.2 °C to –4.1 °C for the concentric-type heat exchanger and to –2.9 °C for U-shaped heat exchanger. That is, the largest temperature difference is observed for the U-shaped heat exchanger (9.3 °C) as opposed tot the value of the concentric-type heat exchanger (8.1 °C).

Analyzing the graphs shown in fig. 10, it was determined that the highest temperature is observed for the air flow path of 28.45 m (concentric-type heat exchanger) and 28.50 m (U-shaped heat exchanger), being –3.5°C and –2.4°C, respectively. This confirms the previous conclusion regarding the need for the air duct's thermal insulation at the depth of 1.50–3.25 m.

In future, we are going to use air ducts' thermal insulation at the depth of 1.50–2.77 m and take into account the lowest temperature at the exit from the heat exchanger.

To substantiate rational mode parameters, we are going to vary them within the following limits:

– inlet air temperatures T_{in} : –12.2°C, –1.225°C, 9.75°C, 20.725°C and 31.7°C;

– air consumption Q_{in} : 200 m^3/h, 350 m^3/h, 500 m^3/h, 650 m^3/h and 800 m^3/h.

According to the results of numerical modeling using Simcenter Star-CCM+ software package, processing of the data obtained using Wolfram Cloud software package made it possible to obtain a second-order regression equation that shows the dependence between the change in air flow temperature ΔT_a and research factors in the following form for each option of heat exchangers (Fig. 11):

– concentric-type heat exchanger

(25) $$\Delta T_{aC} = 5{,}91513 - 0{,}00720618\, Q_{in} - 0{,}323773\, T_{in} + 0{,}0161927\, T_{in}^2;$$

– U-shaped heat exchanger

(26) $$\Delta T_{aU} = 7{,}81976 - 0{,}00955501\, Q_{in} - 0{,}42689\, T_{in} + 0{,}000105189\, Q_{in} T_{in} + 0{,}018366\, T_{in}^2.$$

Figure 11 shows the dependence between changes in air flow temperature ΔT_a and inlet air temperature T_{in}, as well as air consumption Q_{in} for concentric-type and U-shaped vertical-type ground heat exchangers. It has been clearly determined that the difference is greater for the U-shaped heat exchanger. The increase in air consumption Q_{in} also leads to decrease in ΔT_a. The highest value of ΔT_a was observed for high (summer period, 31.7°C) and low (winter period, –12.2°C) temperatures. For the temperature of 9.6°C, the lowest value of ΔT_a was observed, which is quite logical, since soil temperature at the depth of over 12.3 m is T_m = 9.79°C.

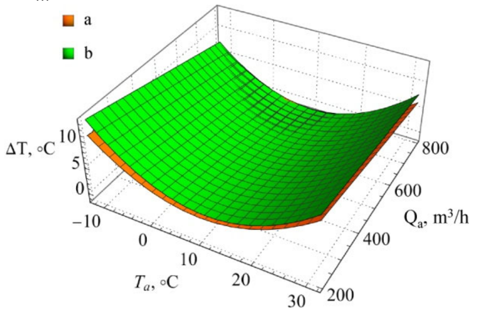

Fig. 11. Graphs of dependence between the change in air flow temperature ΔT_a and research factors for concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers

Statistical analysis of equations (25) and (26) in studied variation range showed that Pearson correlation coefficient is 0.94. Fisher's test is also F = 2.12 < F_t = 2.49. This confirms the adequacy of the model obtained.

As mentioned before, during air flow's passage through the ground heat exchanger, it interacts with the air duct's walls, this causing pneumatic losses. Therefore, ground heat exchanger's effective heat capacity was chosen as the optimization criterion, which can be calculated as follows:

(27) $$N_E = N_T - N_{ST}$$

where N_T is the ground heat exchanger's heat capacity, W and N_{ST} is the power required to pump air through the ground heat exchanger, W.

The ground heat exchanger's heat capacity can be determined as follows:

(28) $$N_T = \frac{Q_{in}}{3600}\rho_a(T_{in})c_a\Delta T_a$$

where Q_{in} – the volumetric air injection, m³/h; $\rho_a(T_{in}) = \frac{273}{273+T_{in}}$ – air density at the heat exchanger's outlet, kg/m ³ ; c_n – specific heat capacity of air assumed as c_n = 1003.62 J/(kg·°C); ΔT_a being the temperature difference at the heat exchanger's inlet and outlet, °C.

By combining the equations and using the Wolfram Cloud software package, we obtain the dependencies between the effective heat capacity of ground heat exchangers N_E and incoming air temperature T_i, as well as air consumption Q_{in}, as shown in Fig. 12.

Taking into account the condition of maximization of ground heat exchangers' effective heat capacity N_E, defined were rational air consumption values Q_{in} = 453.8 m³/hour concentric-type heat exchanger and Q_{in} = 455.2 m³/hour for U-shaped heat exchanger.

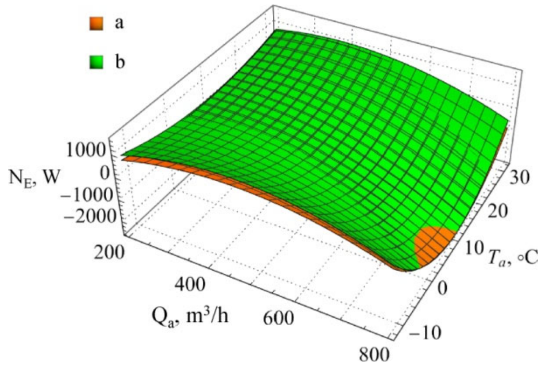

Fig. 12. Graphs of dependence between changes in ground heat exchangers' effective heat capacity N_E and research factors for concentric-type (a) and U-shaped (b) vertical-type ground heat exchangers

Thus, the effective heat capacity of concentric-type heat exchanger is N_{EC} (T_{in} = 31.7 °C) = 1266 W, N_{EU} (T_{in} = –12.2 °C) = 1052 W, which is less than effective heat capacity of U-shaped heat exchanger N_{EU} ($_{in}$ = 31.7°C) = 1575 W, N_{EU} (T_{in} = –12.2°C) = 1235 W. That is, U-shaped vertical-type ground heat exchangers are 17–24% more efficient than concentric-type ones.

Results of experimental studies of air flow heating process in vertical-type ground heat exchangers

At the fourth stage, provision is made for experimental studies of air flow heating process in vertical-type ground heat exchangers in production conditions at a pig farm of Agrofirma Napadivs'ka (Napadivka village, Vinnytsia district).

To raise the efficiency of clean air injection system's operation, a U-shaped vertical-type ground heat exchanger was installed behind the pig fattening premises. The ground heat exchanger's outlet pipe is connected to one of the clean air injection system's lines at the depth of 1 m from the soil surface. The well's total length is 18 m. At the depth of 3 m, the duct was thermally insulated. The last 3 m are made to collect condensed water. The structural-and-technological diagram and the general appearance of the experimental installation are shown in fig. 13.

Cimatic parameters of the research were determined based on the data of temperature and air humidity measurements at the beginning of each series of measurements and entered into Excel data table.

According to the results of numerical modeling of the air flow heating process in a vertical-type ground heat exchanger, it can be concluded that its nature depends on a number of factors. Accordingly, during experimental studies of this process, one should proceed from the technological possibilities of parameter change, and this requires conducting a large number of experiments. To reduce the number of experiments while preserving the reliability of technological process-related information, some methods of the experiment planning theory were used.

The process of heat extraction from the soil strata is influenced by: soil type and moisture, the operating time of geothermal ventilation system in a particular mode (supply air heating or cooling), ambient air temperature, volumetric air injection, diameter, length, number, interaxial distance and location of ground heat exchangers.

The soil type is low-humus black soil, on which the pig farm is located.

The dimensions were selected based on theoretical calculations.

DUNDAR CT 16.4 centrifugal fan was used as a power air plant (with the maximum air flow of 850 m³/h). The research was performed by varying the values of the following factors:

– the air flow at three levels: minimum (200 m³/h), average (500 m³/h) and maximum (800 m³/h), each of them to be determined after calibration by the electric fan motor's rotation speed and regulated by frequency converter FC 51 of VLT Micro series (current frequency 10; 30; 50 Hz) subject to a stable operation at specified frequency; air flow was measured using Solomat MPM 500E multifunctional device;

–air temperature in the ground heat exchanger was measured and recorded 6 times a day.

– Central air pumping duct air at the pig fattening farm was laid at the construction stage already, representing a tray of utility networks L 1-8/2 that passed under the pigs' room. Air injection nozzles that came out of the central air duct were polypropylene pipes.

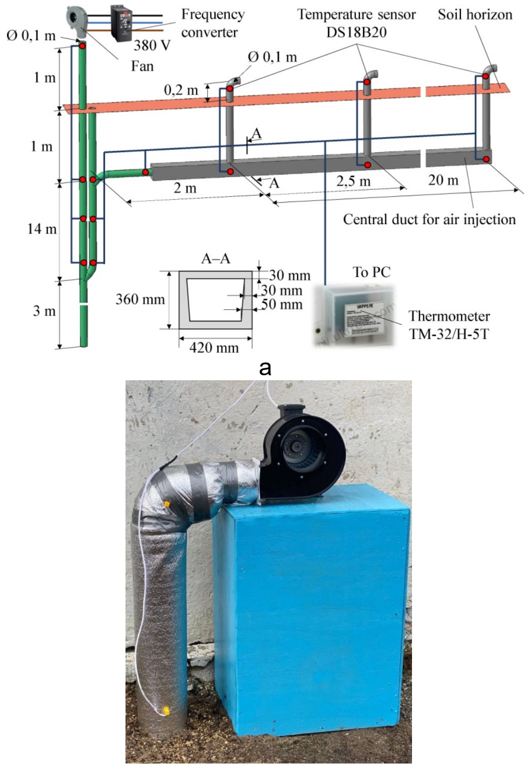

Fig. 13. Diagram (a) and general view (b) of the ground heat exchanger

DS18B20 digital temperature sensors are placed throughout the air injection system, which sensors are connected to «Termometr TM-32/N-5T» data recording system. The distance between the sensors in the ground heat exchanger was 3 m along the entire air duct.

The research was conducted in the period from 02.01.2021 to 02.01.2022.

During the research, the dynamics of temperature obtained from each sensor in the entire research period was recorded.

The centrifugal fan motor's power was determined by the frequency converter [36-37].

For typically low (in winter), high (in summer) and medium (in autumn and spring) temperatures T_{in}, air flow injection Q_{in} was varied.

The change in air flow temperature ΔT_a at the inlet and outlet of the ground heat exchanger was chosen as the research criterion. To optimize the parameters of the ground heat exchanger, heat capacity N_E criterion was used.

Figure 14 shows the dynamics of temperatures in the ground heat exchanger at different distances from duct L_a . The absolute value of the temperature difference at the ground heat exchanger's inlet and outlet Δ is also observed.

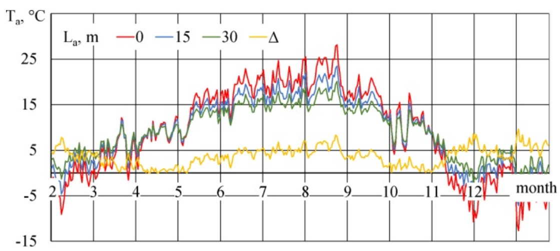

Fig. 14. Temperature dynamics in the ground heat exchanger at the duct's different distances

According to the research technique, air injection Q_{in} was varied at typically low (in winter), high (in summer) and medium (in autumn and spring) temperatures T_{in} . As such temperatures, the following ones were chosen: –12.2 °C, 9.8 °C and 28.1 °C. The graphs of temperature distribution in the heat exchanger's air duct for pumping Q_{in} = 500 m^3/h are shown in Fig. 15.

It can be seen from fig. 15 analysis that there is a temperature change trend along the entire air duct length, which trend is typical for numeric modeling results. That is why let us proceed to calculation of experimental regression equations and their comparison with theoretical ones.

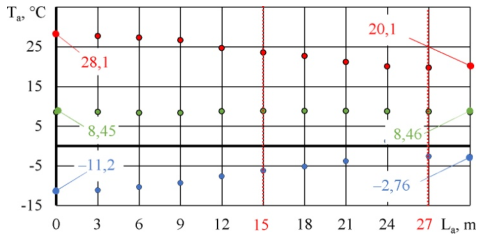

Fig. 15. Graph of experimental dependence between air flow temperature T_a and the path of its movement L_a in a U-shaped vertical-type ground heat exchanger with the air injection of 500 m^3/h

Upon their processing in Wolfram Cloud software package, the second-order regression equation was obtained, which shows the dependence between the change of air flow temperature ΔT_a in the U-shaped vertical-type ground heat exchanger and the research factors in the following form (Fig. 16):

$$\Delta T_{aU} = 8{,}77254 - 0{,}00956083\, Q_{in} - 0{,}322949\, T_{in} + 0{,}0000790378\, Q_{in} T_{in} + 0{,}0145349\, T_{in}^2. \qquad (29)$$

Visual analysis of fig. 16 testifies to the identity of theoretical and experimental dependencies. Pearson correlation coefficient equals to 0.94. Also, Fisher's criterion is $F = 2.02 < F_T = 2.49$. This confirms the adequacy of the model obtained. Therefore, theoretical dependencies can be used in further production calculations.

In connection with the similarity of theoretical and experimental regression equations of air flow temperature change ΔT_a in U-shaped vertical-type ground heat exchangers, let us conduct visual and statistical comparison of calculated thermal efficiency (Fig. 17).

Statistical comparison of experimental data with theoretical dependence according to Pearson's correlation coefficient – 0.95 and Fisher's test – $F = 1.93 < F_T = 2.98$ testifies to a high adequacy of theoretical dependence. Hence, in the future we are going to use obtained theoretical dependencies to calculate the effective heat capacity of the U-shaped vertical-type ground heat exchanger.

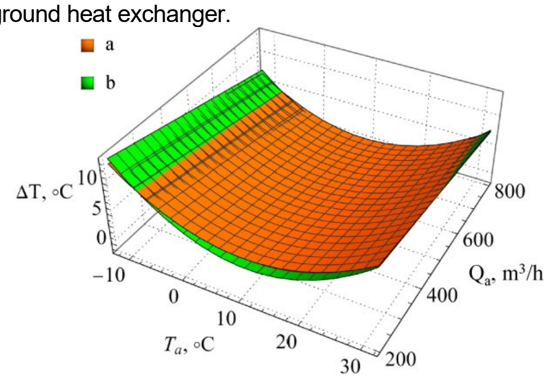

Fig. 16. Graphs of theoretical (a) and experimental (b) dependencies between air flow temperature change ΔT_a and research factors for U-shaped vertical-type ground heat exchangers

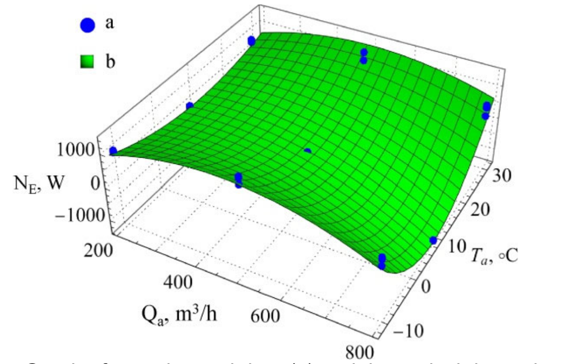

Fig. 17. Graph of experimental data (a) and theoretical dependence (b) between changes in the effective heat capacity of U-shaped vertical-type ground heat exchangers N_E and research factors

Conclusions

The results of analytical studies on the air losses of vertical ground heat exchangers for two proposed schemes (concentric and U-shaped) have determined that the power of the U-shaped heat exchanger is higher by 0.9–1.7%. Therefore, it can be assumed that the air losses of both variants of vertical ground heat exchangers are nearly identical. The dependence of the change in air loss power (NST) of the U-shaped vertical ground heat exchanger on its length (LU1), diameter (DU1), and air flow rate (qin) has been determined.

The physical-mathematical apparatus of the air heating process in a vertical ground heat exchanger has been generalized, based on equations of air flow continuity, Navier-Stokes equations, heat transfer equations, initial, and boundary conditions. An approximated function of ground temperature at time τ and depth z has been used as boundary conditions. Considering the approximated temperature data on the surface of the ground in the Vinnytsia region, the distribution of ground temperature with depth z has been determined.

Based on the results of numerical modeling of the air heating process in concentric and U-shaped vertical ground heat exchangers using the Simcenter Star-CCM+ software package, the distribution of the temperature field during the summer and winter periods was obtained. The temperature of the airflow during the summer (winter) period, which moves along a pipe with a total length of 32 m, decreases (increases) by 7.2°C (8.1°C) for the concentric heat exchanger and up to 8.5°C (9.3°C) for the U-shaped heat exchanger. The lowest (highest) temperature for the summer (winter) period is observed for the airflow path between 26.75–28.50 m. This indicates the necessity of thermal insulation for the duct at a depth of 1.50–3.25 m, ensuring the preservation of the lowest temperature until the exit from the heat exchanger.

Processing of the obtained data in the Simcenter Star-CCM+ software package and subsequent analysis in the Wolfram Cloud software package allowed obtaining second-order regression equations showing the dependencies of the change in airflow temperature ΔT_a and the effective thermal power N_E of ground heat exchangers on the temperature of incoming air T_{in} and airflow rates Q_{in} for each variant of the heat exchangers. Rational values of airflow rates were determined considering the condition of maximizing the effective thermal power of ground heat exchangers: Q_{in} = 453.8 m^3/h for the concentric heat exchanger and Q_{in} = 455.2 m^3/h for the U-shaped heat exchanger. Consequently, the effective thermal power of the concentric heat exchanger is $N_{EC}(T_{in} = 31.7°C) = 1266$ W, $N_{EU}(T_{in} = -12.2°C) = 1052$ W, which is lower than the effective thermal power of the U-shaped heat exchanger: $N_{EU}(T_{in} = 31.7°C) = 1575$ W, $N_{EU}(T_{in} = -12.2°C) = 1235$ W. Thus, the U-shaped vertical ground heat exchanger is 17–24% more efficient than the concentric one.

Experimental studies of the air heating process in the U-shaped vertical ground heat exchanger under industrial conditions determined the dynamics of temperature changes throughout the year. Regression equations of the second order describing the change in airflow temperature ΔT_a and effective thermal power N_E as functions of temperature T_{in} and airflow rates Q_{in} at the inlet of the U-shaped vertical ground heat exchanger were obtained. Statistical comparison of experimental data with theoretical dependencies, using the Pearson correlation coefficient of 0.95 and the Fisher criterion $F = 1.93 < F_T = 2.98$, indicates the high adequacy of the theoretical dependencies.

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