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JUSTIFICATION OF THE MECHATRONIC SYSTEM FOR PIGSTY MICROCLIMATE MAINTENANCE

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ABSTRACT. The main parameters of the microclimate of pig farms are regulated by the norms of technological design. Naturally, such micro-climate parameters at real energy prices require high costs, but these are the most favourable conditions for the life of suckling piglets. According to the presented analysis, the problem of research is the need to provide a micro-climate in the room for comfortable keeping of pigs, which is currently achieved through high energy. The purpose of the development is to increase the efficiency of the microclimate of pig farms by using a mechatronic control system, rational use of utilized air energy and soil thermal potential with reduced energy costs of the ventilation system. The problem is solved by creating such a mechatronic system to ensure the microclimate of pig farms, which can: switch the direction of airflow to ensure the operation of the system in winter and summer; to control the movement of air, which must be disposed of according to the parameters of its quality; to provide a local microclimate in each place where animals are kept; rational use of soil thermal energy as a source of alternative energy; to carry out automatic pumping of the water necessary for humidification of air, and its utilization. The article presents the results of research of parameters of the developed mechatronic system of providing microclimate of pig premises, which were carried out in three stages: research of heat exchanger of side-evaporating type; substantiation of the ventilation system of polluted air intake; substantiation of the ventilation system for injecting clean air taking into account it's geothermal heating/cooling. The advantage of the proposed mechatronic system of the microclimate of pig farms is that it allows increasing the efficiency of microclimate by rational use of energy of utilized air (due to the use of side-evaporator type heat exchanger based on Maisotsenko cycle) and soil heat potential (geothermal energy) at low operating costs of the ventilation system through the additional provision of mechatronic elements. The presented results of numerical simulation of the indirect evaporative heat exchanger allow us to state that the cooling effect obtained in indirect evaporative channels can be quite high in comparison with traditional air conditioning patterns. The presented heat exchanger based on the Maisotsenko cycle (M-cycle) of considered heat carrier flow scheme is characterized by its high cost-effectiveness, low specific cost, small operational costs and structural simplicity, which is confirmed in the works. The models obtained in the Star CCM +software package can be used for optimization analysis of air-cooling with variations in the Reynolds number, humidity, channel length and geometric dimensions of channels. Because of analytical investigations of the contaminated air intake ventilation system, the method was developed and on its basis – the algorithm was implemented for the determination of geometrical arrangement of holes in the air duct of the mechatronic system for pigsty microclimate maintenance.

Introduction

Pigsty microclimate is determined by the combination of air temperature, relative humidity, chemical and mechanical composition. Each of these parameters individually has a significant impact on livestock productivity and should be maintained within a strict framework conditioned by livestock physiological needs and capabilities (Van Wagenberg, 2005; Gunko *et al.*, 2021).

In livestock breeding, microclimate is primarily understood as the climate of livestock building, which is defined as the totality of the air environment's physical state, its gas, microbial and dust contamination, taking into account the state of the building itself and the process equipment. In other words, microclimate means meteorological conditions in closed livestock buildings, the concept of which includes temperature, humidity, chemical composition and air velocity, as well as dust content and illumination intensity. Optimal microclimate allows increasing livestock productivity, reducing the feed expenditures per production unit, contributes to better maintenance of livestock health. Microclimate in the building depends on local (zonal) climate and season, thermal and moisture resistance of buildings' enclosing structures, ventilation conditions, the intensity of the building' illumination and heating, sewerage conditions and manure removal quality, livestock management technology, species and age composition of livestock (Huynh *et al.*, 2005). Basic parameters of pigsty microclimate are regulated by technological designing standards (Godyń *et al.*, 2020). The room temperature for breeding boars should be indoors – from 13 °C to 18 °C, for pregnant sows – from 13 °C to 18 °C, for suckling – from 18 °C to 22 °C, for repair young stock – from 18 °C to 22 °C, for weaned piglets up to 30 days – from 24 °C to 30 °C, in 60 days – 22 °C, for piglets on rearing – from 15 °C to 20 °C, for fattening young animals depending on age – from 12 °C to 20 °C. At local heating of piglets in the first week of life the temperature in a lair should be 30 °C, in the second – 28 °C, in the third – 26 °C, in the fourth – 24 °C, in the fifth – 24 °C. Naturally, such microclimate parameters at real energy prices require high costs, but these are the most favorable conditions for the life of suckling piglets.

At below-critical temperatures, the body has no time to produce heat, based on forage energy, so hypothermia occurs followed by possible colds and even death. At above-critical temperature, convective heat exchange between the body and the environment decreases sharply, so the risk of overheating and heatstroke arises. When temperature conditions are violated (hypothermia, overheating), natural resistance decrease, as well as pulmonary and gastrointestinal diseases are observed (Godyń *et al.*, 2020). However, sharp temperature fluctuations during the day produce a stronger negative impact on the body than permanently elevated or reduced temperatures, and this primarily affects the young stock. Protective humoral factors in first day's livestock are poorly developed, their skin and mucous membranes being very sensitive to pathogenic microbes (Renaudeau *et al.*, 2010).

Ambient humidity also significantly affects livestock body thermoregulation, and particularly its heat elimination, and high relative humidity (85% and above) produces a negative effect on the body and heat elimination at both high and low ambient temperatures (Johnston *et al.*, 2013). High humidity suppresses metabolism and redox processes in the body and reduces pigs' resistance. When livestock is kept in the building with high humidity during the cold season, such diseases as bronchitis, pneumonia, gastrointestinal diseases are often observed in young livestock. High humidity contributes to the preservation of microorganisms, particularly pathogenic and fungal microflora, in the building, this frequently giving rise to skin diseases – ringworm, eczema, scabies and others. In addition, high humidity and low temperature increases the consumption of feed per unit of output, livestock appetite is deteriorating (Kozlovski, 1984).

Air velocity ensures air exchange in the building, enhancing the cooling capacity of the air. Therefore, low air velocities lead to microclimate deterioration, with high ones being able to cause colds at low temperatures (Il'in *et al.*, 2011). Increasing air velocity along animals' body surfaces reduces the "perceived body temperature" – that is, the temperature the animals feel. When using this method, one must monitor the cooling effect, as there is the risk of hypothermia in livestock. Increasing air velocity reduces body temperature, so even during the warm season, one needs to ensure that livestock does not over cool. Normally, excessive air velocity causes drafts. Hence, the evaluation of the cooling effect depending on air velocity is as follows. For 0.2 m s^{-1} air velocity along the animal body, the perceived temperature decrease is 4 °C, for that of 0.5 m s^{-1} – 7 °C, and 1.5 m s^{-1} – 10 °C (Forcada, Abecia, 2019; Song *et al.*, 2013).

In its turn, optimal microclimate maintenance in pigsties is associated with significant heat and electricity consumption, which makes up to 15% of producers' costs. During the heating period, pigsties heat-generating devices for various purposes consume up to 90% of total fuel and energy costs, while in summer air coolers consume up to 50% thereof. In addition, even a partial reduction of these costs will result in a significant reduction in energy-related production expenditures, thus reducing its self-cost. Ever-increasing energy cost complicates the situation and exacerbates the issue of energy-saving technologies implementation, as well as actualizes the economic problem of reducing the specific energy consumption for the manufacture of livestock products (Kaletnik *et al.*, 2020; Braun *et al.*, 2020).

Existing equipment meant for pigsty microclimate maintenance either entirely meets the requirements outlined in the Table 1 while consuming large amounts of energy, or is cost-saving but unable to maintain optimal microclimate parameters (Ivanov, Novikov, 2020). For example, disadvantages of known equipment (Patent UA 129759, 2018) may include the non-use of utilized air's thermal energy and lack of automatic switching of the microclimate system for summer (supply air-cooling) and winter (supply air heating) periods. This leads to an increase in energy consumption for pigsty microclimate maintenance. Except when using this equipment (Patent UA 102567, 2009), utilized air thermal energy is used inexpediently, while this can be used to cool the supplied air in summer and heat the same in winter. Lack of airflow's local regulation does not allow creating an individual microclimate for different livestock groups. Disadvantages of (Patent UA 144887, 2020) and (Patent US 2017/0016645, 2017) equipment include the absence of the system for microclimate parameters' automatic regulation, insufficient degree of airflow cooling and heating, which can be compensated by the use of alternative energy sources. Each of these units operates exclusively to cool or heat the supply airflow, *i.e.* no provision is made for switching between operating modes. In addition, these units' structural and process flow diagrams provide for no automatic pumping of water required for air humidification and its disposal.

According to the presented analysis, the problem of research is the need to provide a microclimate in the room for comfortable keeping of pigs, which is currently achieved through high energy consumption therefore, the development project is aimed at increasing the efficiency of pigsties microclimate by using a mechatronic control system, expedient use of utilized air energy and thermal soil potential (geothermal energy) with reduced energy costs for ventilation system functioning.

The designated problem was solved by the generation of such mechatronic system for pigsty microclimate maintenance, which provides for the opportunity to:

- switch between airflow directions to ensure the system's operation in winter (0 °C) and summer (25 °C) periods;
- control air movement, which must be disposed of according to its quality parameters (hydrogen sulfide, carbon dioxide and ammonia content);
- maintain local microclimate in each stall, where livestock are kept;
- expediently to use thermal soil energy as an alternative energy source;
- carry out automatic pumping of water required for air humidification and its disposal.

Methods

To solve the designated problem, a structural and process flow diagram of the mechatronic system for pigsty microclimate maintenance has been developed (Fig. 1).

The mechatronic system for pigsty microclimate maintenance operates as follows. The operator on control unit (5) sets given ranges of local microclimate parameters (temperature, humidity, airflow rate) for each stall, where livestock are kept. The operator of control unit (5) also sets air quality limits in terms of hydrogen sulfide, carbon dioxide and ammonia content. Next goes the launch of the mechatronic system for pigsty microclimate maintenance. Information on temperature, humidity and air quality values (hydrogen sulfide, carbon dioxide and ammonia) from temperature, humidity and air quality sensors (9), temperature and humidity sensors (14), external temperature and humidity sensors (36) is transmitted to control unit (5) via electrical wire devices (10). These data are compared between each other and with the data set by the operator.

The operator must set the temperature in the pigsty higher than the temperature on the outside of the pigsty (winter period), the control unit transmits a signal to rotating servo-operated rotating valve (18), with the valve having been set to the position that allows connecting contaminated air intake ventilation system (1) with inner working air pipe (20) of the indirect evaporative heat exchanger (3). In its turn, clean air charging ventilation system (2) is connected to inner recycled air pipe (22) of the indirect evaporative heat exchanger (3). Control unit (5) starts blower fan (24) in the direction of air supply from the pigsty to its outer side, and exhaust fan (25) in the opposite direction of air supply from the outside of the pigsty.

Depending on the air quality above the place where the pigs are kept, as determined using temperature, humidity and air quality sensors (9) and the limit values set by the operator, control unit (5) via electrical wires transmits the signal to servo-operated intake valves (8). Measured air quality parameters should be less than the limit values set by the operator, servo-operated intake valve (8) is closed. Otherwise, the servo-operated intake valve (8) opens at an angle that is directly proportional to the corresponding difference between air quality parameters and limit parameters. Air is sucked into air intake nozzles (7) and formed into the stream that moves along central air intake duct (6) of contaminated air intake system (1). Next, the airflow enters nozzles (16) and central cavity (17) of four-way valve (4). After that, the airflow enters inner working air pipe (20) and then to cross-channel set (23) of the indirect evaporative heat exchanger (3). This airflow passes through working channels (26), where it is cooled and its humidity is reduced along with condensate generation. This process occurs due to heat exchange through the walls connecting working channels (26) and wet channels (27). Next, cooling airflow is supplied to the pigsties outside through the outer working air pipe (19).

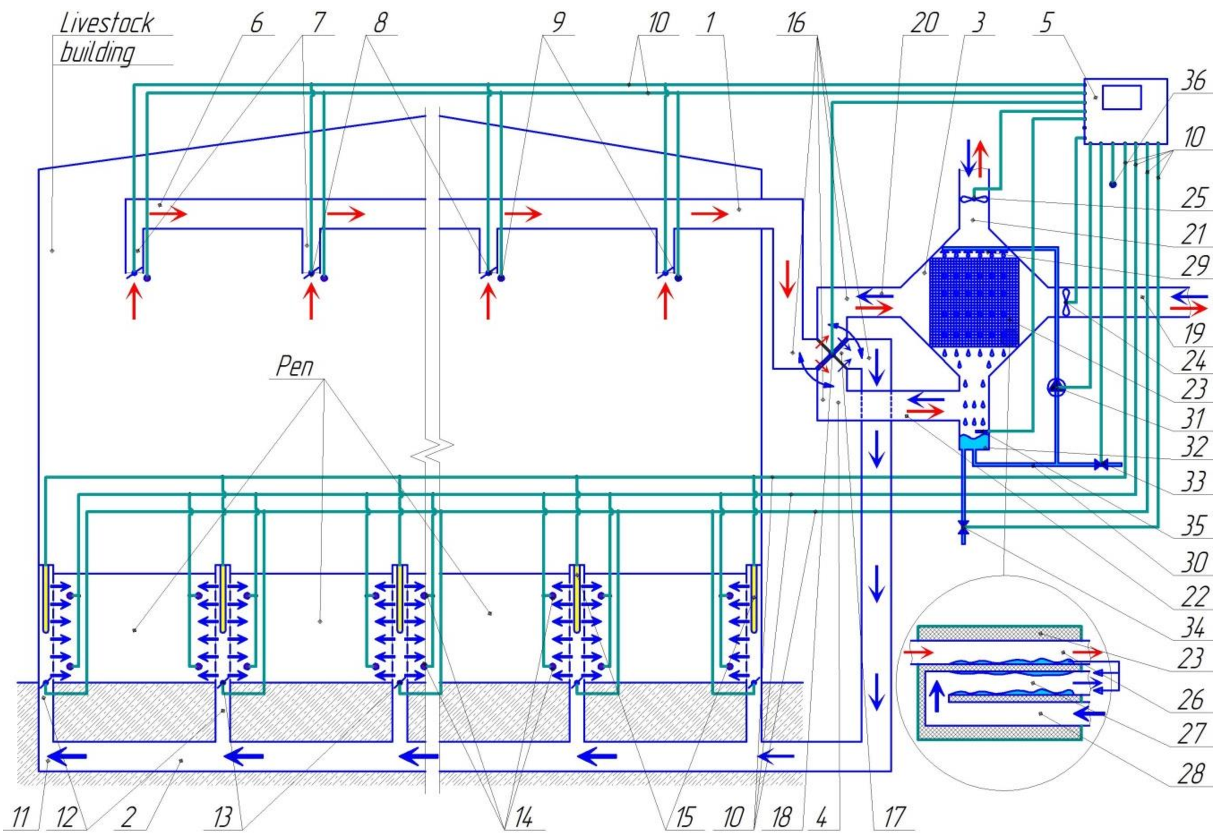

Figure 1. Structural and process flow diagram of the mechatronic system for pigsty microclimate maintenance: 1 – contaminated air intake ventilation system; 2 – clean air charging ventilation system; 3 – indirect evaporative heat exchanger; 4 – four-way valve; 5 – control unit; 6 – central air intake duct; 7 – air intake nozzles; 8 – servo-operated intake valves; 9 – temperature, humidity and air quality sensors; 10 – electrical wires; 11 – central air injection duct; 12 – branch air injection pipes; 13 – servo-operated discharge valves; 14 – temperature and humidity sensors; 15 – heating elements; 16 – nozzles; 17 – central cavity; 18 – servo-operated rotating valve; 19 – outer working air pipe; 20 – inner working air pipe; 21 – outer recycled air pipe; 22 – inner recycled air pipe; 23 – cross-channel set; 24 – blower fan; 25 – exhaust fan; 26 – working channels; 27 – wet channels; 28 – dry channels; 29 – water injection nozzles; 30 – pipeline system; 31 – water pump; 32 – water intake tank; 33 – electromagnetic water top-up valve; 34 – electromagnetic water draining valve; 35 – level sensor; 36 – external temperature and humidity sensors

Simultaneously with the foregoing, cold and dry airflow from the pigsties outside enters the outer recycled air pipe (21). Next, cold and dry air flow moves to cross-channel set (23) of the indirect evaporative heat exchanger (3), where it continues moving first along dry channel (28) and then along wet channel (27). In dry channel (28), cold and dry airflow is heated due to heat exchange through the walls connecting wet channels (27) and dry channels (28), and in wet channel (27) already warm and dry airflow is enriched with moisture, continuing to be heated.

Under the action of gravity, excess moisture flows to the bottom of water intake tank (32). Initially, control unit (5) via electrical wires closes electromagnetic water draining valve (34). In addition, the control unit (5) turns on water pump (31), which supplies collected water from water intake tank (32) and directs the same through pipeline system (30) to water injection nozzles (29). Water from water injection nozzles (29) washes wet channels (27). The water level in water intake tank (32) is determined using level sensor (35), which via electrical wires (10) transmits this information to control unit (5). The water level should be lower than the intended one, the control unit (5) opens electromagnetic water top-up valve (33), and water from the pigsty's water consumption system enters pipeline system (30). Otherwise, the control unit (5) closes electromagnetic water top-up valve (33) and opens electromagnetic water draining valve (34), and the water from water intake tank (32) enters the manure removal system.

After wet channels (27), warm and wet airflow enters inner recycled air pipe (22), which is connected to nozzles (16) and central cavity (17) of four-way valve (4). Next, warm and wet airflow enters central air injection duct (11) where through branch air injection pipes (12) enters the pigsty's middle part directly into the stalls, where livestock is kept. Since central air injection duct (11) is located under the floor below the soil freezing level, the process of geothermal heating of warm and wet airflow additionally occurs.

Information from temperature and humidity sensors (14) via electrical wires (10) is supplied to control unit (5), where it is compared with local microclimate parameters set by the operator for each stall. Where additional airflow heating is required in some particular stall, control unit (5) switches on respective heating element (15) and opens servo-operated discharge valve (13) entirely. Air temperature should be above or equal to the one required, control unit (5) switches off respective heating element (15) and closes servo-operated discharge valve (13) partially. The servo-operated discharge valve (13) closure degree is directly proportional to the difference between the pre-set and the measured temperature.

Let us consider the case when the temperature pre-set by the operator is lower than the temperature on the pigsty's outside (summer period). The control unit transmits the signal to servo-operated rotating valve (18), which is set to the position that allows connecting contaminated air intake ventilation system (1) with inner recycled air pipe (22) of the indirect evaporative heat exchanger (3). In its turn, clean air charging ventilation system (2) is connected to inner working air pipe (20) of the indirect evaporative heat exchanger (3). Control unit (5) starts blower fan (24) in the direction of air supply from the pigsty's outside to the pigsty's middle part, while exhaust fan (25) does the same in the opposite direction of air supply from the pigsty to its outer side.

Depending on air quality above the stalls, as determined using temperature, humidity and air quality sensors (9) and limit values set by the operator, control unit (5) via electrical wires transmits the signal to servo-operated intake valves (8). Measured air quality parameters should be lower than limit values set by the operator, servo-operated intake valve (8) is closed. Otherwise, the servo-operated intake valve (8) opens at an angle that is directly proportional to the respective difference between air quality values and limit values. Air is sucked into air intake nozzles (7), being formed into the stream that moves along central air intake duct (6) of contaminated air intake system (1). Further, airflow enters nozzles (16) and central cavity (17) of four-way valve (4). After that, airflow enters inner recycled air pipe (22) and then cross-channel set (23) of the indirect evaporative heat exchanger (3). This airflow passes through dry channel (28) and wet channel (27), where it is heated using heat exchange through the walls adjacent to working channel (26). Further, airflow is supplied to the pigsties outside through outer recycled air pipe (21).

Simultaneously with the foregoing, warm and dry airflow from the pigsties outside enters the outer working air pipe (19). Next, warm and dry air flow moves to cross-channel set (23) of indirect evaporative heat exchanger (3), where it continues to flow through working channels (26), where it is cooled and its humidity is reduced along with condensate generation. This process occurs due to heat exchange through the walls that connect working channels (26) and wet channels (27).

Under the action of gravity, excess moisture flows to the bottom of water intake tank (32). Initially, control unit (5) via electrical wires closes electromagnetic water draining valve (34). In addition, control unit (5) turns on water pump (31), which supplies collected water from water intake tank (32) and directs the same through pipeline system (30) to water injection nozzles (29). Water from water injection nozzles (29) washes wet channels (27). The water level in water intake tank (32) is determined using level sensor (35), which via electrical wires (10) transmits this information to control unit (5). The water level should be lower than the intended one, the control unit (5) opens electromagnetic water top-up valve (33), and water from the pigsty's water consumption system enters pipeline system (30). Otherwise, control unit (5) closes electromagnetic water top-up valve (33) and opens electromagnetic water draining valve (34), and the water from water intake tank (32) enters the manure removal system.

After wet channels (27), warm and wet airflow enters inner working air pipe (20), which is connected to nozzles (16) and central cavity (17) of four-way valve (4). Next, warm and wet airflow enters central air injection duct (11) where through branch air injection pipes (12) enters the pigsty's middle part directly into the stalls, where livestock are kept. Since central air injection duct (11) is located under the floor below the soil freezing level, the process of geothermal heating of warm and wet airflow additionally occurs.

Information from temperature and humidity sensors (14) via electrical wires (10) is supplied to control unit (5), where it is compared with local microclimate parameters set by the operator for each stall. Where additional airflow heating is required in some particular stall, control unit (5) switches on respective heating element (15) and opens servo-operated discharge valve (13) entirely. Air temperature should be above or equal to the one required, control unit 5 switches off respective heating element (15) and closes servo-operated discharge valve (13) partially. The servo-operated discharge valve (13) closure degree is directly proportional to the difference between the pre-set and the measured temperature.

Investigation of parameters of the developed mechatronic system for pigsty microclimate maintenance was carried out in three stages:

- investigation of the indirect evaporative heat exchanger;
- substantiation of the contaminated air intake ventilation system;
- substantiation of clean air charging ventilation system taking into account its geothermal heating/cooling.

The first investigation stage was intended for determining the feasibility of using an indirect evaporative heat exchanger based on the Maisotsenko cycle (M-cycle) in the proposed mechatronic system. Heat and mass transfer processes taking place in such heat exchangers are close to thermodynamically inverse processes, thus allowing obtaining maximum air cooling effect with minimum energy consumption. The dew point temperature is the theoretical limit of wet air-cooling in such a device. The invariability of airflow concentration also is a positive effect of cooling in the working channel. Studies of this heat exchanger were conducted based on numerical simulations in the Star

CCM+ software package (Aliev *et al.*, 2018; Honcharuk *et al.*, 2021). The diagram for simulating the channels of the indirect evaporative heat exchanger is shown in Fig. 2.

The second investigation stage, and namely the substantiation of the contaminated air intake ventilation system, was carried out based on analytical calculations. The model for determining the geometrical arrangement of holes in the contaminated air intake ventilation system is shown in Fig. 3.

As the abscissa axis, the air duct axis with coordinate origin in the centre of its end section was selected. Air duct length is L, along which there are n holes of the same plane σ. The airflow rate at the duct's beginning is v_n. It is necessary to establish how the distance between the holes varies along the air duct length to ensure air's uniform distribution between the holes.

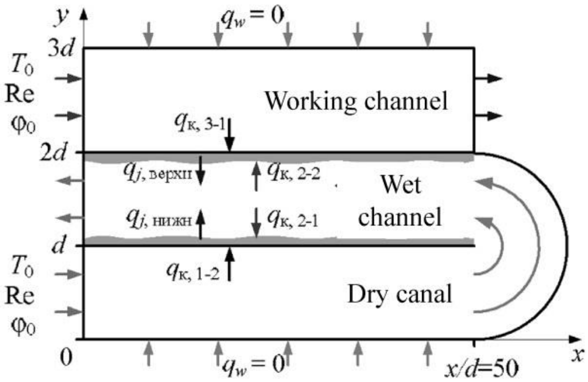

Figure 2. Diagram for simulating the channels of the indirect evaporative heat exchanger

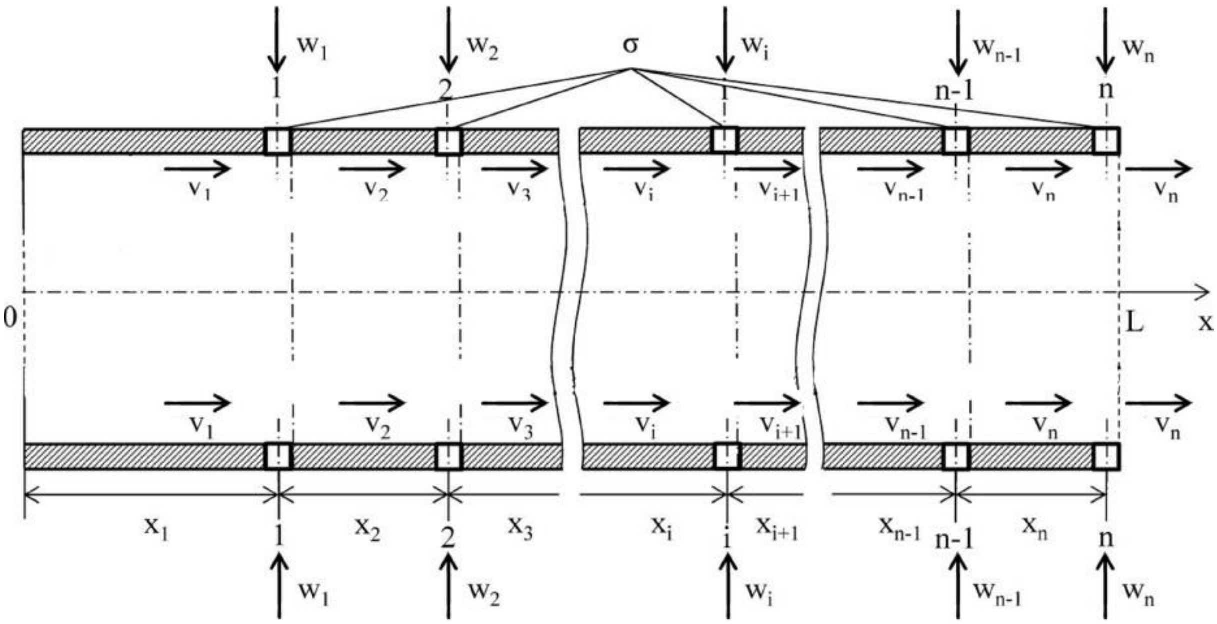

Figure 3. A computational model for determining the geometrical arrangement of holes in the contaminated air intake ventilation system

To determine distance x_i, velocity w_i, hole area σ and the number of holes n, let us make the method (Fig. 4) consisting of the following steps (Yaropud, Aliyev, 2015, Solona, Kupchuk, 2020).

1. The setting of parameters: air duct length L, hole discharge ratio φ, shock smoothing ratio α, friction resistance ratio κ, air duct effective diameter d_e, airflow rate at air duct beginning v_n, air duct's cross-sectional area A, the rate of the airflow passing through hole w_1, the airflow rate at air duct's cross-section v_1.
2. Setting the hole area variation increment $\sigma = 0.001 \cdot j$, where j is the hole number.
3. Calculation of distance x_i using the formula where i is the hole number.

$$x_i = \frac{L\varphi^2\sigma^2 v_i(\kappa x_{i-1} + d_e)}{v_n\left(\varphi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha\alpha_e) - A^2 d_e\right)} - \frac{\sqrt{L^2\varphi^4\sigma^4 v_i^2(\kappa x_{i-1} + d_e)^2 - \left(L^2\kappa\varphi^2\sigma^2 v_i^2 x_{i-1} + A^2 v_n^2 d_e x_{i-1}^2\right)\left(\varphi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha\alpha_e) - A^2 d_e\right)}}{v_n\left(\varphi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha\alpha_e) - A^2 d_e\right)} \quad (1)$$

4. Calculation of distance w_i using the formula

$$w_i = \frac{x_i \cdot A \cdot v_n}{\sigma \cdot L} \quad (2)$$

5. Calculation of the sum of distances

$$L_{calc} = \sum_{i=1}^{n} x_i \quad (3)$$

6. Meeting the condition: if the sum of distances $L_{calc} > L$, then clause 7 actions of is performed, otherwise clause 3 is fulfilled.
7. Determining the number of holes $n = i$.
8. To ensure required convergence of the air duct's total length, we fulfil the condition: if the modulus is the difference between the sum of distances and accepted air duct length $| L_{calc} – L | <0.01$, clause 9 action is performed, otherwise clause 2 is fulfilled.
9. Determination of hole area $\sigma_{calc} = \sigma$.
10. Determination of n, $_{\sigma calc}$, $_{xi}$, $_{wi}$ parameters.

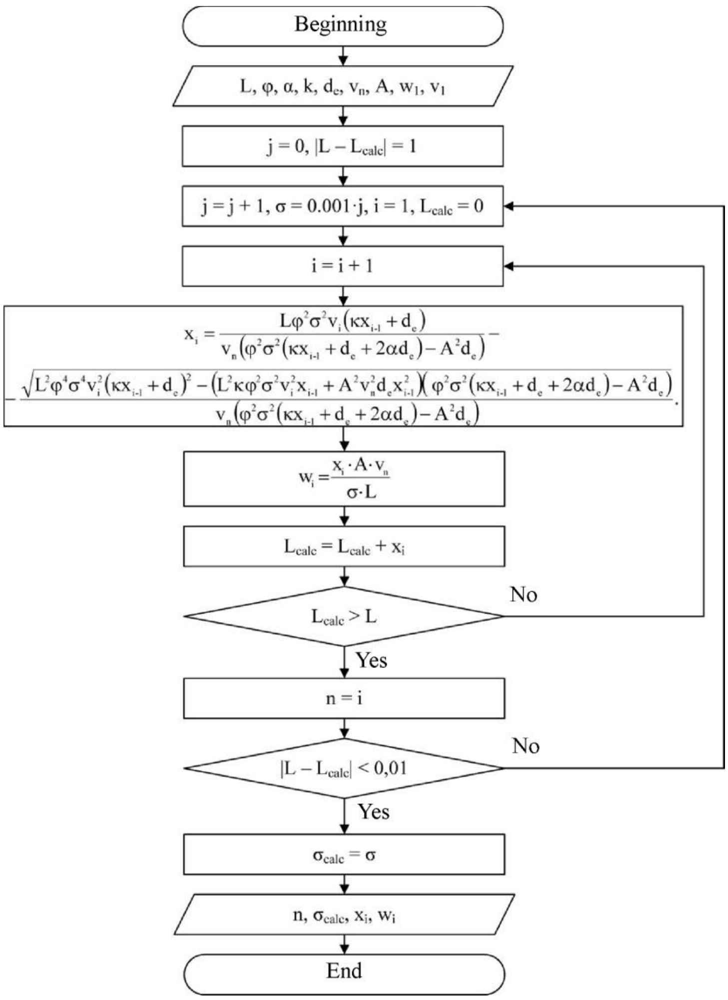

Figure 4. Algorithm for calculating the geometrical arrangement of holes in the microclimate system's air duct

The method so developed and the algorithm implemented on its basis (Fig. 4) are implemented in the Mathematica software package.

The third stage of investigations, and namely the substantiation of clean air charging ventilation system taking into account its geothermal heating/cooling, was conducted based on analytical studies in the Mathematica software package (Aliev *et al.*, 2018). To develop the mathematical model for heat elimination process in clean air charging ventilation system taking into account it's geothermal heating/cooling, we make the following assumptions: heat elimination process through air duct walls takes place in their thickness only; the process of heat elimination through air duct walls is instantaneous; due to a slight change in airflow pressure (Δp = 10–200 Pa) during its movement along the air duct, the system's thermodynamic process is considered isobaric; airflow in air ducts is homogeneous and isotropic; the air duct is at the depth of 3–5 m from the surface (soil temperature ranging from 7 °C to 13 °C depending on the season). To investigate the heat elimination process, the computational model was generated, which is shown in Fig. 5.

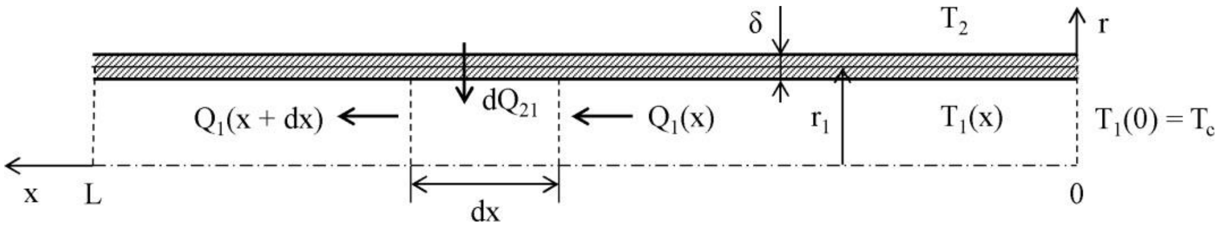

Figure 5. A computational model for heat elimination process in clean air charging ventilation system taking into account it's geothermal heating/cooling

According to Newton-Richman's law (Schmidt, 2019) and the equation for heat elimination through a cylindrical wall (Kreith *et al.*, 2010) we obtain a differential equation for heat elimination process within clean air charging ventilation system taking into account its geothermal heating/cooling:

$$\dot{m}_1 C_p dT_1(x) - \pi K_1 (T_2 - T_1(x)) dx = 0, \qquad (4)$$

where x is the coordinate, m; $\dot{m}_1 = V_1 \cdot \rho(T_1)$ – mass air consumption in the air duct, kg s 3^s^{-1}; $V_1 = v_1 A_1$ – volumetric air consumption in the i-th air duct, m^3 s^{-1}; v_i – air velocity in the air duct, m s^{-1}; $A_1 = \pi \cdot r_1^2$ – air duct's cross-sectional area, m^2; r_1 – air duct radius, m; C_p – specific air heat; T_1 – air flow temperature in the air duct, K; T_2 – soil temperature, K; $K_1 = ((\alpha_1(2r_1 - \delta))^{-1} + (\alpha_1(2r_1 + \delta))^{-1} + \lg((2r_1 + \delta)/(2r_1 - \delta))/(2\lambda))^{-1}$ – linear heat elimination ratio for a cylindrical air duct can be calculated using the formula; $\alpha_1 = \lambda \cdot Nu_1/d_1$ – heat elimination ratio in the air duct, W (m^2·K)$^{-1}$; δ is the air duct wall thickness, m; λ is the air duct wall specific thermal conductivity, for polyethene; $Nu_1 = 0.018\, Re_1^{0.8}$ – Nusselt number for air flow in the i-th air duct; $d_1 = 2r_1$ – air duct diameter, m; $Re_1 = d_1 v_1 \rho(T_1)/\mu$ – Reynolds number for air flow in the air duct; μ – dynamic air viscosity; v_i – air velocity in the i-th air duct, m s^{-1}; $\rho(T_1) = 273\rho_0/T_1$ – air density in the i-th air duct at constant pressure, which is associated with its temperature, kg m$^{3\,-1}$; ρ_0. – air density under normal conditions ($T_{n.c.}$ = 273 K, P_0 = 101325 Pa), ρ_0 = 1.293 kg m$^{3\,-1}$; L is the air duct length, m. According to Fig. 5, boundary conditions are $T_1(0) = T_c$, where T_c – the temperature at air duct inlet, K.

Results

As a result of the first stage of investigations, and namely, the numerical simulation, obtained was the field of temperatures and mass concentrations in the channels of the indirect evaporative heat exchanger (with Reynolds number Re = 100, the temperature at the working channel inlet t_0 = 30 °C, initial humidity φ_0 = 30% shown in Fig. 6. It follows from the figure that the average air temperature in the working channel (under these conditions) is lower than the wet channel temperature (P-value for numerical simulation was 0.05). At the same time, the mass concentration of air in the wet channel increases. The data also indicate that a fairly large part of the heat exchange area in the wet channel is in the saturation state. Moreover, temperature change along the wet channel length is not monotonous, and at a certain distance from the inlet, its minimum is observed. Reduction of relative humidity value at the inlet to the heat exchanger will increase the intensity of water film evaporation in the wet channel, and therefore, will reduce temperatures parameters in the dry and the working channels.

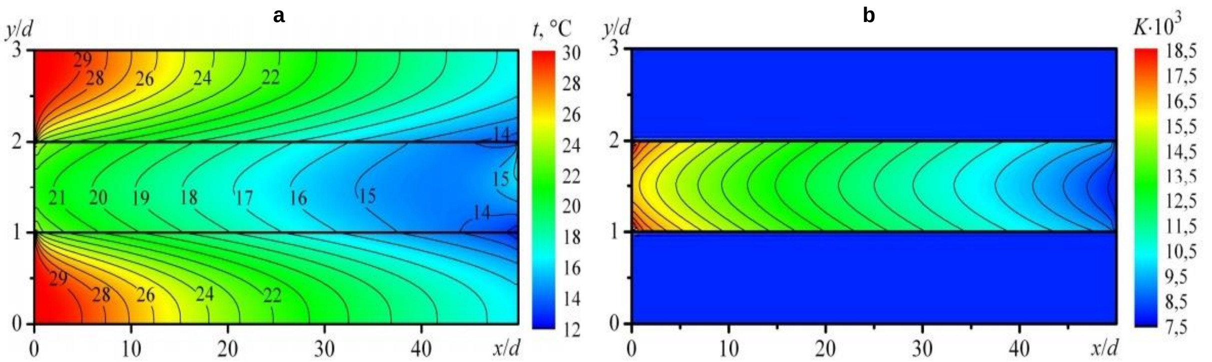

Figure 6. Fields of temperatures (a) and mass concentrations (b) in channels of the indirect evaporative heat exchanger (Re = 100, t_0 = 30 °C, ϕ_0 = 30%)

As a result of the second stage of investigations, and namely the substantiation of the contaminated air intake ventilation system, taking into account its structural and process parameters (L = 5.8 m; φ = 0.65; α= 0.4; κ= 0.01717 m; r_2 = 0.14 m; r_3 = 0.2 m; V_0 = 0.14 m^3 s^{-1}; x_1 = 0.9 m; v_1 = 0 m s^{-1}), determined was the number of holes n = 7 and their area σ = 0.011 m$_2$, as well as the distribution of distance between the holes according to Figure 7 and air velocities through the holes.

Analyzing Figure 7, one can state that the distance between the holes gradually decreases from 0.94 to 0.6 m in the direction opposite to the airflow. However, at the end of the duct, observed is a slight decrease in the distance by 0.04 m, which is caused by back airflow, which collides with the muffled end. A similar phenomenon is also observed with the distribution of air velocities through holes.

As a result of the third stage, and namely the solution of the differential equation (4) together with boundary conditions in Mathematica software package and assuming the structural and process parameters ($L = 20$ m; $r_1 = 0.25$ m; $V_1 = 0.14$ m^3 s^{-1}; $\delta = 0.0002$ m) of the clean air charging ventilation system air, we obtain the temperature distribution of air flows along the air duct length for different periods of the year (Fig. 8).

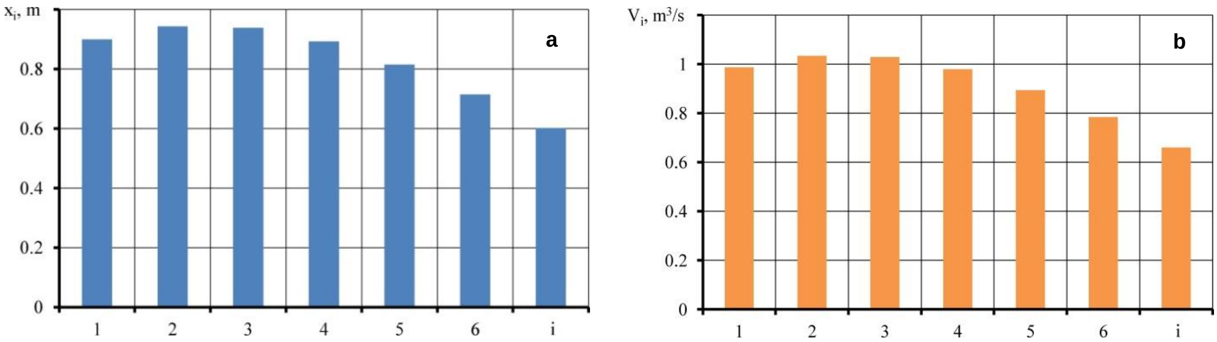

Figure 7. Distribution of distance between the holes (a) and air velocities through them (b) along the length of the air duct of the contaminated air intake ventilation system

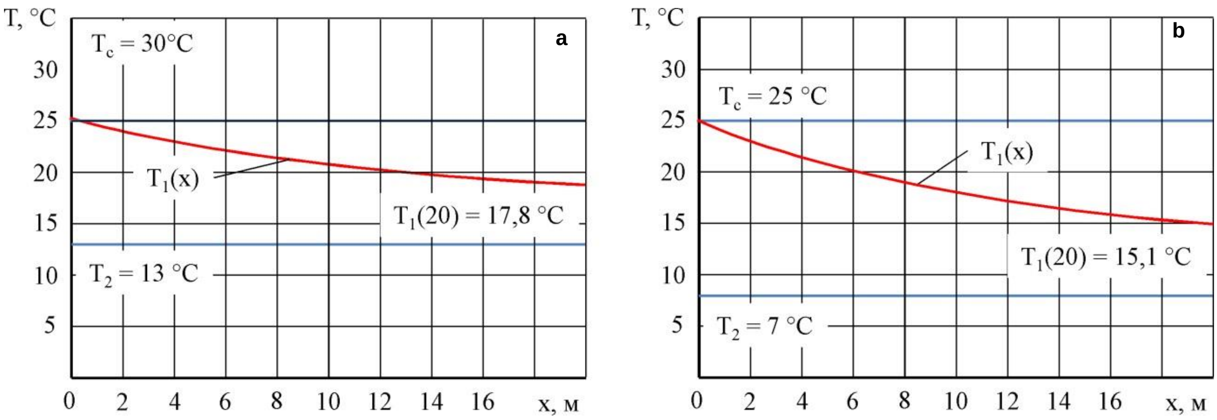

Figure 8. Distribution of air flow temperature in the air duct of the ventilation system for injecting clean air along its length in summer (a) and winter (b) periods of the year

As can be seen from Figure 8 (a), in the summer period (soil temperature – 13 °C) the airflow from the indirect evaporative heat exchanger (temperature – 25 °C), moving inside the air duct (length 20 m) of the clean air charging ventilation system, reduce its temperature down to 17.8 °C. In its turn, in the winter period (soil temperature – 8 °C) the airflow from the indirect evaporative heat exchanger (temperature – 25 °C), moving inside the air duct, reduces its temperature down to 15.1 °C.

Discussion

The presented results of numerical simulation of the indirect evaporative heat exchanger allow us to state that the cooling effect obtained in indirect evaporative channels can be quite high in comparison with traditional air conditioning patterns. The presented heat exchanger based on the Maisotsenko cycle (M-cycle) of considered heat carrier flow scheme is characterized by its high cost-effectiveness, low specific cost, small operational costs and structural simplicity, which is confirmed in the works (Mahmood *et al.*, 2016). The models obtained in the Star CCM +software package can be used for optimization analysis of air-cooling with variations in the Reynolds number, humidity, channel length and geometric dimensions of channels. One can assume that similar trends will be observed at other temperature and air humidity parameters at the heat exchanger inlet, but this conclusion requires additional calculations. These calculations will allow you to build an automated control system of the heat exchanger side-evaporator type to control the humidity in the room depending on the temperature. Indoor humidity is very significant and must be constantly adjusted depending on the temperature. At low humidity, livestock tolerates high temperatures more easily. Pigs are most resistant to high humidity. At the temperature of 32 °C, pigs weighing 100 kg respond equally both to the humidity of 30% and to that of 90% (Zhizhka, Povod, 2019).

Because of analytical investigations of the contaminated air intake ventilation system, the method was developed and on its basis – the algorithm was implemented for the determination of geometrical arrangement of holes in the air duct of the mechatronic system for pigsty microclimate maintenance. It was established that the distance between the holes gradually decreases to a certain value in the direction opposite to airflow movement. However, a slight reduction in the distance at the air duct end is observed due to the backflow of air colliding with the muffled end. Previous experimental studies allowed concluding that generated algorithm for the calculation of geometrical arrangement of holes in the air duct of the mechatronic system for pigsty microclimate maintenance is adequate and may be used in engineering calculations, as evidenced by a high correlation between theoretical and experimental data $R = 0.92–0.98$.

Because of theoretical investigations of the clean air charging ventilation system taking into account its geothermal heating/cooling, developed was the mathematical model of the heat transfer process in the pipe heat exchanger, which allows determining the distribution of airflow temperature along its length. Further investigations will be aimed at optimizing the results of theoretical studies to determine the dependence between structural parameters of the clean air charging ventilation system (length, radius and air duct material) and volumetric consumption of the air passing through the same under the condition of maximum useful heat output.

Presented results of numerical simulation, analytical and theoretical investigations allow asserting the feasibility of using the indirect evaporative heat exchanger based on the Maisotsenko cycle, the contaminated air intake ventilation system and the clean air charging ventilation system taking into account its geothermal heating/cooling.

In addition, it is expedient to use the indirect evaporative heat exchanger based on the Maisotsenko cycle not only as a cooler but at the same time as a humidifier of airflows. It should also be noted that in the management of the above processes, only mechanical energy is consumed to drive fans to blow air through respective channels.

Conclusion

The advantage of the proposed mechatronic system for pigsty microclimate maintenance is that it allows increasing the efficiency of microclimate maintenance thanks to expedient use of recycled air energy (due to the use of indirect evaporative heat exchanger based on Maisotsenko cycle) and thermal potential of soil (geothermal energy) with the ventilation system's reduced energy costs thanks to the additional provision of mechatronic elements.

It was established as a result of analytical investigations of the expedient geometrical arrangement of holes in the contaminated air intake ventilation system that the option of hole arrangement (as obtained according to the calculation algorithm so developed) is the most efficient one, as it ensures a uniform airflow distribution along the air duct length.

Analysis of results of theoretical investigations of heat elimination process in the clean air charging ventilation system taking into account its geothermal heating/cooling has proved its feasibility and efficiency, which allows reducing the specific energy consumption for maintenance of the entire microclimate system.

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Conflict of interest

The author declares that there is no conflict of interest regarding the publication of this paper.

Author contributions

VY – study conception and design, drafting of the manuscript;
IG – critical revision and approval of the final manuscript;
EA – analysis and interpretation of data;
IK – acquisition of data, drafting of the manuscript.

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