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# THEORETICAL STUDY AND MODELING OF AERODYNAMIC LOSSES IN THE CHANNELS OF AN INDIRECT EVAPORATIVE HEAT EXCHANGER

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*The article provides a theoretical substantiation of the aerodynamic characteristics of an indirect evaporative air heat exchanger operating according to the Maisotsenko thermodynamic cycle (M-cycle) and intended for microclimate control systems in livestock buildings. The study is motivated by increasing summer thermal loads, the limited performance of conventional direct adiabatic cooling, and the need to reduce ventilation energy use without increasing the humidity of supply air.*

*A design concept featuring cross-flow interaction of dry and wet air streams is proposed, enabling deep cooling to temperatures close to the ambient dew point. A comprehensive mathematical model for pressure-loss prediction in the heat exchanger channels is developed. The model is based on an analytical decomposition of the airflow path into characteristic sections and accounts for channel geometry, flow regime, and the equivalent roughness of wetted surfaces.*

*Numerical analysis establishes the influence of channel cross-sectional area, number of channels, and airflow velocity on total aerodynamic resistance and required fan power. The channel cross-sectional shape is shown to be a decisive factor for energy performance: for equal cross-sectional areas, circular channels reduce total pressure losses by up to 26% compared with square and triangular channels. The results can be used to optimize the design of indirect evaporative heat exchangers and to improve the energy efficiency of ventilation and air-conditioning systems in livestock buildings.*

***Keywords:*** *indirect evaporative cooling; Maisotsenko cycle; heat exchanger; pressure losses; aerodynamic resistance; channel shape; energy efficiency; microclimate.*

***Eq. 15. Fig. 12. Ref. 23.***

## 1. Problem formulation

The current state of global and domestic livestock production is characterized by the intensification of production processes, which imposes stringent requirements on the microclimate parameters in livestock buildings. Under conditions of global climate change, accompanied by rising average annual temperatures and an increasing duration of abnormal heat periods, the problem of effective cooling of supply air has become strategically important. For pigs, as a biological species with limited thermoregulatory mechanisms, heat stress leads to a critical reduction in average daily weight gain, deterioration of reproductive performance, and an increase in morbidity, which collectively undermines the economic profitability of the industry [1].

Traditional approaches to mitigating animal overheating are based on the use of vapor-compression refrigeration systems or direct evaporative cooling technologies. However, an analysis of the operational experience of such systems reveals a number of fundamental limitations. Vapor-compression cooling units (split-type air conditioners) are characterized by excessive energy consumption, high material intensity, and low durability in the aggressive environment of pig housing facilities, where high concentrations of ammonia and organic dust accelerate corrosion of heat exchange surfaces [2].

On the other hand, widely used direct evaporative cooling systems (spray nozzles, Cool-Pad systems), although energy efficient, significantly increase the relative humidity of the supply air. Under high ambient temperature conditions, this creates a “greenhouse effect” that inhibits moisture evaporation from the respiratory tract of animals and intensifies heat stress. Thus, there is an urgent scientific and practical need to develop systems capable of reducing air temperature without increasing its moisture content.


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One of the most promising solutions to this problem is the application of indirect evaporative cooling based on the Maisotsenko thermodynamic cycle (M-cycle) [3]. A fundamental feature of this process is the ability to cool air to temperatures close to the ambient dew point, providing a substantially higher cooling potential compared to conventional adiabatic methods. The implementation of the M-cycle in indirect evaporative air heat exchangers allows the separation of dry (supply) and wet (working) air channels, thereby eliminating direct contact between moisture and the air supplied to the occupied zone [4].

Despite the high theoretical thermodynamic efficiency, the practical implementation of M-cycle heat exchangers is constrained by the complexity of their aerodynamic structure. The intricate configuration of channels with separating elements, the presence of perforations for flow redistribution, and the specific roughness of hygroscopic wetted surfaces result in significant aerodynamic resistance.

Pressure losses in such devices directly affect the required power of ventilation equipment and, consequently, the overall energy efficiency of the system [5]. At present, there is a noticeable scientific gap in the field of precise theoretical substantiation of the aerodynamic characteristics of thin-layer channels in indirect evaporative coolers, particularly with regard to the influence of cross-sectional geometry and the physical properties of wetted surfaces.

Numerous researchers have addressed the improvement of microclimate systems and the investigation of heat and mass transfer processes. The fundamental principles of thermodynamics and heat transfer were established in the works of leading scientists [6–9]. In the context of the development of the Maisotsenko cycle, worldwide recognition has been gained by the works of V. Maisotsenko himself, as well as by research conducted by the Coolerado Corp and Munters scientific groups. In Ukraine, a significant contribution to the development of energy-efficient ventilation systems for the agro-industrial complex has been made by researchers of Vinnytsia National Agrarian University [10–13], particularly with respect to adapting these technologies to the specific operating conditions of livestock buildings. However, the aerodynamic aspects of pressure losses in multichannel indirect evaporative structures with cross-flow configuration remain insufficiently addressed in the scientific literature.

The relevance of this study is determined by the necessity to overcome the “aerodynamic barrier” in the design of high-performance cooling systems for the agro-industrial sector. The search for a compromise between heat transfer intensity and pneumatic resistance is the key to creating next-generation microclimate control systems that will not only ensure comfortable conditions for animal housing but also significantly reduce operational costs. Therefore, the theoretical investigation of pressure losses in an indirect evaporative air heat exchanger represents an important step in the advancement of modern agro-engineering science.

## 2. Analysis of recent research and publications

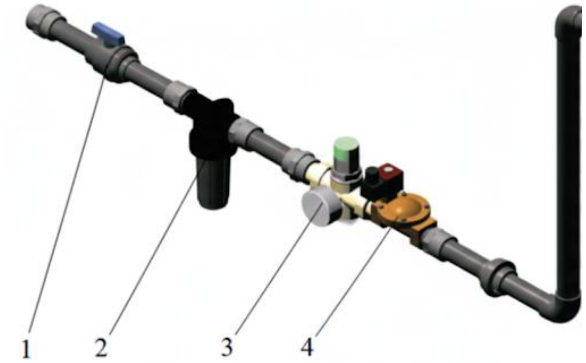
Ensuring regulatory microclimate parameters in livestock buildings, especially under conditions of global climate change, is a critical factor in preserving the genetic potential and productivity of animals. Modern technologies for transforming the parameters of supply air are generally classified into two key categories: energy-intensive vapor-compression systems and resource-efficient adiabatic cooling systems.

Vapor-compression «split» systems, despite their ability to provide precise thermal control, have significant limitations for industrial pig production. The main constraining factors include the high specific cost of cooling capacity, substantial metal intensity, and low operational reliability in environments characterized by elevated concentrations of ammonia, hydrogen sulfide, and organic dust. From the standpoint of energy management, water-evaporative cooling systems based on the absorption of latent heat during the phase change of water represent a more promising alternative.

Low-pressure spray systems (2–3 bar) constitute the simplest structural solution (Fig. 1).

However, analysis of their operation indicates low efficiency due to the large droplet size, which results in incomplete evaporation of moisture within the airflow. This leads to excessive wetting of the floor, an increase in relative air humidity beyond 80%, and a critical growth in the volume of the liquid fraction of manure slurry, thereby requiring additional costs for its handling and disposal [14].

As a technological alternative, the German company LUBING offers high-pressure systems (up to 70 bar) equipped with precision nozzles (Fig. 2) that generate a fine-dispersed mist. Although such systems minimize moisture deposition, they require highly sophisticated water treatment, including demineralization, in order to prevent salt clogging of nozzles with diameters ranging from 0.15 to 0.5 mm [15].



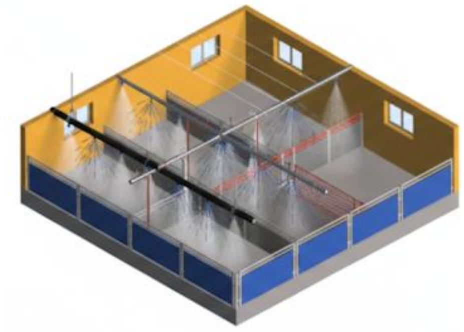



***Fig. 1. Connection scheme of the Egebjerg cooling system: 1 – ball valve; 2 – filter; 3 – pressure reducing valve; 4 – solenoid valve***

***Fig. 2. Cooling system manufactured by LUBING***

For localized cooling and disinfection in specific technological zones, centrifugal air coolers (e.g., units from the AOLAN product lines) are widely employed. Such zones typically include areas with elevated heat loads or increased sanitary requirements, where targeted treatment is preferable to full-scale conditioning of the entire building volume. These devices are manufactured in both stationary and mobile configurations (Fig. 3), which allows their application either as permanently installed equipment or as flexible, relocatable units adapted to changing operational needs. The operating principle is based on the mechanical disintegration (atomization) of a water jet by rapidly rotating discs, resulting in the formation of a fine droplet aerosol. Owing to the large interfacial area of the generated droplets, heat and mass transfer intensifies, providing a localized cooling effect and facilitating hygienic treatment of the surrounding air environment [16].


***a*** ***b***

***Fig. 3. Centrifugal air cooler:***
***a – stationary; b – mobile; 1 – drive mechanism; 2 – housing; 3 – fan; 4 – disk; 5 – tank; 6 – trolley***

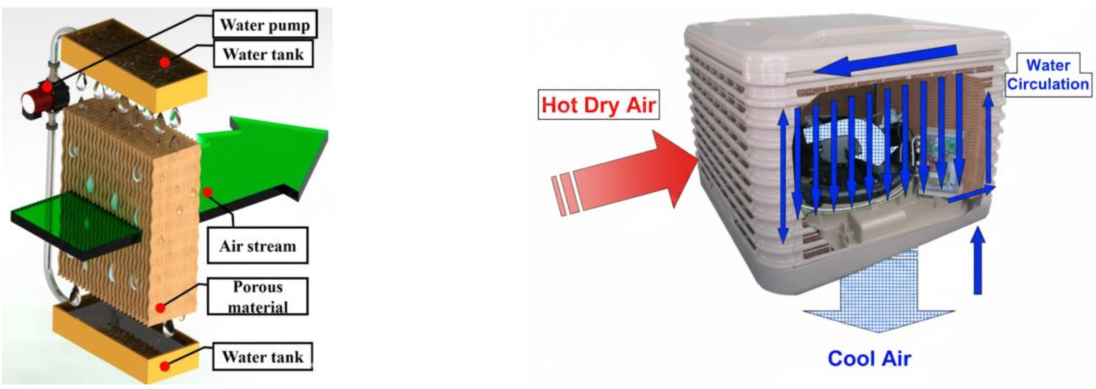
At present, systems based on irrigated porous media (Cool-Pad systems) are considered the most adapted to the operating conditions of pig production facilities, particularly modular BREEZAIR coolers (Fig. 4) [17].


*a* *b* *c*

*d* *e*

***Fig. 4. BREEZAIR coolers:***
***(a) with a central fan; (b) with a central fan and inverter; (c) built-in unit; (d) with an axial fan; (e) mobile unit***

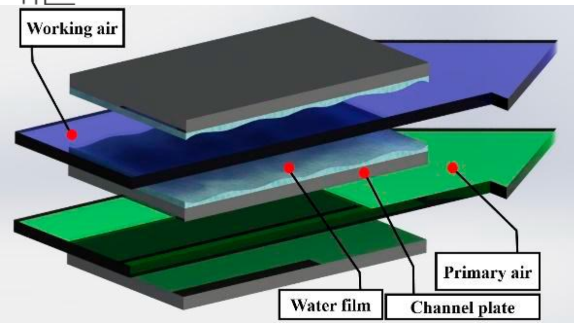
Despite their relatively high adiabatic efficiency (0.70–0.95), direct evaporative cooling units (Fig. 5) exhibit a fundamental limitation associated with a pronounced increase in the relative humidity of the air. Under ambient temperatures exceeding 30 °C, elevated humidity restricts the effectiveness of animal thermoregulation mechanisms and intensifies heat stress, thereby reducing animal comfort and productivity [18].



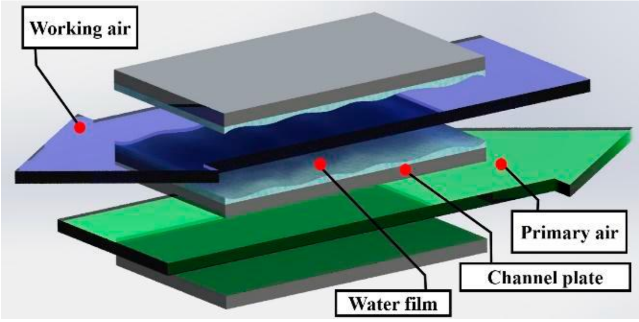

a b

***Fig. 5. Operating scheme (a) and general view (b) of direct evaporative air coolers***

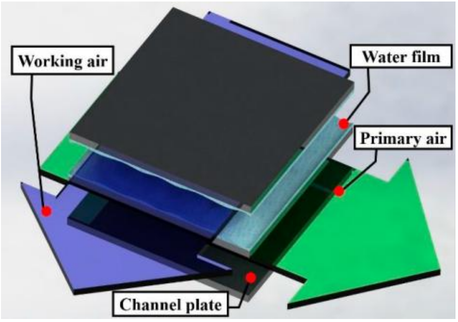
An innovative paradigm in agro-engineering is the transition to indirect evaporative cooling (IEC), which enables air temperature reduction without increasing the moisture content of the supply air [19]. This is achieved through the physical separation of dry (primary) and wet (secondary) air channels by a thermally conductive partition. Modern IEC units are designed using parallel-flow, cross-flow, or counter-flow configurations (Fig. 6).




***a***



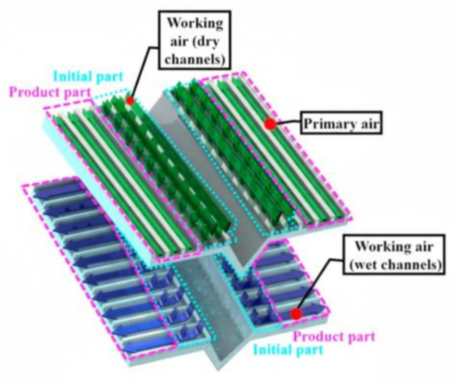

***b***




***c***

***Fig. 6. Indirect evaporative cooling:***
***(a) parallel-flow configuration; (b) counter-flow configuration; (c) cross-flow configuration***

Particular scientific and practical interest is attracted by the Maisotsenko thermodynamic cycle (M-cycle), implemented in heat exchangers developed by Coolerado Corp (Fig. 7) [20,21]. The fundamental concept of the M-cycle is based on the successive branching of the air stream through perforations in the plates, which provides staged pre-cooling of the working air prior to evaporation. This approach theoretically enables cooling to the ambient air dew point temperature, which is significantly lower than the wet-bulb temperature that represents the limiting threshold for conventional adiabatic cooling systems.




a

b

***Fig. 7. Operating scheme (a) and general view (b) of the cross-flow***
***Maisotsenko cycle implemented in an air cooler manufactured by Coolerado Corp***

However, the widespread implementation of the M-cycle in ventilation systems for the agro-industrial sector is constrained by aerodynamic factors. The complex configuration of channels with perforations, together with the roughness of hygroscopic wetted surfaces, generates substantial flow resistance. The principal challenge in designing such devices is to identify a compromise between the intensity of heat and mass transfer and pneumatic losses. Considering that pressure losses are proportional to the square of the airflow velocity, while effective cooling requires a certain contact time between the air stream and the wetted wall, there arises a need for an in-depth theoretical investigation of the aerodynamics of indirect evaporative heat exchangers. Therefore, the development of mathematical models for pressure loss prediction constitutes the foundation for creating next-generation energy-efficient microclimate control systems.

## 3. The purpose of the article

The aim of this study is to provide a theoretical substantiation and to establish the regularities governing pressure loss variations of airflows in the channels of an indirect evaporative heat exchanger, with the purpose of optimizing its structural parameters and reducing energy consumption for microclimate control in livestock buildings.

To achieve this aim, the following objectives were defined:

1. To analyze the influence of channel geometric parameters (height, width, and length) and the physical properties of the airflow on the overall aerodynamic resistance of the heat exchanger.
2. To develop a theoretical mathematical model for calculating pressure losses in the dry and wet channels of an indirect evaporative heat exchanger, taking into account the specific features of the Maisotsenko thermodynamic cycle.
3. To establish functional relationships between pneumatic friction coefficients and local loss coefficients and the Reynolds number for laminar and transitional airflow regimes in thin-layer channels.
4. To perform a comparative analysis of the theoretically predicted pressure losses with existing analogues in order to verify the adequacy of the proposed models and to assess the energy efficiency of the device.

## 4. Results and discussion

Object of the Study and Design Features of the Heat Exchanger. The object of the theoretical study is the process of pressure loss formation in an indirect evaporative air heat exchanger, the design of which is based on the principles of implementing the Maisotsenko thermodynamic cycle with a cross-flow configuration of air streams [11,22].

The main element of the investigated system is a heat and mass exchanger designed as a multilayer package of parallel plates (Fig. 9), which form an alternating arrangement of dry and wet channels.

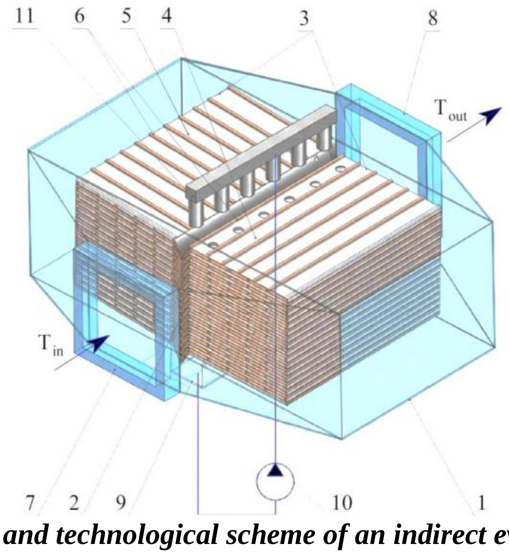



*Fig. 8. Structural and technological scheme of an indirect evaporative heat exchanger*

According to the structural and technological scheme (Fig. 9), the heat exchanger consists of a thermally insulated casing (1), inside which a plate package (3) is installed. The plates exhibit asymmetric properties: the surface facing the dry channel (4) is hydrophobic (moisture-impermeable), while the surface forming the wet channel (5) is hygroscopic in order to retain a stable water film. Channel separation and the provision of cross-flow air movement are ensured by dividing elements (6). The irrigation system includes a reservoir (9), a pump (10), and a drip distributor (11), which together provide continuous and uniform wetting of the surfaces.

The operating principle and the organization of heat and mass transfer processes in the indirect evaporative heat exchanger are as follows. Ambient air (Tin) enters the unit through the inlet opening (7) and is distributed between the dry and wet channels. The airflow in the dry channels (primary flow) moves along moisture-impermeable surfaces without a change in its moisture content.

Simultaneously, an auxiliary airflow is supplied to the wet channels. The irrigation system, consisting of the reservoir (9), pump (10), and drip distributor (11), ensures continuous water supply to the hygroscopic surface of the plates.

In the wet channels, intensive evaporation of the water film occurs, accompanied by the absorption of the latent heat of vaporization. This energy is extracted from the plate material, which, in turn, cools the airflow in the adjacent dry channel through heat conduction across the plate wall.

The cooled air (Tout) exits through the outlet openings (8) and is directed to the occupied zone of the building. In this process, the temperature of the supply air is significantly reduced without an increase in its absolute humidity, which represents a key advantage over direct evaporative cooling systems.

The proposed configuration enables high energy efficiency due to the staged pre-cooling of the working air; however, it necessitates a detailed investigation of aerodynamic resistance, since the presence of a water film and dividing elements within the channels substantially affects the overall pressure losses of the system.

Computational Scheme of the Channel and Aerodynamic Parameters.

For the theoretical investigation of pressure losses, a computational model of a single heat exchanger channel with length L, height h, and width b was adopted. Given the small geometric dimension of the channel height ($h \ll b$), the flow is considered as a fully developed viscous gas flow.

The total pressure loss ΔP (Pa) in the system is determined as the sum of distributed friction losses along the channel length and local losses at the inlet and outlet of the plate package [23]:

$$\Delta P = \Delta P_{fr} + \Delta P_{loc}. \quad (1)$$

The fundamental equation used to determine distributed (linear) pressure losses during airflow through the channels of the heat exchanger is the Darcy–Weisbach equation. For an airflow, it takes the following form:

$$\Delta P_{fr} = \lambda \cdot \frac{L}{d_e} \cdot \frac{\rho v^2}{2}, \quad (2)$$

where λ – pneumatic friction factor; L – channel length, m; $d_e$ – equivalent (hydraulic) diameter of the channel, m; ρ – air density, kg/m³; v – mean air velocity in the channel, m/s

For a planar channel, the hydraulic (equivalent) diameter is calculated based on its geometric dimensions [23]:

$$d_e = \frac{2hb}{h+b} \approx 2h. \quad (3)$$

Considering that, in indirect evaporative coolers, the airflow velocity is typically kept relatively low in order to ensure sufficient contact time with the wetted surface, the flow regime is often laminar (Re < 2300). In this case, the friction factor λ is determined theoretically as a function of the Reynolds number:

$$\lambda = \frac{K}{\mathrm{Re}}, \quad (4)$$

where K – a constant dependent on the channel cross-sectional shape (for planar slit channels, K ≈ 64–96) [23].

The Reynolds number for the heat exchanger channel is defined as:

$$\mathrm{Re} = \frac{\upsilon \cdot d_e}{\nu}, \quad (5)$$

where ν – kinematic viscosity of air, m²/s.

A specific feature of the proposed model is the consideration of the presence of a water film in the wet channels, which alters the equivalent surface roughness as well as the effective free cross-sectional area of the channel. This, in turn, introduces corrections into the calculation of the airflow velocity v.

Local pressure losses $\Delta P_{loc}$ are calculated using the following equation:

$$\Delta P_{loc} = \sum \zeta \cdot \frac{\rho \upsilon^2}{2}, \quad (6)$$

where ζ – local loss coefficients (inlet to the plate package, flow turning/bend, outlet).

The theoretical investigation of the origin and distribution of pressure losses in an indirect evaporative air heat exchanger made it possible to represent the device as a complex multicomponent system. According to the proposed computational scheme (Fig. 10), the structure of the unit consists of an array of autonomous working channels, as well as consecutively integrated dry and wet flow paths.

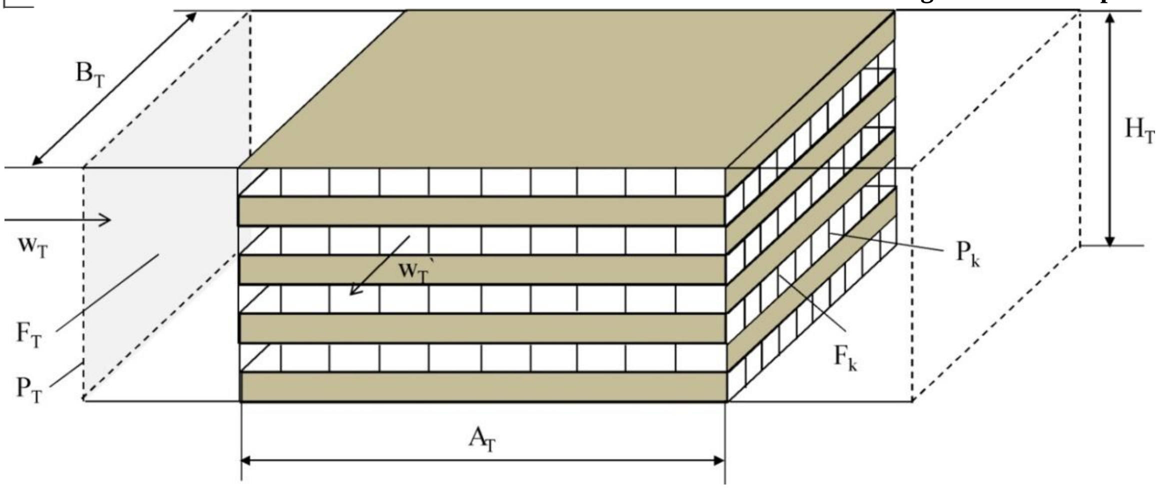



***Fig. 9. Computational scheme of an indirect evaporative heat exchanger***

The process of airflow through the functional channels of the heat exchanger was decomposed into three characteristic stages. At the initial stage, the total airflow supplied from a main air duct with a relatively large cross-sectional area $F_t$ is divided into a system of narrow channels with a total cross-sectional area $N_kF_k$. The pressure losses $\Delta p_{rk}$ in this section are calculated using a refined expression:

$$\Delta p_{Tk}=\eta_1\left(1-\frac{N_kF_k}{F_T}\right)\frac{\rho\left(w_T\right)^2}{2}, \tag{7}$$

where $\eta_1$ – a coefficient characterizing the degree of impact mitigation ($\eta_1$ = 0.5 was adopted) [23]; $N_k$ – total number of channels; $F_t$ – cross-sectional area of the inlet manifold, m²; $F_k$ – cross-sectional area of a single channel, m²; $w_t$ – mean airflow velocity, m/s.

The second stage involves fully developed airflow along the working channels of length $A_t$. It was established that the pressure losses $\Delta p_k$ in this section strongly depend on the viscous properties of the medium and the condition of the internal surfaces:

$$\Delta p_k=0{,}11N_k\;\frac{P_kA_T}{4F_k}\,\frac{\rho\,w_T^{\,2}}{2}\,\sqrt[4]{\frac{17\mu_aP_k}{F_k\times w_k\times\rho}+\frac{\psi_kP_k}{4F_k}}, \tag{8}$$

where $P_k$ – wetted perimeter of the channel, m; $\psi_k$ – equivalent wall roughness parameter; $\mu_a$ – dynamic viscosity of air ($\mu_a = 18.27 \times 10^{-6}$ N·s/m²).

The final stage of flow through the working channel is associated with the discharge of the air stream into the outlet air duct, which is accompanied by pressure losses $\Delta p_{kt}$ due to sudden expansion and merging of the individual jets:

$$\Delta p_{kT}=\eta_2\left(1-\frac{N_kF_k}{F_T}\right)^2\frac{\rho\left(w_T\right)^2}{2}, \tag{9}$$

where $\eta_2$ – the corresponding impact mitigation coefficient, equal to 0.5 [23].

A specific feature of the operation of indirect evaporative cooling systems is the requirement for the airflow to pass through five successive resistance zones. A key factor in this process is the spatial annular turning of the flow by 180° during the transition between the dry and wet channels. The pressure losses in this zone, $\Delta p'_{k180}$, are determined using the following relationship:

$$\Delta p_{k180}'=\zeta_{k80}'\frac{\rho\left(w_T'\right)^2}{2}, \tag{10}$$

where $\zeta_{k180}$ – the local loss coefficient associated with flow turning, adopted according to source [23] as $\zeta_{k180} = 2$.

By generalizing the obtained results for all flow sections, a comprehensive expression for determining the integral pressure losses of the indirect evaporative air heat exchanger, $\Delta p_t$, was derived:

$$\Delta p_T=\Delta p_{Tk}+\Delta p_k+\Delta p_{kT}+\Delta p_{Tk}`+\Delta p_k`+\Delta p_{k180}'+\Delta p_k``+\Delta p_{kT}`. \tag{11}$$

After performing the necessary mathematical transformations for a system with identical parameters of the dry and wet channels, the final governing equation was obtained:

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$$\Delta p_T=\frac{\rho\left(w_T\right)^2}{2}\left[\eta_1\left(1-\frac{N_kF_k}{F_T}\right)+0{,}11N_k\frac{P_kA_T}{4F_k}\sqrt[4]{\frac{17\mu_aP_k}{F_k\times w_T\times\rho}+\frac{\psi_kP_k}{4F_k}}+\eta_2\left(1-\frac{N_kF_k}{F_T}\right)^2\right]+$$
$$+\frac{\rho\left(w_T`\right)^2}{2}\left[\eta_1\left(1-\frac{N_kF_k}{F_T}\right)+0{,}11N_k\frac{P_kB_T}{4F_k}\sqrt[4]{\frac{17\mu_aP_k}{F_k\cdot w_T`\cdot\rho}+\frac{\psi_k`P_k}{4F_k}}+\zeta_{k80}+\right. \quad (12)$$
$$\left.+0{,}11N_k\frac{P_kB_T}{4F_k}\sqrt[4]{\frac{17\mu_aP_k}{F_k\cdot w_k``\rho}+\frac{\psi_k``P_k}{4F_k}}+\eta_2\left(1-\frac{N_kF_k}{F_T}\right)^2\right].$$

To determine the optimal heat exchanger design, a comparative analysis of three basic channel cross-sectional shapes - square, triangular, and circular - was conducted:

– square: $P_k=4\sqrt{F_k}$, (13)

– triangular: $P_k=2\sqrt{3F_k\sqrt{3}}$, (14)

– circular: $P_k=2\sqrt{\pi F_k}$. (15)

The dependence described by Eq. (11) and the required fan power required to drive the airflow through the indirect evaporative heat exchanger, determined using Eq. (7), are presented in Figs. 11 and 12.

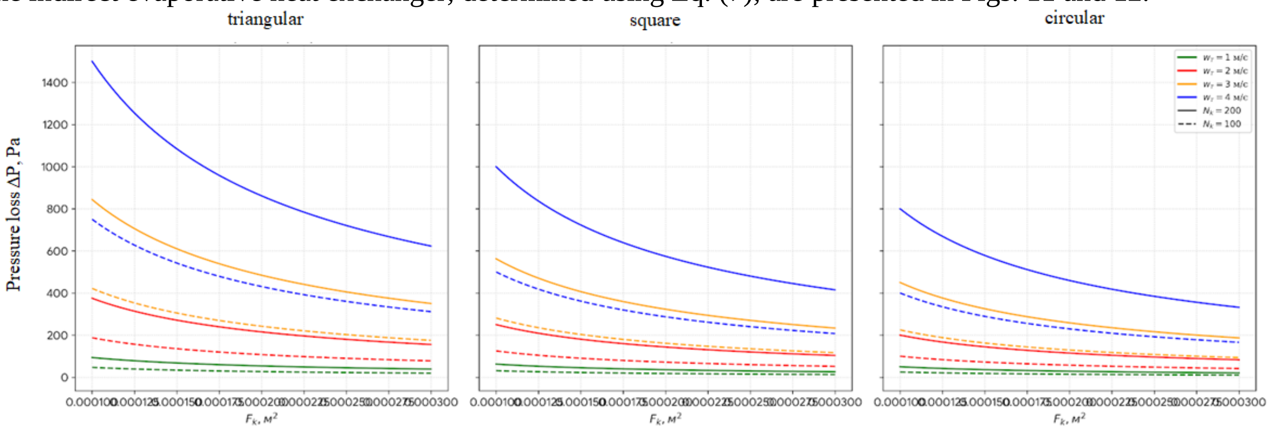


***Fig. 10. Dependence of total pressure losses $\Delta p_t$ on the number of channels $N_k$, channel cross-sectional area $F_k$, and mean air velocity in the channels, under the condition $w_t' = w_t$, for different channel shapes (square, triangular, and circular)***

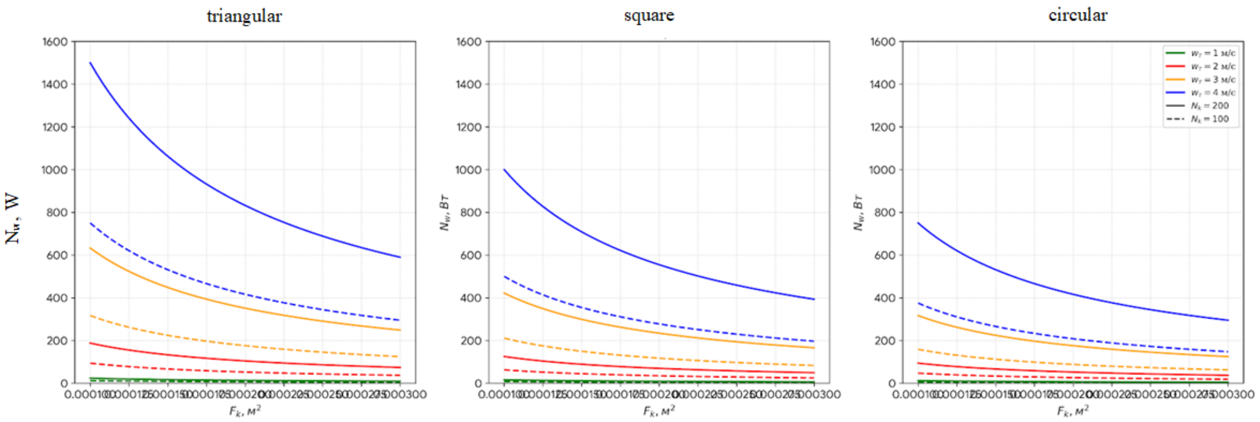


***Fig. 11. Dependence of the required power $N_{w1}$ on the number of channels $N_k$, channel cross-sectional area $F_k$, and mean air velocity in the channels, under the condition $w'_T = w_T$, for different channel shapes (square, triangular, and circular)***

Based on the developed mathematical framework and the obtained analytical relationships, a comprehensive analysis of the influence of geometric and kinematic parameters on the aerodynamic efficiency of the indirect evaporative heat exchanger was performed. The results of numerical simulations presented in figs. 11 and 12 make it possible to determine the nature of variations in total pressure losses $\Delta p_T$ and the required power $N_{w1}$ as functions of the channel cross-sectional area $F_k$, the number of channels $N_k$, and the mean airflow velocity $w_T$.

An analysis of the graphical dependencies of pressure losses $\Delta p_T$ (Fig. 11) reveals a pronounced nonlinear, nonlinear decreasing trend relationship with respect to the cross-sectional area of a single channel $F_k$. As $F_k$ increases, a significant reduction in aerodynamic resistance is observed for all investigated channel shapes. This behavior is explained by a decrease in the specific friction between the airflow and the channel walls as the effective flow area increases. In particular, at low airflow velocities ($w_T$ = 1–2 m/s), the curves exhibit a relatively gentle slope, whereas at higher velocities up to 4 m/s (blue curve), the resistance increases sharply. This indicates a transition toward turbulent flow regimes and an increased contribution of quadratic pressure losses.

The number of channels $N_k$ also has a substantial effect on the overall resistance of the system. As shown in Fig. 11, increasing the number of channels from 100 (dashed line) to 200 (solid line) leads to higher pressure losses. This is attributed to an increase in the total wetted perimeter of the heat exchange surface at a constant airflow velocity, which intensifies viscous friction forces.

The plots of the required power $N_{w1}$ to overcome aerodynamic resistance (Fig. 12) demonstrate similar trends but with a more pronounced amplitude of variation. The highest energy consumption is observed for channels with small cross-sectional areas ($F_k$ < 0.00015 m²) and high airflow velocities. It was established that increasing the airflow velocity from 1 m/s to 4 m/s results in an increase in fan power consumption by more than 8–10 times, which emphasizes the necessity of optimizing the velocity regime to ensure energy-efficient operation of microclimate control systems.

A key outcome of the comparative analysis is the determination of the influence of channel cross-sectional shape on energy performance. The highest aerodynamic resistance and, consequently, the greatest power consumption are characteristic of triangular channels. Square channels occupy an intermediate position, whereas circular channels exhibit the lowest pressure losses. Analysis of Figs. 11 and 12 indicates that, for equal cross-sectional areas $F_k$ and airflow velocities $w_T$, circular channels provide a reduction in pressure losses of up to 26% compared to other configurations. From a physical standpoint, this is explained by the fact that a circle has the minimum wetted perimeter for a given cross-sectional area, which minimizes the contact area between the airflow and the plate surfaces and reduces the total friction force.

Within the scope of this study, the functional relationships between friction coefficients, local loss coefficients, and the Reynolds number were employed in analytical form in accordance with classical fluid-dynamic correlations. This approach is justified for laminar and transitional airflow regimes in thin-layer channels.

Thus, the modeling results indicate the feasibility of designing indirect evaporative heat exchangers with circular or near-circular (polygonal) channels and a cross-sectional area exceeding 0.0002 m² in order to achieve an optimal balance between device compactness and energy consumption.

## 5. Conclusion

1. A comprehensive theoretical model for calculating pressure losses in an indirect evaporative air heat exchanger operating according to the Maisotsenko thermodynamic cycle has been developed. The model is based on the decomposition of the pneumatic flow path into characteristic zones and makes it possible to account for both distributed and local aerodynamic resistances caused by channel geometry and the presence of wetted surfaces.

2. Quantitative relationships describing the influence of geometric and kinematic parameters of the heat exchanger channels on total pressure losses and the power required for air transportation have been established. It is shown that an increase in the channel cross-sectional area leads to a nonlinear reduction in aerodynamic resistance, whereas an increase in airflow velocity beyond 3–4 m/s is accompanied by a sharp rise in energy consumption.

3. Based on a comparative analysis of different channel cross-sectional shapes, it has been demonstrated that a circular cross-section is the most energy-efficient. For an equal cross-sectional area, circular channels provide up to a 26% reduction in total pressure losses compared to square and triangular channels, which is explained by the minimized wetted perimeter and reduced viscous friction forces.

4. The practical significance of the obtained results lies in the possibility of applying the developed model to the engineering design and optimization of indirect evaporative heat exchangers in order to improve the energy efficiency of ventilation and air conditioning systems in livestock buildings and to reduce operational costs.

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## ТЕОРЕТИЧНЕ ДОСЛІДЖЕННЯ ТА МОДЕЛЮВАННЯ АЕРОДИНАМІЧНИХ ВТРАТ У КАНАЛАХ ТЕПЛООБМІННИКА ПОБІЧНО-ВИПАРНОГО ТИПУ

*Стаття присвячена теоретичному обґрунтуванню аеродинамічних параметрів повітряного теплообмінника побічно-випарного типу, що працює за термодинамічним циклом Майсоценка (М-циклом) та призначений для систем мікроклімату тваринницьких приміщень. Актуальність дослідження зумовлена зростанням теплових навантажень у літній період, обмеженою ефективністю традиційних систем прямого адіабатного охолодження та необхідністю зниження енерговитрат на вентиляцію без підвищення вологості припливного повітря.*

*Запропоновано конструктивно-технологічну схему теплообмінника з перехресним рухом сухого та вологого повітряних потоків, яка забезпечує глибоке охолодження повітря до температур, близьких до точки роси зовнішнього середовища. Розроблено комплексну математичну модель розрахунку втрат тиску в каналах теплообмінника, що базується на аналітичній декомпозиції пневматичного тракту на характерні ділянки та враховує вплив геометрії каналів, режиму руху повітря й еквівалентної шорсткості зволожених поверхонь.*

*На основі чисельного аналізу встановлено закономірності впливу площі поперечного перерізу, кількості каналів і швидкості повітряного потоку на загальний аеродинамічний опір і необхідну потужність вентиляційного обладнання. Доведено, що форма поперечного перерізу каналів має вирішальний вплив на енергоефективність теплообмінника. Показано, що використання каналів круглого перерізу забезпечує зниження сумарних втрат тиску до 26 % порівняно з квадратними та трикутними каналами за однакових площ перерізу. Отримані результати можуть бути використані для оптимізації конструктивних параметрів побічно-випарних теплообмінників та підвищення енергоефективності систем кондиціювання і вентиляції тваринницьких приміщень.*

***Ключові слова:*** *побічно-випарне охолодження, цикл Майсоценка, теплообмінник, втрати тиску, аеродинамічний опір, форма каналу, енергоефективність, мікроклімат.*

***Ф. 15. Рис. 11. Літ. 23.***

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