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INVESTIGATION OF THE EFFECT OF STRUCTURAL PARAMETERS OF AN INDIRECT EVAPORATIVE HEAT EXCHANGER ON AIR COOLING EFFICIENCY

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The article is devoted to investigating the influence of the structural parameters of an indirect evaporative heat exchanger on the efficiency of air cooling in microclimate control systems for livestock buildings. The relevance of the study is driven by increasing thermal loads during the summer period, higher requirements for maintaining comfortable conditions for animals, and the need to reduce energy consumption associated with air conditioning and ventilation processes in the agro-industrial sector.

The feasibility of using indirect evaporative cooling systems as an energy-efficient alternative to conventional compressor-based refrigeration units is substantiated. Such systems allow a reduction in the temperature of supply air without increasing its moisture content. A laboratory-scale experimental prototype of an indirect evaporative heat exchanger with a cross-flow air movement configuration was designed and manufactured. Experimental investigations were carried out by varying the inlet air temperature, absolute humidity, and airflow rate, as well as for two structural configurations of the heat and mass exchanger, featuring identical and different interchannel opening areas.

Based on statistical processing of the experimental data, second-order regression equations were obtained for the outlet temperature of the primary air stream, the thermal efficiency coefficient, and the effective cooling capacity of the heat exchanger. The adequacy of the developed mathematical models was confirmed using the Cochran and Fisher criteria, as well as by high correlation coefficients between experimental and calculated results.

It was established that the use of different interchannel opening areas intensifies heat and mass transfer processes, reduces the outlet air temperature, and increases thermal efficiency and effective cooling capacity compared to the configuration with identical opening areas. Rational operating modes of the heat exchanger were determined in terms of minimizing outlet air temperature and maximizing energy efficiency. The obtained results can be applied in the design and optimization of microclimate systems for livestock buildings and in the development of automatic control algorithms for air cooling processes.

Key words: *indirect evaporative cooling, heat exchanger, microclimate, livestock buildings, air cooling, heat and mass transfer, thermal efficiency coefficient, effective cooling capacity, energy efficiency, ventilation, structural parameters, modeling, agro-industrial complex.*

Eq. 9. Fig. 6. Table. 4. Ref. 24.

1. Problem formulation

Scientific and technological progress in the modern agro-industrial sector is inseparably linked to solving complex engineering problems aimed at optimizing housing conditions for farm animals. The creation and maintenance of an optimal microclimate form the foundation of intensive livestock production, as the biological potential of high-yield breeds can be fully realized only when environmental stress factors are minimized. Among these factors, the temperature–humidity regime plays a dominant role, affecting metabolic processes, immune response, reproductive performance, and overall livestock productivity. In the context of global climate change and the observed trend toward rising summer temperatures, the development and implementation of highly efficient, energy-saving air cooling systems have become a critical prerequisite for ensuring the competitiveness of domestic livestock production [1].

The problem of ensuring thermal comfort for animals has a well-defined physiological basis (Table 1) [1]. Each species of farm animals has its own thermoneutral zone-a range of ambient temperatures within which the organism expends minimal energy on thermoregulation. Deviations from this zone, particularly toward

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elevated temperatures, trigger adaptive responses that negatively affect production performance. For example, in cattle, heat stress leads to reduced dry matter intake, resulting in a decline in milk yield and deterioration of milk quality. In poultry production, overheating is even more critical due to the absence of sweat glands in birds, making them dependent on evaporative cooling through the respiratory system.

Table 1

Optimal microclimate parameters and their effect on animal productivity [1]

Livestock sector	Optimal temperature range (°C)	Critical humidity (%)	Impact of heat stress
Dairy cattle	10–18	60–75	Milk yield reduction by 15–25%
Pig production (fattening)	16–20	50–70	Deterioration of feed conversion efficiency
Poultry farming (broilers)	18–24	50–60	High risk of mass mortality
Poultry farming (laying hens)	18–22	50–65	Reduction in egg weight and shell thickness

Conventional methods of mitigating overheating, such as basic ventilation, often prove insufficient when the ambient air temperature exceeds 30 °C. The application of compressor-based air conditioning systems in livestock buildings is constrained by their high energy consumption and operational complexity under conditions of elevated dust levels and chemically aggressive environments. In this context, indirect evaporative cooling systems based on the utilization of the latent heat of water evaporation represent a rational alternative, as they enable substantial air temperature reduction while maintaining minimal electrical energy demand.

In contrast to direct evaporative cooling systems, indirect evaporative air coolers provide air cooling without increasing its moisture content, which is fundamentally important for livestock buildings with elevated sanitary and hygienic requirements. The absence of additional humidification of supply air has driven growing interest in such systems and stimulated their intensive development over recent decades [2]. This has contributed to the intensification of scientific research, the improvement of heat exchanger designs, and the expansion of industrial production and practical implementation of indirect evaporative cooling units in ventilation and air-conditioning systems [3].

The significant potential of indirect evaporative cooling has been recognized by leading engineering and industrial corporations worldwide, which has resulted in the application of corresponding heat exchangers in ventilation units for various purposes [4]. This trend reflects an increasing demand for alternative energy-efficient solutions capable of partially or fully replacing conventional mechanical vapor compression systems characterized by high energy consumption and substantial operating costs.

The principle of evaporative cooling has been known to humanity for thousands of years; however, its modern engineering implementation has undergone significant evolution. The process is based on the physical phenomenon of phase transition of water from the liquid to the gaseous state, which is accompanied by the absorption of a considerable amount of energy–approximately 2260 kJ per kilogram of evaporated water. In direct evaporative cooling (DEC) systems, this process occurs directly within the air stream supplied to the room. Although DEC represents an extremely simple and low-cost method, it has a significant drawback: each degree of temperature reduction is accompanied by an increase in the relative humidity of the air. In livestock buildings, where humidity is often already excessive due to physiological emissions from animals, this leads to a «steam-room» effect that further exacerbates heat stress.

Indirect evaporative cooling (IEC) fundamentally resolves this issue by separating the air streams. In the main channel of the heat exchanger, supply (primary) air flows and is cooled through the wall without direct contact with water. In the auxiliary (secondary) channel, water evaporation takes place, which removes heat from the heat exchanger wall. As a result, the moisture content of the supply air remains unchanged, which is a key requirement for facilities subject to strict sanitary and hygienic standards.

The heat transfer efficiency in IEC systems is described by complex heat and mass transfer equations. The temperature of the dry heat exchanger wall surface, T_{ω}, is determined by the balance between convective heat transfer from the dry air stream and the combined heat and mass transfer on the wet side. For the idealized case of counterflow air movement, the equation describing the variation of the supply air temperature T_1 along the coordinate xxx can be expressed as:

$$L_1 c_{p1} \frac{dT}{dx} = k \cdot P \cdot \left(T_{\omega}\left(x\right) + T_1\left(x\right)\right), \qquad (1)$$

where L_1 – is the mass flow rate of air; c_{p1} – is the specific heat capacity; k – is the overall heat transfer coefficient; P – is the wetted perimeter of the channel.

In the wet channel, the process is complicated by vapor diffusion. The variation in the enthalpy of the secondary air flow i_2 is described in terms of the evaporation potential:

$$L_2 \frac{di_2}{dx} = \beta \cdot P \cdot \left(i_{sat}\left(T_{\omega}\right) + i_2\left(x\right)\right), \qquad (2)$$

where β – is the mass transfer coefficient; i_{sat} – is the enthalpy of saturated air at the wall temperature.

It is the enthalpy gradient that acts as the driving force in indirect evaporative cooling (IEC) systems, enabling air temperatures lower than the ambient wet-bulb temperature to be achieved.

At the same time, a key limitation hindering the widespread adoption of indirect evaporative cooling units remains their relatively low thermal efficiency [6]. Typical indirect evaporative heat exchangers predominantly employ a cross-flow configuration, which restricts the achievable temperature driving force and, consequently, the cooling effectiveness. The application of counterflow arrangements, which potentially provide superior thermal performance, is constrained by structural and aerodynamic challenges associated with organizing air movement within the heat exchanger channels.

In this regard, a substantial body of research has focused on improving the efficiency of indirect evaporative air coolers in order to expand their applicability under various climatic conditions. The proposed approaches range from complex hybrid systems combining multiple types of heat exchangers to simpler structural and technological solutions. One of the most widely recognized improvement strategies is the so-called M-Cycle, whose concept aims to approximate the heat transfer process as closely as possible to ideal counterflow conditions. However, the implementation of pure counterflow in plate-type heat exchangers is extremely challenging due to channel geometry and the specific characteristics of air inlet and outlet arrangements. As a result, the M-Cycle has been practically realized in the form of perforated heat exchangers with a modified cross-flow configuration [7].

2. Analysis of recent research and publications

The intensive development of the livestock sector within the agro-industrial complex is accompanied by a steady increase in energy consumption, which has led to heightened attention to issues of energy efficiency and the rational use of fuel and energy resources. The growth in energy demand in livestock complexes is driven, on the one hand, by an increase in herd size and, on the other hand, by stricter requirements for microclimate parameters in production facilities in accordance with modern animal housing technologies [8, 9].

According to numerous studies, primary energy consumption in livestock production has increased by nearly 29% over recent decades. A significant share of final energy use is associated with microclimate control systems, particularly ventilation, heating, and air cooling. Of particular concern are the energy costs of air cooling, which exhibit a persistent upward trend due to increasing requirements for maintaining comfortable conditions for animals during the warm season [10].

In this context, the energy policies of many countries are aimed at reducing electricity consumption, decreasing reliance on fossil fuels, and promoting environmentally friendly and energy-efficient technologies. The growing demand for air conditioning, combined with the need for energy conservation, underscores the relevance of seeking alternative technical solutions capable of reducing energy consumption and expanding the use of renewable energy sources in the formation of microclimate conditions in livestock buildings.

The livestock sector of the agro-industrial complex is characterized by one of the highest potentials for improving energy efficiency among agricultural production sectors. Studies indicate that the energy required to create and maintain regulated microclimate parameters–particularly air temperature and humidity–constitutes a substantial portion of the total energy consumption of livestock facilities [11–14]. The largest energy expenditures are associated with heating supply ventilation air during the heating season, when heat-generating equipment consumes, according to various estimates, between 40% and 90% of total fuel and energy resources [15, 16]. Consequently, even partial reductions in energy consumption within these systems can result in a significant decrease in livestock production costs.

One effective approach to reducing energy consumption involves the utilization of heat from ventilation exhaust air to preheat supply air [17]. At the same time, the recovery of exhaust air heat is complicated by its low-grade nature, which necessitates the use of specialized technical solutions to ensure its effective utilization [18].

The most promising method for recovering heat from ventilation exhaust air is the application of heat recovery exchangers, which, due to their relatively simple design and high efficiency, have become widely used in residential, administrative, and industrial buildings. In microclimate control systems for livestock buildings, the use of ventilation heat exchangers makes it possible to reduce energy consumption for heating supply air by up to 80% [19].

Over the past two decades, a variety of energy-efficient and renewable technologies have been actively implemented in agro-industrial facilities, including heat recovery systems, heat pumps, solar thermal installations, and hybrid energy systems [20]. However, in contrast to heating systems, technologies based on renewable energy sources have not yet gained widespread practical application in air cooling systems for livestock buildings, which highlights the need for further research in this area.

To ensure effective removal of contaminated air from pig housing facilities, automated ventilation systems have been developed that implement controlled air extraction from the occupied zone of the premises [15, 21]. As a result of analytical studies of such systems, mathematical conditions for their efficient operation have been obtained. The proposed methodologies and the algorithms developed on their basis allow for determining optimal opening areas of intake dampers equipped with servo drives in ventilation ducts. In addition, relationships have been established between pressure losses and the power required to move air through an indirect evaporative heat exchanger operating according to the Maisotsenko cycle, and the geometric parameters of the system, air flow rates, distances between ducts, and their number [22].

3. The purpose of the article

The purpose of this study is to improve the efficiency of microclimate control systems for livestock buildings by experimentally investigating an indirect evaporative heat exchanger and establishing empirical relationships between the parameters of the primary air stream and the key thermal performance indicators of the indirect evaporative cooling process.

4. Results and discussion

To conduct the experimental investigations, a laboratory-scale experimental prototype of an indirect evaporative heat exchanger was fabricated. The prototype was designed to study the processes of indirect evaporative air cooling. A three-dimensional (3D) model of the heat exchanger with the adopted geometric dimensions is shown in Fig. 1.

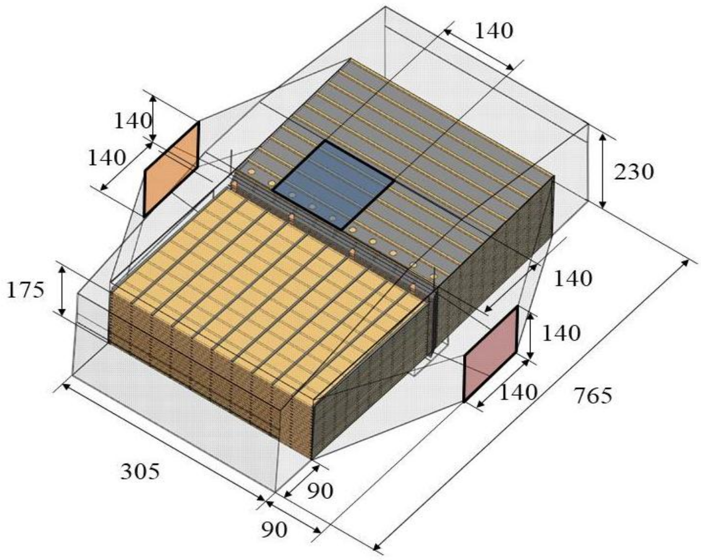


Fig. 1. 3D model of the indirect evaporative heat exchanger with adopted geometric dimensions (mm)

The proposed indirect evaporative heat exchanger is designed as a heat and mass transfer module housed within a thermally insulated enclosure. The primary functional element of the system is the heat and mass exchanger, which directly provides heat transfer between the primary (dry) and secondary (wet) air streams and, consequently, enables cooling of the primary air.

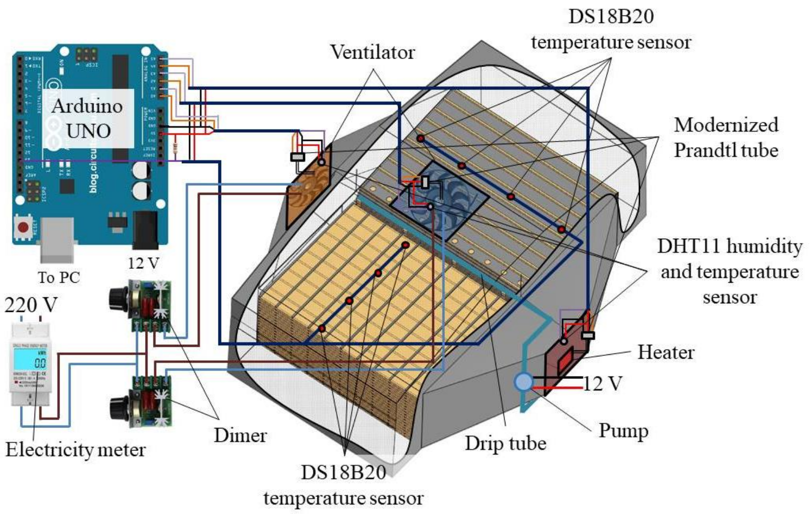
The heat and mass exchanger was fabricated using A3-format photo paper with a basis weight of 210 g/m² and double-sided adhesive tape with a thickness of 4 mm and a width of 4 mm. The selection of these materials was justified by their availability, low cost, and suitability for forming dry and wet channels on opposite sides of the heat transfer surface. The photo paper has a glossy coating on one side and a fibrous structure on the other, which allows it to be used as a heat-conducting partition that ensures heat transfer without mass exchange. Water impermeability on the dry channel side was further enhanced by laminating the glossy surface with aluminum foil of 0.1 mm thickness, which also reduces the likelihood of mechanical damage to the material.

The double-sided adhesive tape was used as a spacer element to form air channels between the photo paper plates. The configuration of the dry and wet channels was created by arranging the tape at a 90° angle, corresponding to a cross-flow air movement scheme. Water distribution within the wet channels was performed after channel formation, ensuring uniform wetting of the heat exchange surface.

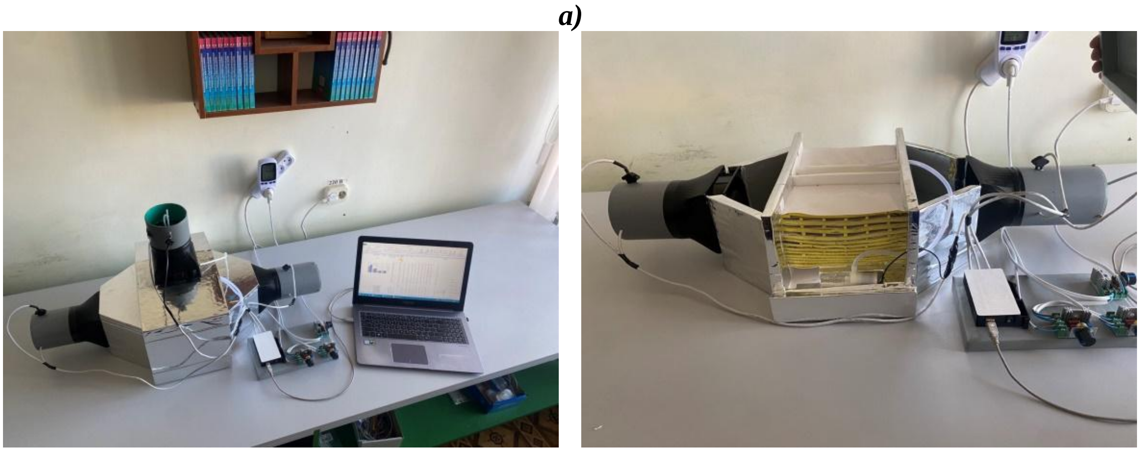
The enclosure of the indirect evaporative heat exchanger (Fig. 2) was manufactured from expanded polystyrene and additionally laminated with aluminum foil of 0.1 mm thickness to reduce heat losses to the surrounding environment. The enclosure has a modular, demountable design, providing convenient access to the heat and mass exchanger during experimental investigations. It is equipped with one inlet and two outlet rectangular openings measuring 140×140 mm, which can be connected to a ventilation system or directly fitted with fans.

A water reservoir is located at the bottom of the heat exchanger and connected to a pump that delivers water to the upper section of the heat and mass exchanger, followed by its distribution via a drip tube.

The laboratory test bench used for the experiments consisted of the experimental indirect evaporative heat exchanger prototype equipped with two fans (Fig. 2). Fan capacity was controlled using a dimmer connected to a 220 V AC power supply through an electricity meter, which was used to determine the electrical power consumption.



a)



b)

Fig. 2. Schematic diagram (a) and general view (b) of the laboratory test bench for investigating an indirect evaporative heat exchanger

Modified Prandtl tubes based on the MPX5100DP differential pressure sensor were installed in the inlet and two outlet air ducts, along with DHT11 air temperature and humidity sensors. All sensors were connected to an Arduino UNO control board. In addition, digital DS18B20 temperature sensors were installed in the central section of the heat and mass exchanger, in both the dry and wet channels, and connected to the control board via a 1-Wire communication bus.

The data collected from the sensors were transmitted in real time to a personal computer operating in oscilloscope mode, with the capability of continuous recording of experimental data.

The experimental investigations were carried out by varying the following factors [23, 24]:

– the opening area between the channels of the heat and mass exchanger (with identical and different opening areas) – Fig. 3;

– air flow in the outlet channels Q_{in} (100 m^3/h, 300 m^3/h, 500 m^3/h);

– inlet air temperature T_{in} (20, 26 and 32 °C);

– inlet air absolute humidity x_{in} (5, 15 and 25 g/kg).

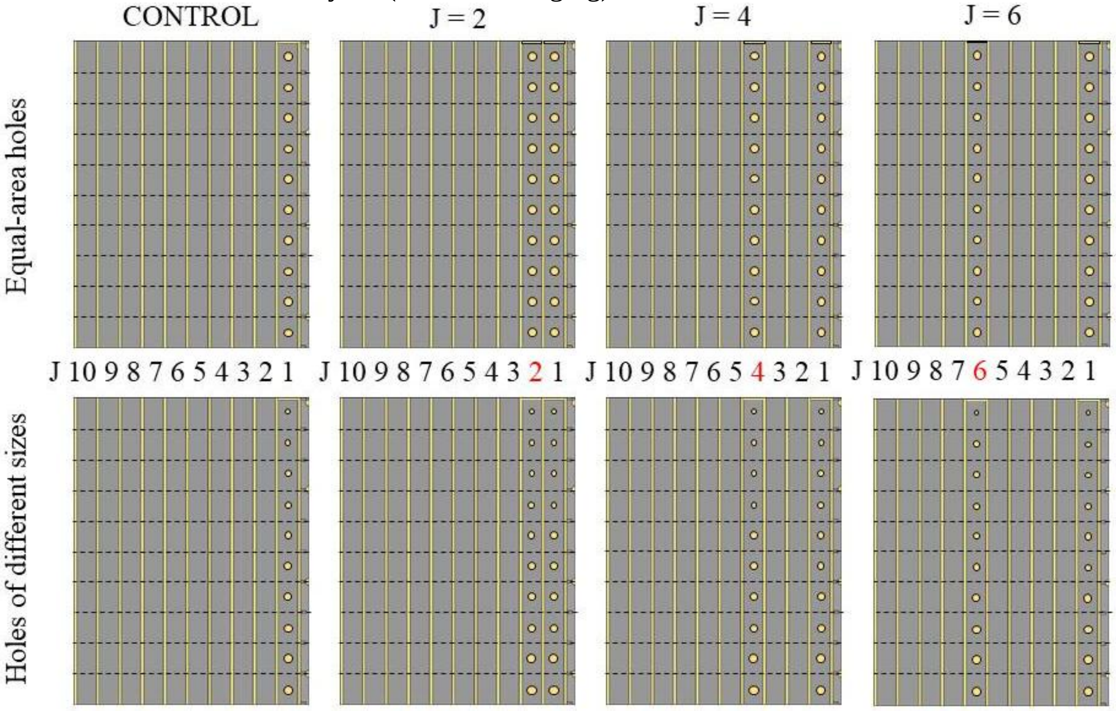


Fig. 3. Arrangement of openings between the channels of the heat and mass exchanger

The performance of the indirect evaporative heat exchanger was evaluated using the following criteria:

– dynamics of air temperature variation inside the heat and mass exchanger;

– air temperature and humidity at the outlet of the heat exchanger;

– thermal efficiency coefficient η_t;

– electrical power consumed by the fans for forcing air through the heat exchanger.

The results of the experimental investigations were compared with the results of numerical simulations, on the basis of which the obtained empirical relationships were refined. The updated relationships were subsequently used in the development of automatic control algorithms for microclimate control systems in pig housing facilities.

Experimental data processing was performed using Microsoft Excel and Wolfram Cloud software packages.

Based on the conducted experimental investigations of the indirect evaporative air heat exchanger, a representative dataset of primary experimental data was formed, reflecting the patterns of heat and mass transfer processes in the investigated system under various operating conditions.

Subsequent statistical and analytical processing of the obtained results was performed using modern software tools, namely Microsoft Excel and Wolfram Cloud, which ensured computational accuracy, enabled the development of regression models, and facilitated the visualization of multifactor relationships. The application of these software packages made it possible to approximate the experimental data, verify the adequacy of the models, and determine the statistical significance of the influence of the investigated factors.

As a result of experimental data processing, second-order regression equations were obtained that describe the variation in the outlet temperature of the primary air stream as a function of the combined research factors. The choice of a quadratic model is justified by the nonlinear nature of heat and mass transfer processes inherent in indirect evaporative air cooling. The values of the regression coefficients for the corresponding models are presented in Table 1.

The statistical significance of the regression coefficients was evaluated using Student's *t*-test. At a significance level of $\alpha = 0,05$ and with 30 degrees of freedom, the tabulated critical value of the test statistic is $t_{0,05(30)} = 2,044$.

Regression coefficients whose absolute values exceeded this threshold were considered statistically significant and were assumed to have a substantial effect on the formation of the outlet temperature of the primary air stream.

Taking into account only the statistically significant regression coefficients (Table 2), and after performing the decoding procedure of the normalized regression equations, analytical relationships for the outlet temperature of the primary air stream $\overline{t}_{out}$ as a function of the investigated factors were obtained in natural units. This made it possible to provide a physical interpretation of the influence of each factor and to determine their relative contribution to the overall cooling effect.

Table 2

Results of the statistical analysis of the coded second-order regression equations for the outlet temperature of the primary air stream $\overline{t}_{out}$

	b_0	b_1	b_2	b_3	b_{12}	b_{13}	b_{23}	b_{11}	b_{22}	b_{33}
Identical opening areas between the channels of the heat and mass exchanger										
Computed	23,3611	3,4534	2,4392	1,5086	0,5088	0,4082	-0,1303	-0,3019	-0,0583	0,0750
S^2	0,0337	0,0126	0,0126	0,0126	0,0253	0,0253	0,0253	0,0274	0,0274	0,0274
Δb	0,2624	0,1607	0,1607	0,1607	0,2272	0,2272	0,2272	0,2365	0,2365	0,2365
Significant	23,3611	3,4534	2,4392	1,5086	0,5088	0,4082	0	-0,3019	0	0
Different opening areas between the channels of the heat and mass exchanger										
	b_0	b_1	b_2	b_3	b_{12}	b_{13}	b_{23}	b_{11}	b_{22}	b_{33}
Computed	18,9546	4,1188	0,2946	0,8371	0,5542	0,5396	-0,3429	-0,1693	-0,1281	-0,0014
S^2	0,0538	0,0202	0,0202	0,0202	0,0404	0,0404	0,0404	0,0437	0,0437	0,0437
Δb	0,3316	0,2030	0,2030	0,2030	0,2871	0,2871	0,2871	0,2989	0,2989	0,2989
Significant	18,9546	4,1188	0,2946	0,8371	0,5542	0,5396	-0,3429	0	0	0

In particular, for the case of identical opening areas between the channels of the heat and mass exchanger, the corresponding functional relationships were obtained. These relationships adequately describe the heat transfer processes and confirm the suitability of the selected structural and technological parameters of the indirect evaporative heat exchanger:

$$\overline{t}_{out} = 2,65 - 0,0013\,Q_{in} + 0,00034\,t_{in}\,Q_{in} - 0,00839\,t_{in}^2 + 0,74\,t_{in} + 0,01696\,x_{in}\,t_{in} + 0,0469\,x_{in}; \quad (3)$$

For the case of different opening areas between the channels of the heat and mass exchanger:

$$\overline{t}_{out} = 6,54 - 0,00408\,Q_{in} + 0,000449\,t_{in}\,Q_{in} - 0,000343\,x_{in}\,Q_{in} + 0,367\,t_{in} + 0,0185\,x_{in}\,t_{in} - 0,318\,x_{in}. \quad (4)$$

Statistical analysis of the obtained regression equations within the investigated range of factor variation confirmed the homogeneity of variances of the experimental data. In particular, the calculated Cochran's test values were $G_{(3)} = 0,3146$ and $G_{(4)} = 0,1875$, which are lower than the tabulated critical value $G_{0,05(4;\ 15)} = 0,3346$. This confirms that the condition of variance homogeneity is satisfied and indicates the correctness of applying the developed regression models.

The adequacy of the developed regression models was evaluated using Fisher's test. The calculated values of the test statistic were $F_{(3)} = 1,5098$ and $F_{(4)} = 2,2466$, which are lower than the tabulated critical value $F_{0,05(8;30)} = 2,27$. These results demonstrate the adequacy of the developed mathematical models and confirm the possibility of their use for analyzing and predicting the outlet temperature of the primary air stream within the investigated factor space.

As a result of optimization calculations based on the criterion of minimizing the outlet temperature of the primary air stream, optimal values of the investigated factors were determined. In particular, the minimum outlet temperature values were $\overline{t}_{out}$ = 14,5 °C for the configuration with different interchannel opening areas and $\overline{t}_{out}$ = 16,5 °C for the configuration with identical opening areas. Under these conditions, the optimal inlet air parameters were: temperature t_{in} = 20 °C, absolute humidity x_{in} = 5 g/kg, and airflow rate Q_{in} = 100 m³/h.

A graphical interpretation of the experimental relationships for the configurations with different (3) and identical (4) opening areas between the channels, as well as the corresponding theoretical relationship, is presented in Fig. 4.

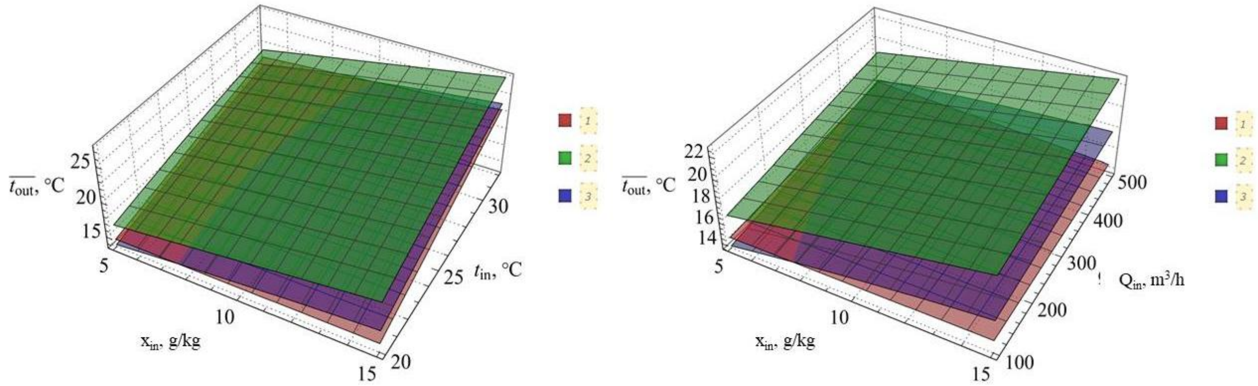


Fig. 4. Dependence of the outlet air stream temperature $\overline{t}_{out}$ on the inlet air temperature t_{in}, absolute humidity x_{in}, and inlet primary air flow rate Q_{in} at the heat exchanger inlet

Figure 4 illustrates the spatial relationships between the outlet temperature of the primary air stream $\overline{t}_{out}$ and the inlet air temperature tin, its absolute humidity x_{in}, and the inlet air flow rate Q_{in} for an indirect evaporative air heat exchanger. The presented three-dimensional response surfaces reflect the results of numerical modeling and experimental investigations for different structural configurations of the heat and mass exchanger.

A visual analysis of the presented plots indicates a high degree of agreement between the experimental data and the results of numerical simulation. Quantitative confirmation of this agreement is provided by the calculated Pearson correlation coefficient, which equals 0.93, indicating a strong linear relationship between the corresponding datasets and confirming the adequacy of the developed mathematical model.

An analysis of the response surfaces shows that with increasing inlet air temperature t_{in} and absolute humidity x_{in}, the outlet temperature of the primary air stream increases in a predictable manner, which is associated with a reduction in the indirect evaporative cooling potential. The influence of the air flow rate Q_{in} exhibits a less linear character and is determined by the combined effects of hydrodynamic phenomena and heat and mass transfer processes within the channels of the heat and mass exchanger.

A comparison of the results obtained for configurations with identical and different opening areas between the channels of the heat and mass exchanger demonstrates that, under identical factor values, the outlet temperature of the primary air stream is higher in the case of identical opening areas. This indicates lower heat and mass transfer efficiency under such structural conditions. The difference in the outlet temperature of the primary air stream between the considered configurations within the investigated range of factors varies from 1.1 to 4.2 °C, which constitutes an experimentally and practically significant result.

The obtained relationships confirm the feasibility of using different opening areas between the channels as an effective structural approach for intensifying indirect evaporative air cooling. For a more comprehensive quantitative assessment of the heat exchanger performance, it is reasonable to determine the thermal efficiency coefficient η_t and the effective cooling capacity N_E, which allow for a generalized energy-based evaluation of the investigated structural solutions.

Based on the statistical processing of the experimental data, second-order regression equations were obtained that describe the dependence of the thermal efficiency coefficient η_t on the combined set of investigated factors. The developed quadratic models account for both the main effects of the factors and their pairwise interactions, which is essential for adequately representing the nonlinear heat and mass transfer processes occurring in the indirect evaporative heat exchanger.

The numerical values of the regression coefficients for the corresponding models are presented in Table 3. The statistical significance of the regression coefficients was evaluated using Student's *t*-test. At a significance level of α = 0.05 and with 30 degrees of freedom, the tabulated critical value of the test statistic is $t_{0.05(30)}$ = 2.04.

Coefficients whose absolute values exceeded this threshold were considered statistically significant and were included in the subsequent analysis of factor effects on the thermal efficiency coefficient η_t.

Table 3

Results of the statistical analysis of the coded second-order regression equations for the thermal efficiency coefficient η_t

	b_0	b_1	b_2	b_3	b_{12}	b_{13}	b_{23}	b_{11}	b_{22}	b_{33}
Identical opening areas between the channels of the heat and mass exchanger										
Computed	0,4258	0,0884	0,0439	-0,0540	0,0060	-0,0234	0,0136	-0,0182	0,0165	0,0418
S_2	0,0009	0,0003	0,0003	0,0003	0,0006	0,0006	0,0006	0,0007	0,0007	0,0007
Δb	0,0420	0,0257	0,0257	0,0257	0,0364	0,0364	0,0364	0,0379	0,0379	0,0379
Significant	0,4258	0,0884	0,0439	-0,0540	0	0	0	0	0	0,0418
Different opening areas between the channels of the heat and mass exchanger										
	b_0	b_1	b_2	b_3	b_{12}	b_{13}	b_{23}	b_{11}	b_{22}	b_{33}
Computed	0,4624	0,0793	0,0340	-0,0752	-0,0013	-0,0008	-0,0310	0,0032	0,0011	0,0508
S_2	0,0006	0,0002	0,0002	0,0002	0,0005	0,0005	0,0005	0,0005	0,0005	0,0005
Δb	0,0359	0,0220	0,0220	0,0220	0,0311	0,0311	0,0311	0,0324	0,0324	0,0324
Significant	0,4624	0,0793	0,0340	-0,0752	0	0	0	0	0	0,0508

Taking into account the statistically significant regression coefficients presented in Table 2, and after performing the decoding procedure of the regression equations, analytical relationships describing the thermal efficiency coefficient η_t as a function of the investigated factors were obtained for the case of identical opening areas between the channels of the heat and mass exchanger.

$$\eta_t = 0{,}13 - 0{,}000897\ Q_{in} + 1{,}05\cdot10^{-6}\ Q_{in}^2 + 0{,}0147\ t_{in} + 0{,}00878\ x_{in}. \quad (5)$$

For the case of different opening areas between the channels of the heat and mass exchanger

$$\eta_t = 0{,}278 - 0{,}00114\ Q_{in} + 1{,}27\cdot10^{-6}\ Q_{in}^2 + 0{,}0132\ t_{in} + 0{,}00679\ x_{in}. \quad (6)$$

The optimal values of the investigated factors determined according to the criterion of maximizing the thermal efficiency coefficient η_t are $\eta_t = 0.70$ for the configuration with different opening areas between the channels of the heat and mass exchanger and $\eta_t = 0.65$ for the configuration with identical opening areas. Under these conditions, the rational inlet primary air parameters are as follows: inlet air temperature $t_{in} = 32$ °C, absolute humidity $x_{in} = 15$ g/kg, and air flow rate $Q_{in} = 100$ m³/h.

A graphical representation of the experimental relationships (5), (6), along with the corresponding theoretical relationship, is presented in Fig. 5.

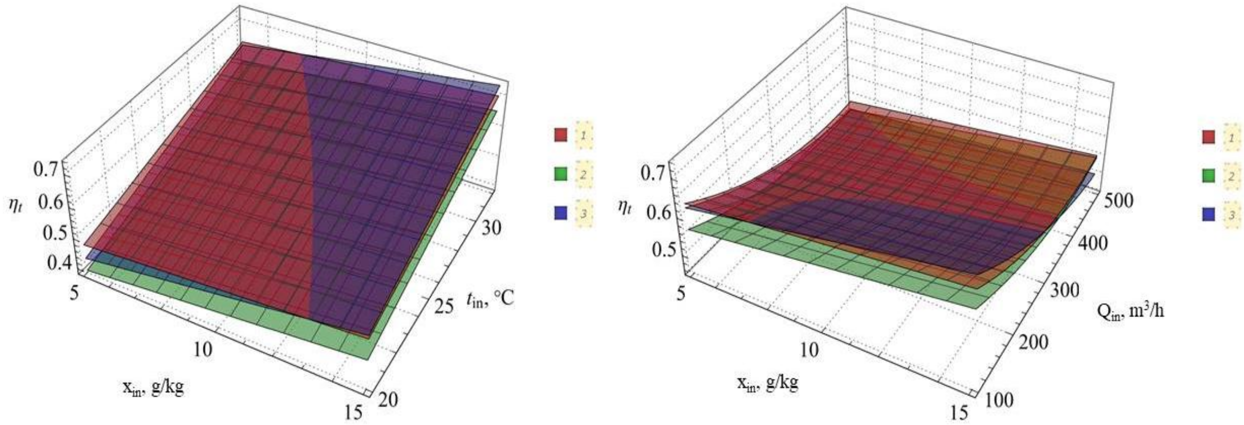


Fig. 5. Dependence of the thermal efficiency coefficient η_t on the inlet air temperature t_{in}, absolute humidity x_{in}, and inlet primary air flow rate Q_{in} at the heat exchanger inlet

Figure 5 illustrates the spatial relationships between the thermal efficiency coefficient η_t and the inlet primary air temperature t_{in}, its absolute humidity x_{in}, and the air flow rate Q_{in} at the inlet of the indirect evaporative heat exchanger. The presented response surfaces were constructed based on the results of experimental investigations and numerical modeling for different structural configurations of the heat and mass exchanger.

Statistical analysis of the developed regression equations within the investigated range of factor variation confirmed the homogeneity of variances of the experimental data. In particular, the calculated

Cochran's test values were $G_{(5)}$=0.3089 and $G_{(6)}$=0.1588, which are lower than the tabulated critical value $G_{0.05(4;15)}$=0.3346. Evaluation of model adequacy using Fisher's test showed that the calculated values $F_{(5)}$=1.6257 and $F_{(6)}$=2.1260 do not exceed the tabulated value $F_{0.05(10;30)}$=2.16, confirming the adequacy of the obtained regression models.

Visual analysis of the response surfaces indicates a high level of agreement between the results of numerical modeling and experimental investigations. Quantitative confirmation of this agreement is provided by the Pearson correlation coefficient, which equals 0.94, indicating a strong statistical relationship between the theoretical and experimental data.

Analysis of the plots shows that the thermal efficiency coefficient η_t increases with increasing inlet air temperature t_{in}, which is associated with an increase in the potential of indirect evaporative heat transfer. The influence of absolute humidity x_{in} is inverse: as the moisture content of the inlet air increases, the thermal efficiency coefficient decreases due to a reduction in the partial pressure difference and, consequently, the intensity of mass transfer processes. Changes in the air flow rate Q_{in} affect η_t in a nonlinear manner, which can be explained by the competition between increased convective heat transfer coefficients and reduced air-surface contact time.

A comparison of the results for heat exchanger configurations with different and identical opening areas between the channels shows that, for identical values of the investigated factors, the thermal efficiency coefficient η_t is higher in the case of different opening areas. This indicates more intensive heat and mass transfer and, consequently, higher performance of the heat exchanger under such a structural arrangement. The difference in the thermal efficiency coefficient between the considered configurations within the investigated factor range varies from 0.023 to 0.065, which constitutes a statistically and practically significant result.

The obtained relationships confirm the feasibility of using non-uniform opening areas between the channels of the heat and mass exchanger as an effective means of increasing the thermal efficiency of indirect evaporative air cooling systems and provide a scientific basis for further optimization of the structural and technological parameters of the heat exchanger.

Based on the statistical processing of the experimental data, second-order regression equations were obtained to describe the effective cooling capacity $\overline{N}_t$, defined as the difference between the direct cooling capacity of the heat exchanger and the power consumed by the fans for forcing air through the heat and mass exchanger. The numerical values of the regression coefficients are presented in Table 3. Their statistical significance was evaluated using Student's *t*-test, with the tabulated critical value at a significance level of α=0,05 and 30 degrees of freedom equal to $t_{0,05}(30)$ = 2,04.

Taking into account the statistically significant regression coefficients (Table 4), and after performing the decoding procedure of the regression equations, analytical relationships for the effective cooling capacity $\overline{N}_t$ as a function of the investigated factors were obtained for the case of identical opening areas between the channels of the heat and mass exchanger:

$$\overline{N}_t = 0{,}824 - 0{,}00396\, Q_{in}^2 + 8{,}92\, t_{in} + 1{,}355\, Q_{in} - 0{,}01008\, x_{in} Q_{in} - 2{,}726\, x_{in}. \quad (7)$$

For the case of different opening areas between the channels of the heat and mass exchanger

$$\begin{gathered}\overline{N}_t = 176{,}02 - 0{,}00389\, Q_{in}^2 + 5{,}45\, t_{in} + \\ + 1{,}09\, Q_{in} + 0{,}0106\, t_{in} Q_{in} - 0{,}022\, x_{in} Q_{in} - 7{,}033\, x_{in}.\end{gathered} \quad (8)$$

The optimal values of the investigated factors determined according to the criterion of maximizing the effective cooling capacity N_E are as follows: $\overline{N}_t$ = 426 W for the configuration with different opening areas between the channels and $\overline{N}_t$ = 380 W for the configuration with identical opening areas. Under these conditions, the rational inlet primary air parameters are t_{in} = 32 °C, x_{in} = 5 g/kg, and Q_{in} = 169 m³/h.

Next, the theoretical relationship will be transformed for the laboratory-scale heat exchanger prototype with the geometric parameters V_t = 2 × 155 mm × 283 mm × 283 mm, taking into account power losses associated with air forcing. As a result, the following expression is obtained:

$$\begin{gathered}\overline{N}_t = 84{,}11 - 120{,}43\, \overline{l_y^w}^{\,2} - 0{,}0041\, Q_{in}^2 + 8{,}35\, t_{in} - \\ - 0{,}0163\, t_{in}^2 + 1{,}354\, Q_{in} + 0{,}00387\, Q_{in} t_{in} - \\ - 0{,}01704\, Q_{in} x_{in} + 90{,}96\, \overline{l}_y^w - 0{,}1664\, Q_{in} \overline{l}_y^w - 1{,}408\, t_{in} \overline{l}_y^w + \\ + 0{,}5332\, x_{in} \overline{l}_y^w - 1{,}5434\, x_{in} - 0{,}225\, t_{in} x_{in} + 0{,}0979\, x_{in}^2.\end{gathered} \quad (9)$$

A graphical representation of the experimental relationships (7), (8) and the corresponding theoretical relationship (9) at $\overline{l}_y^{\,w}$ = 0.125 is presented in Fig. 6.

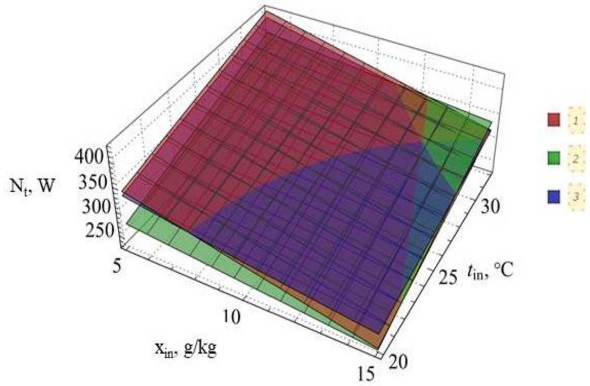
Table 4

Results of the statistical analysis of the coded second-order regression equations for the effective cooling capacity N_E

Identical opening areas between the channels of the heat and mass exchanger										
	b_0	b_1	b_2	b_3	b_{12}	b_{13}	b_{23}	b_{11}	b_{22}	b_{33}
Computed	225,14	53,544	-28,745	-224,77	-6,619	2,835	-10,076	0,751	5,305	-158,54
S_2	41,0316	15,386	15,386	15,386	30,77	30,77	30,77	33,33	33,33	0,0007
Δb	9,1541	5,6057	5,6057	5,6057	7,927	7,927	7,927	8,251	8,251	8,2514
Significant	225,14	53,544	-28,745	-224,77	0	0	-10,07	0	0	-158,54
Different opening areas between the channels of the heat and mass exchanger										
	b_0	b_1	b_2	b_3	b_{12}	b_{13}	b_{23}	b_{11}	b_{22}	b_{33}
Computed	239,86	51,856	-68,347	-238,51	-1,159	12,774	-22,122	2,200	2,203	-155,67
S_2	56,3931	21,147	21,147	21,147	42,29	42,294	42,294	45,81	45,81	45,819
Δb	10,7317	6,5718	6,5718	6,5718	9,294	9,294	9,294	9,673	9,673	9,6735
Significant	239,86	51,856	-68,347	-238,51	0	12,77	-22,122	0	0	-155,67

Statistical analysis of the obtained equations within the investigated range of factor variation confirmed the homogeneity of variances of the experimental data. In particular, the calculated Cochran's test values were $G_{(7)} = 0{,}1948$ and $G_{(8)} = 0{,}1974$, which are lower than the tabulated critical value $G_{0{,}05(4;\,15)} = 0{,}3346$.

Evaluation of the adequacy of the developed models using Fisher's test showed that the calculated values $F_{(7)} = 2{,}1941$ and $F_{(8)} = 1{,}9020$ do not exceed the tabulated critical value $F_{0{,}05(10;30)} = 2{,}27$. This confirms the adequacy of the obtained mathematical models and their suitability for further analysis of the effective cooling capacity of the heat exchanger.



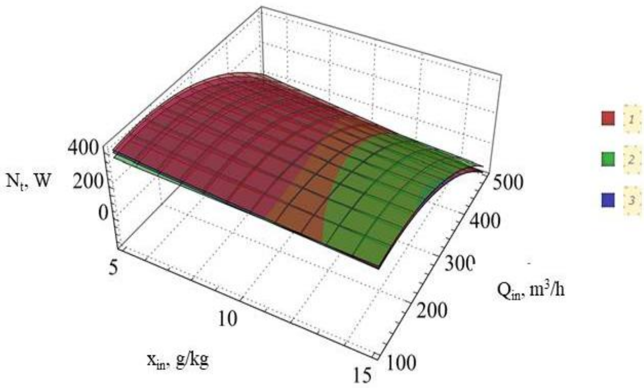


Fig. 6. Dependence of the effective cooling capacity $\overline{N}_t$ on the inlet air temperature t_{in}, absolute humidity x_{in}, and inlet primary air flow rate Q_{in} at the heat exchanger inlet

Figure 6 presents the spatial relationships between the effective cooling capacity $\overline{N}_t$ and the inlet primary air temperature t_{in}, its absolute humidity x_{in}, and the air flow rate Q_{in} at the inlet of the indirect evaporative heat exchanger. The displayed three-dimensional response surfaces were constructed based on the results of experimental investigations and numerical modeling and illustrate the patterns of variation in the cooling capacity of the heat exchanger under different operating conditions.

Analysis of the presented plots indicates a high degree of agreement between the results of numerical modeling and experimental studies. Quantitative confirmation of this agreement is provided by the Pearson correlation coefficient, which equals 0.92, indicating a strong statistical relationship between the corresponding datasets and confirming the adequacy of the applied mathematical models.

The analysis of the response surfaces shows that the effective cooling capacity $\overline{N}_t$ is strongly dependent on the inlet air flow rate Q_{in}, whereas the influence of the inlet air temperature t_{in} and absolute humidity x_{in} exhibits a more complex, nonlinear character. An increase in the absolute humidity of the inlet air reduces the indirect evaporative cooling potential, which consequently leads to a decrease in the values of $\bar{N}_t$.

A comparative analysis of the structural configurations of the heat exchanger demonstrates that, for identical values of the investigated factors, the effective cooling capacity is higher for the configuration with different opening areas between the channels of the heat and mass exchanger. This indicates an intensification of heat and mass transfer processes and, overall, higher performance of the heat exchanger under this structural arrangement. The difference in effective cooling capacity between the configurations with different and identical opening areas within the investigated factor range varies from −36 to 44 W, which constitutes an experimentally and practically significant result.

Visual analysis of the plots also shows that the maximum value of the effective cooling capacity $\bar{N}_t$ is achieved at an inlet primary air flow rate of Q_{in}=169 m³/h, which makes it possible to determine a rational operating regime of the indirect evaporative heat exchanger from the standpoint of energy efficiency.

5. Conclusion

1. It has been confirmed that ensuring regulated microclimate parameters in livestock buildings is a critical factor for maintaining animal productivity and health. Under conditions of elevated summer temperatures, conventional ventilation systems often fail to provide the required cooling effect, while compressor-based air conditioning systems for farms are limited by high energy consumption and operational complexity in dusty and aggressive environments.

2. The feasibility of applying indirect evaporative cooling systems as an energy-efficient alternative has been substantiated. Such systems ensure a reduction in supply air temperature without increasing its moisture content, which is fundamentally important for livestock facilities with stringent sanitary and hygienic requirements.

3. A laboratory-scale experimental prototype of an indirect evaporative heat exchanger was fabricated, and a representative dataset of primary experimental data was obtained by varying inlet air temperature, absolute humidity, and airflow rate, as well as for two structural configurations of the heat and mass exchanger (with identical and different interchannel opening areas).

4. Based on statistical analysis, second-order regression equations were obtained for the outlet temperature of the primary air stream, the thermal efficiency coefficient, and the effective cooling capacity. The statistical validity of the models was demonstrated by confirming the homogeneity of variances (Cochran's test), model adequacy (Fisher's test), and a high level of agreement between experimental results and numerical modeling (Pearson correlation coefficient in the range of 0.92–0.94).

5. The influence patterns of the investigated factors were established. An increase in inlet air temperature and absolute humidity leads to a reduction in the indirect evaporative cooling potential, manifested by a higher outlet air temperature and lower effective cooling capacity. The effect of airflow rate is nonlinear due to the competition between enhanced heat transfer and reduced contact time within the heat exchanger channels.

6. Comparative analysis demonstrated the superiority of the structural configuration with different interchannel opening areas, which intensifies heat and mass transfer processes and, as a result, provides a lower outlet air temperature and higher values of η_t and N_E within the investigated operating range.

7. Rational operating modes were determined. The minimum outlet air temperature is achieved at t_{in} = 20 °C, x_{in} = 5 g/kg та Q_{in} = 100 m³/h. Maximum thermal efficiency is observed at elevated inlet air temperatures combined with moderate humidity and low airflow rates. The maximum effective cooling capacity is achieved at Q_{in}≈169 m³/h (for the configuration with different opening areas), which is of practical importance for energy optimization of the system.

8. The obtained empirical relationships can be used as a basis for developing automatic microclimate control algorithms in livestock buildings (in particular, pig housing facilities) and for further optimization of the structural and technological parameters of indirect evaporative heat exchangers.

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ДОСЛІДЖЕННЯ ВПЛИВУ КОНСТРУКТИВНИХ ПАРАМЕТРІВ ТЕПЛООБМІННИКА ПОБІЧНО-ВИПАРНОГО ТИПУ НА ЕФЕКТИВНІСТЬ ОХОЛОДЖЕННЯ ПОВІТРЯ

Стаття присвячена дослідженню впливу конструктивних параметрів теплообмінника побічно-випарного типу на ефективність охолодження повітря в системах забезпечення мікроклімату тваринницьких приміщень. Актуальність роботи зумовлена зростанням теплових навантажень у літній період, підвищенням вимог до комфортних умов утримання тварин та необхідністю зниження енерговитрат на процеси кондиціювання і вентиляції повітря в агропромисловому комплексі.

У роботі обґрунтовано доцільність використання систем непрямого (побічно-випарного) охолодження як енергоефективної альтернативи традиційним компресорним холодильним установкам, що дозволяють знижувати температуру припливного повітря без збільшення його вологовмісту. Розроблено та виготовлено лабораторно-експериментальний зразок теплообмінника побічно-випарного типу з перехресною схемою руху повітряних потоків. Проведено експериментальні дослідження за варіювання температури, абсолютної вологості та витрати вхідного повітря, а також для двох конструктивних варіантів тепломасообмінника – з однаковими та різними площами отворів між каналами.

За результатами статистичної обробки експериментальних даних отримано рівняння регресії другого порядку для температури вихідного первинного повітряного потоку, коефіцієнта теплової ефективності та ефективної холодопродуктивної потужності теплообмінника. Адекватність математичних моделей підтверджена за критеріями Кохрена та Фішера, а також високими значеннями коефіцієнта кореляції між експериментальними та розрахунковими даними.

Встановлено, що використання різних площ отворів між каналами тепломасообмінника забезпечує інтенсифікацію тепло- та масообмінних процесів, зниження температури вихідного повітря та підвищення теплової ефективності й ефективної холодопродуктивності порівняно з варіантом однакових площ отворів. Визначено раціональні режими роботи теплообмінника з позицій мінімізації температури вихідного повітря та максимізації енергетичної ефективності. Отримані результати можуть бути використані при розробленні та оптимізації систем мікроклімату тваринницьких приміщень і створенні алгоритмів автоматичного керування процесами охолодження повітря.

Ключові слова: *побічно-випарне охолодження, теплообмінник, мікроклімат, тваринницькі приміщення, охолодження, повітря, тепломасообмін, коефіцієнт теплової ефективності, ефективна холодопродуктивна потужність, енергоефективність, вентиляція, конструктивні параметри, моделювання, агропромисловий комплекс.*

Ф. 9. Рис. 6. Табл. 4. Літ. 24.

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