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ATC Enhancement in Restructured Power
System using Whale Optimization Algorithm
Page 34

Contents

01	Grzegorz KICANA, Andrzej MICHALSKI, Zbigniew WATRAL - Simple irrigation control system used in small-scale retention	1
02	Karol WRÓBEL, Kacper ŚLESZYCKI - Robust states controller for two mass system	9
03	Bahri PREBREZA, Nuri BERISHA, Bashkim STATOVCI - Analysis of switching overvoltages and protection from atmospheric overvoltages for 400kV switchgears in the Kosovo Power System using ATP/EMTP	13
04	Tarik MOHAMMED CHIKOUCHE, Kada HARTANI, Mohamed MANKOUR - Model Predictive Direct Power Control with Duty Cycle Control of PWM Rectifier	18
05	Chahinez BELDJAATIT, Toufik SEBBAGH, Hocine GUENTRI - Discrete Wavelet Transform and Energy Distribution for Effective Bearing Fault Detection and Analysis	24
06	Samer S. Wahdain, A. I. Mohamed, Mohd Herwan Sulaiman, Mohd Razali Daud, Raja Mohd Taufika Raja Ismail - Analysis of electric field behaviour for wind turbine blades under the influence of various gas	29
07	Sibbala Bhargava Reddy, R. Ashok Kumar, G. Sreenivasan - ATC Enhancement in Restructured Power System using Whale Optimization Algorithm	34
08	Oussama Djaidja, Hemza Mekki, Samir Zeghleche, Ali Djerioui - Power Quality Enhancement in Double Fed Induction Generator Using Iterative Learning Control	41
09	Azdiana MD YUSOP*, Mohamad Nazrul NAZRIN, Ahmad Nizam MOHD JAHARI@JOHARI, Noor Asyikin SULAIMAN, Khairun Nisa KHAMIL, Ramizi MOHAMMED, Juwita MOHD SULTAN - An Evaluation Of Wireless Real-Time Data Of Solar Tracking System	49
10	Wittawat POONTHONG, Toshifumi YUJI, Toshio BOUNO, Tanes TANITTEERAPAN, Narong MUNGKUNG, Apidat SONGRUK, Charathip CHUNKUL, Somchai ARUNRUNGRUSMI, Noritsugu KAMATA, Sarizam MAMAT, Shinichi HARADA⁵ - Suggestion on Electrical Courses by Survey Data for Career Education in Thailand Community College	54
11	Nattakun POUNGPRAKHON, Chanchai THONGSOPA, Samran SANTALUNAI, Thanaset THOSDEEKORAPHAT, Nuchanart SANTALUNAI, Pichaya CHAIPANYA - The Study of Water Reconditioning using Magnetic Field for Plant Industry	59
12	Raghad H. Saeed, Farah N. Yaseen, Lubna M. Saeed, Farhad Mahmood - Vehicle Accessibility Using RFID Technology	65
13	Syarifuddin Nojeng, Reny Murniati² - Improving of Transmission Cost Allocation Method to Accelerate the Investment Recovery for Transmission lines in Deregulated Power System	69
14	Fahrizal, Muhidin Arifin, Lukman Hakim Nasution, Anuar Ahmad - Experimental matrix palm oil empty fruit bunch composite concrete K300 (POEFB-cc K300) as a reinforcement of concrete road structure	74
15	Sri SUWASTI, Muhammad Ruswandi DJALAL - Design of Continuous Water Heater Hybrid Solar And Gas System	78
16	Chandra BUANA, Muhammad Ruswandi DJALAL, Ikram IKRAM, Muh. IQBAL, La Ode MUSA, Lewi LEWI - Performance Analysis of Micro Hydro Power Plants Using a Pelton Turbine with Two Nozzle Variations	84
17	Thanpitcha Atiwanwong, Adirek Jantakun, Adisak Sangsongfa - Comparative Analysis of Optimization Value Between Artificial Neural Network and Long Short-Term Memory for Prediction Particulate Matter (PM2.5) in Bangkok Thailand	89
18	N. A. Shairi, F. N. M. Yasin, A. Othman, Z. Zakaria, I. M. Ibrahim, H. A. Majid, M. H. Jamaluddin, A. M. Ibrahim - Design and Analyses of Reconfigurable Dumbbell-Shaped and Modified H-Shaped DGSs in Millimeter Wave Band	96
19	Dhaouadi GUIZA, Djamel OUNNAS, Youcef SOUFI, Naoual TIDJANI - Design and Real Hardware Implementation of Fuzzy Logic Controller for DC-DC Boost Converter	101
20	Brahim CHEROUATI, Mohamed SENOUCI - Patient specific channel optimization using entropy and CNN deep learning for epileptic seizure prediction	106
21	Khadidja DAHLI, Nawal CHEGGAGA - A New Solar Tracking Strategy to Upgrade Solar Power Efficiency using ZIP codes as an Alternative to Sensors	111
22	Krzysztof OPRZĘDKIEWICZ - Numerical properties of discrete approximations of an elementary fractional order transfer function	117
23	Marek Przybylski - Performance and Losses Measurements of Switched Reluctance Motors with Powder and Laminated Magnetic Cores	124
24	Damian TERLECKI, Teodora Dimitrova-GREKOW, Jacek GREKOW - Indoor Localisation Based on Wi-Fi Infrastructure	131
25	Justyna HERLENDER, Jan IŻYKOWSKI - Localizing faults in power transmission line with applying signals of directional elements at both line ends	135

Contents

26	Leszek NOWOSIELSKI, Bartosz DUDZIŃSKI, Michał NOWOSIELSKI, Aleksandra ŚLUBOWSKA - Recognition of Convolutional Codes	140
27	Sławomir KOZAK - The influence of the mass of copper plates on the parameters of a superconducting surge current limiter	147
28	Mikołaj Koszel, Piotr Grzejszczak, Kornel Wolski, Tomasz Świąchowicz¹, Bartosz Nowatkiewicz- Development of a high-efficiency bidirectional grid converter designed for a DC microgrid	152
29	Wiesława MALSKA - The use of the multiple regression model in the aspect of the analysis of the thermal load capacity of the 110kV overhead transmission line	158
30	Markiyan NAKONECHNYI, Orest IVAKHIV, Yurii NAKONECHNYI, Roman VELGAN, Jolanta PLEWAKO – Modelling of an electric drive based on a DC motor and variable load influence examination	163
31	Krzysztof SOŁTYS, Sebastian BARTEL, Krzysztof KLUSZCZYŃSKI - Laboratory method of producing ferromagnetic powders and flexible magnetic materials and experimental studies of their behavior in a 3D electromagnetic chamber	169
32	Szymon CZERWIŃSKI, Mariusz KUCHAREK, Łukasz WALAS, Karol MAKOWIECKI, Przemysław WISZNIEWSKI - A system of sensors for detecting and predicting damage to the running gear of a freight wagon of the rolling stock	176
33	Krzysztof KRĘCISZ, Dawid BĄCZKOWICZ, Adam ŁYSIAK, Mirosław SZMAJDA, Aleksandra KAWALA-STERNIUK - Correlation between Linear and Non-Linear Vibroarthrographic Parameter	180
34	Amier Hafizun Ab. RASHID, Badrul Hisham AHMAD, .Mohamad Zoinol Abidin ABD AZIZ, Nonikman HASSAN, Mazlee MAZALAN - Parametric Studies of CPW Pentagonal Sierpinski Gasket Fractal Patch Antenna	186
35	Nur HAMZAH, Firman FIRMAN, Muhammad Ruswandi DJALAL - Performance of Solar Panels on Spandex and Asbestos Roofs	192
36	Mateusz SZABLICKI, Piotr RZEPKA, Adrian HALINKA - The concept of modifying the measurement algorithm for transmission line distance protection	199
37	Stanisław BEDNAREK - Project of a stratospheric photovoltaic power station	203
38	Wojciech TRZASKO, Jarosław WIATER, Renata MARKOWSKA - EMC-LabNet: Laboratory of High Voltage Technique named after prof. Andrzej Sowa, DSc PhD Eng. at Białystok University of Technology	209
39	Mateusz Feldzensztajn, Jerzy Pluciński, Sebastian Siedlecki¹, Adam Mazikowski - 16-channel programmable horticulture LED module controlled by Raspberry Pi	215
40	Iztok BRINOVAR, Benjamin BOŽIČ, Gregor SRPČIČ, Amor CHOWDHURY, Sebastijan SEME, Bojan ŠTUMBERGER, Miralem HADŽISELIMOVIĆ - Three-phase full-wave diode rectifier with DC voltage stabilization	220
41	Blagoja MARKOVSKI, Leonid GRCEV, Vladimir GJORGIEVSKI, Bodan VELKOVSKI, Marija MARKOVSKA DIMITROVSKA - Parametric Analysis of Conductive Coupling of Transmission Line Tower Grounding and Pipeline in Multilayer Soil	223
42	Riad Dib, Yassine Bensafia, Khatir Khettab - "Fractionalization": A new approach for comparing different approximation methods of fractional order systems and disturbances Rejection in PID Control	227
43	Róbert Štefko, Zsolt Čonka, Michal Kolcun, Viktor Jurák, Judith Pálfi - Communication research of protective relays for microgrids and active distribution networks	232
44	SAUDI MOHAMMED. BENGUESMIA HANI, CHOUDER AISSA - Efficient Deadbeat Control of Single-Phase Inverter with Observer for High Performance Applications	237
45	Enaam ALBANNA, Alya H. AL-RIFAIE, Ahmed A. Abdullah AL-KARAKCHI - High impedance Fault Detection in Low Voltage Overhead Distribution based Wavelet and Harmonic Indices	243
46	Hryhorii KALETNIK, Vitalii YAROPUD - Research of pressure losses and justification of forms of side-evaporative heat exchangers channels in livestock premises	247
47	Hazura HAROON, Noor Erina AZMAN, Siti Khadijah IDRIS @ OTHMAN, Hanim ABDUL RAZAK, Anis Suhaila MOHD ZAIN, Fauziyah SALEHUDDIN. Maisara OTHMAN - Fiber Optic Sensor Based on Lateral Offset Displacement for Water Quality Analysis in Agricultural Applications	253
48	Mostefa GAMRA, Hocine GUENTRI, Tayeb ALAOU³, Abdelkader CHAKER - Modelling and an adaptive fuzzy logic controller of solar thermal power plant	257
49	Hanim ABDUL RAZAK, Khairul Anuar AMINODDIN, Hazura HAROON, Siti Khadijah IDRIS, Anis Suhaila MOHD ZAIN, Fauziyah SALEHUDDIN., Ahmad Mubasyir ABDUL GHAFAR - Modelling Of Different Tapered Structures of Multimode Interference (MMI) Couplers	264
50	Haider Alrudainy, Muaad Hussein, Afrah Abood Abdul Kadhim - An Approach for Designing FSSs with Different Response	268
51	Jerzy HICKIEWICZ, Piotr RATAJ, Przemysław SADŁOWSK - SEP Patron of the Year 2023 Professor Tadeusz Malarski (1883-1952)	272

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Research of pressure losses and justification of forms of side-evaporative heat exchangers channels in livestock premises

Abstract. As a result of the numerical simulation of the indirect-evaporative air heat exchanger in the Star CCM+ software package, the distribution of the temperature field, the vector field of velocities and the absolute air humidity in channels of different shapes (square, equilateral triangle, circle) was established. The calculated coefficient of thermal efficiency of the heat exchanger with triangular channels is the whitest in contrast to the square and round forms.

Streszczenie. W wyniku symulacji numerycznej pośrednio-wyparnego powietrznego wymiennika ciepła w pakiecie Star CCM+ uzyskano rozkład pola temperatury, pola wektorowego prędkości oraz bezwzględnej wilgotności powietrza w kanałach o różnych kształtach (kwadrat, trójkąt równoboczny, koło) powstało. Obliczony współczynnik sprawności cieplnej wymiennika ciepła z kanałami trójkątnymi jest najbielszy w porównaniu do form kwadratowych i okrągłych (*Badanie strat ciśnienia i uzasadnienie kształtów kanałów bocznych wymienników ciepła w obiektach inwentarskich*)

Keywords: microclimate, livestock premises, ventilation, heat exchanger, parameters, research, dependencies. **Słowa kluczowe:** mikroklimat, wymienniki ciepła

Introduction

Rapid growth of global power consumption in livestock complexes has caused serious concern about depletion of power resources. Increasing power consumption by animal husbandry complexes is caused by such factors as increasing number of animal populations and toughening requirements to microclimate maintenance in the premises [1].

The livestock sector of agro-industrial production has the greatest potential for increasing the efficiency of power use [2]. It can be seen that power used for air cooling represents a significant part of total power consumption, which is constantly growing due to increased requirements to optimal microclimate maintenance in livestock premises [3, 4, 5].

The largest share of the power consumption in animal husbandry premises falls on generation of standard microclimate parameters, in particular, on the heating of supply ventilation air. In the heating period, heat-generating devices of these premises consume, according to various estimates, from 40 to 90% of the total cost of fuel and power resources [1]. Therefore, even partial reduction of these costs will lead to a significant reduction in costs required for manufacture of animal husbandry products.

An effective way to reduce power consumption in livestock premises is to use the heat of ventilation emissions to heat supply ventilation air. The difficulty of using the air heat of ventilation emissions is that the exhaust air is a low-potential source of thermal power [6-9].

The most promising ways of using the heat of ventilation emissions are the use of heat exchangers (heat utilizers). Due to their efficiency, heat exchangers (heat utilizers) of ventilation emissions are becoming increasingly widespread both in residential and administrative premises, as well as in industrial buildings. The use of a heat exchanger (heat utilizer) of ventilation emissions in the system of microclimate maintenance in livestock premises allows reducing power consumption for supply air heating by up to 80% [10].

Over the past two decades, many new renewable power-based devices were introduced for the purpose of heating the agricultural industry premises: new heat recovery units, heat pumps, solar systems and many others

[11, 12, 13]. However, no renewable power-based devices have been widely applied in the field of cooling until now.

Automated ventilation system was created to remove air from piggery premises [14, 15, 16]. As a result of analytical studies of this system, the condition for its effective operation was mathematically represented [16, 17].

The factors causing difficulties in utilization of heat of ventilation emissions in livestock premises include [18-21]:

- significant air dustiness (up to 6 mg/cu m³);
- high air humidity in the premises, reaching 80% if standard parameters of air environment are observed;
- presence of high concentrations of aggressive gases in the air: ammonia – up to 20 mg/cu m³, hydrogen sulfide – up to 10 mg/cu m³, carbon dioxide – up to 0.28%;
- unacceptability of even partial recirculation of exhaust air for most livestock premises;
- significant amount of technological equipment, which is characteristic of modern animal husbandry premises and air exchange arrangement pattern determined thereby.

There are no such difficult conditions in whatever branch of national economy for use of heat exchangers (heat utilizers) as in agricultural production. Therefore, creation of workable and cost-effective designs of heat-utilizers for livestock premises, capable of being aggregated with a set of ventilation equipment, is a complex scientific and engineering task.

Mathematical Foundation

Heat exchanger of the side-evaporative type have a system of channels, which is shown in Fig. 1. The heat exchanger contains independent working channels and connected wet and dry channels.

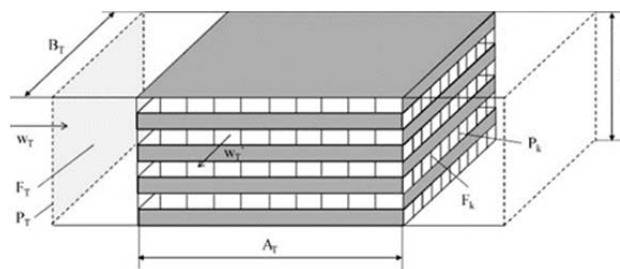


Fig. 1. Design diagram of side-evaporative heat exchanger P

Let's first consider the air flow through working channels. The air passes through three sections. The first section is distribution of air flow from large-diameter air duct into channels of a smaller diameter. Pressure losses are calculated using the formula

$$(1) \quad \Delta p_{Tk} = \eta_1 \left(1 - \frac{N_k F_k}{F_T}\right) \frac{\rho (w_T)^2}{2}$$

where η_1 – is impact mitigation factor, $\eta_1 = 0.5$ [22]; N_k – number of channels; F_T – cross-sectional area of the inlet duct, sq m; F_k – cross-sectional area of the channel, sq m; w_T – is the average air velocity in working channels of side-evaporative heat exchanger.

The second section is air flow movement through the channels. Pressure losses are calculated using the formula

$$(2) \quad \Delta p_k = 0,11 N_k \frac{P_k A_T \rho w_T^2}{4 F_k} \sqrt{\frac{17 \mu_a P_k}{F_k \cdot w_k \cdot \rho} + \frac{\psi_k P_k}{4 F_k}}$$

Where P_k – is the perimeter of the channel section, sq m; A_T – the length of the working channel, sq m; ψ_k – equivalent roughness of the working channel walls [23, 24]; μ_a – dynamic air viscosity, $\mu = 18.27 \cdot 10^{-6}$ N·s/sq m [25].

The third section is merging of the air flow from channels into the large-diameter duct. Pressure losses are calculated using the formula

$$(3) \quad \Delta p_{kT} = \eta_2 \left(1 - \frac{N_k F_k}{F_T}\right)^2 \frac{\rho (w_T)^2}{2}$$

Where η_2 – is impact mitigation factor, $\eta_2 = 0.5$ [23, 24].

The next step is to consider air movement through connected wet and dry ducts. At the same time, air passes through five areas. The first section is separation of air flow from the large-diameter air duct into channels of a smaller diameter. Pressure losses Δp_{Tk} are calculated using formula (1). The second section is air flow movement through dry channels. Pressure losses Δp_k are calculated using formula (2). The third section is pressure loss in a spatial (circular) turn by 180° Δp_{k180} [18]

$$(4) \quad \Delta p_{k180} = \zeta_{k80} \frac{\rho (w_T)^2}{2}$$

Where ζ_{k180} – is local resistance coefficient for the spatial (circular) turn by 180°, according to [23] $\zeta_{k180} = 2$.

The fourth section is air flow movement through wet channels. Pressure losses Δp_k are calculated using formula (2). The fifth section is merging of the air flow from the wet ducts into the large-diameter air duct. Pressure losses Δp_{kT} are calculated using formula (3).

Total pressure losses through side-evaporative heat exchanger are equal to the sum of the pressure losses in all the above-mentioned sections. Assuming the same number N_k and size (area F_k and perimeter P_k) of working, dry and wet channels, we obtain a formula for calculation of air losses in side-evaporative air heat exchanger

$$(5) \quad \begin{aligned} \Delta \Delta p_T &= \Delta p_{Tk} + \Delta p_k + \Delta p_{kT} + \Delta p_{Tk} + \Delta p_k + \Delta p_{k180} \\ &+ \Delta p_k + \Delta p_{kT} = \\ &= \frac{\rho (w_T)^2}{2} \left[\eta_1 \left(1 - \frac{N_k F_k}{F_T}\right) + 0,11 N_k \frac{P_k A_T}{4 F_k} \times \right. \\ &\quad \left. \sqrt{\frac{17 \mu_a P_k}{F_k \cdot w_T \cdot \rho} + \frac{\psi_k P_k}{4 F_k}} + \eta_2 \left(1 - \frac{N_k F_k}{F_T}\right)^2 \right] \\ &\quad + \frac{\rho (w_T)^2}{2} \left[\eta_1 \left(1 - \frac{N_k F_k}{F_T}\right) + \right. \\ &\quad \left. + 0,11 N_k \frac{P_k B_T}{4 F_k} \sqrt{\frac{17 \mu_a P_k}{F_k \cdot w_T \cdot \rho} + \frac{\psi_k P_k}{4 F_k}} + \zeta_{k80} + \right. \\ &\quad \left. + 0,11 N_k \frac{P_k B_T}{4 F_k} \sqrt{\frac{17 \mu_a P_k}{F_k \cdot w_k \cdot \rho} + \frac{\psi_k P_k}{4 F_k}} \right. \\ &\quad \left. + \eta_2 \left(1 - \frac{N_k F_k}{F_T}\right)^2 \right] \end{aligned}$$

Where B_T – is the length of dry and wet channels, sq m; ψ_k – equivalent roughness of the walls of dry and wet channels [23]; w_T – average air velocity in the dry and wet channels of side-evaporative heat exchanger, m/s.

Let's plot the dependence between pressure losses through side-evaporative heat exchanger Δp_T and the number of channels N_k , the channels' cross-sectional area F_k , the average air velocity in the channels under condition $w_T = w_T$ and their shape (square, equilateral triangle, circle). To do this, we will use the dependence between perimeters of channels of different shapes and their area:

- square:

$$(6) \quad P_k = 4\sqrt{F_k}$$

- equilateral triangle:

$$(7) \quad P_k = 2\sqrt{F_k\sqrt{3}}$$

- circle:

$$(8) \quad P_k = 2\sqrt{\pi F_k}$$

Dependence (5) and calculated power required for air pumping through side-evaporative heat exchanger are calculated using the formula

$$(9) \quad N_{W1} = \frac{V \Delta p_T}{\eta_n}$$

Where η_n is total efficiency of the ventilator, $\eta_n = 0.8$ [22] shown in fig. 2-3.

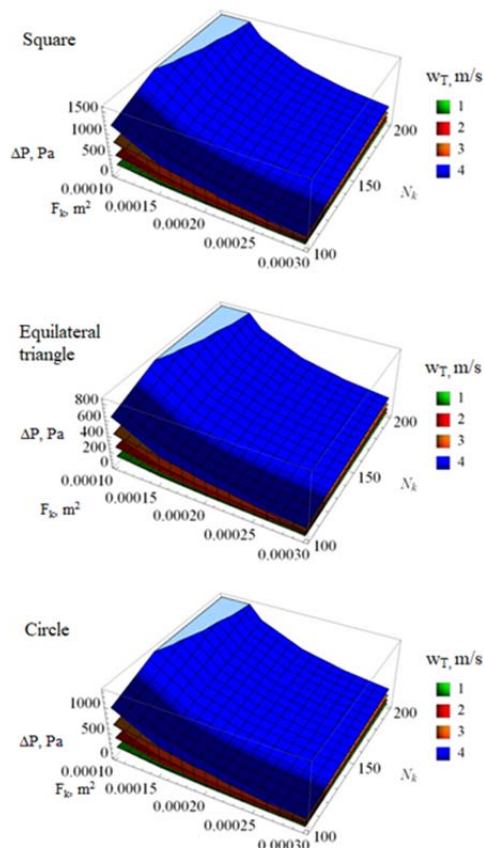


Fig. 2. Dependence between pressure losses Δp_T and the number of channels N_k , the channels' cross-sectional area F_k , the average air velocity in the channels under condition $w_T = w_T$ and their shape (square, equilateral triangle, circle)

Analysis of fig. 2-3 allows stating that with the increase in the channels' cross-sectional area F_k , there occurs decrease in pressure losses Δp_T and power N_{W1} , which is required for air pumping through the side-evaporative heat exchanger. At the same time, the increase in the average air velocity in the channels w_T and their number N_k leads to the increase in pressure losses Δp_T and power N_{W1} . The

most effective form of channels (pressure loss reduction by 23%) in terms of pneumatic losses is an equilateral triangle.

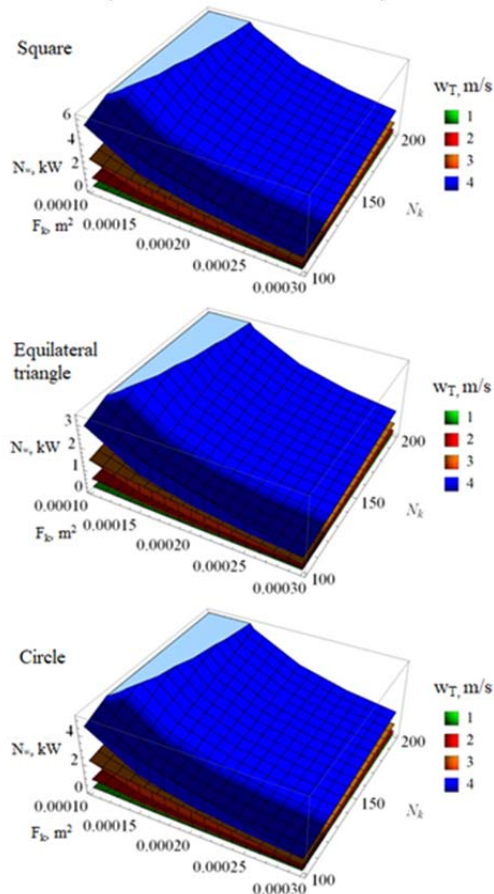


Fig. 3. Power N_{w1} dependence on the number of channels N_k , the cross-sectional area of the channels F_k , the average air velocity in the channels under condition $w_T = w_T$ and their shape (square, equilateral triangle, circle).

Numerical Methods

The main criteria for choosing recuperative heat exchangers (heat utilizers) are [11, 22]:

- thermal efficiency ratio, which is determined using formula [22, 25]:

$$(10) \quad \eta_t = \frac{T_{k2} - T_{n2}}{T_{n1} - T_{k1}}$$

Where T_{k2} – is the temperature of supply air at the outlet from the heat exchanger (heat recovery unit), °C;

T_{n2} – is the temperature of supply (external) air at the inlet to the heat exchanger (heat utilizer), °C;

T_{n1} – is the temperature of exhaust air at the inlet to the heat exchanger (heat utilizer), °C;

T_{k1} – is the temperature of exhaust air at the outlet from the heat exchanger (heat utilizer), °C;

- sanitary and hygienic parameters: there should be no pollutants passing through the recuperator, it is necessary to ensure the possibility of controlling air quality and its purification to the greatest possible extent;

- power efficiency: this value characterizes specific power consumption, that is, how much the recuperative heat exchanger (heat recovery unit) consumes to return the unit of heat from removed air;

- operational characteristics: the structure must be suitable for repair, have a long service life, require minimal maintenance;

- structure cost.

To study the process of heat and mass transfer in a side-evaporative heat exchanger with different channel shapes, we will conduct numerical simulations in Star

CCM+ software package. As models for the continuum grid, the following were chosen: generator of polyhedral cells, generator of surface grid and extruder of cells. The basic size of a cell was 0.001 m, with maximum ratio of the sizes of connected mesh edges being 1.3. The layout of the channels and the general view of calculated grid of the side-evaporative heat exchanger are shown in fig. 4-6.

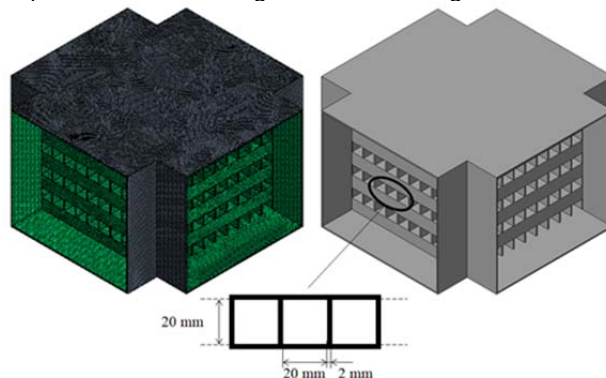


Fig. 4. Layout diagram of the channels and general view of calculated grid of side-evaporative heat exchanger with square channels

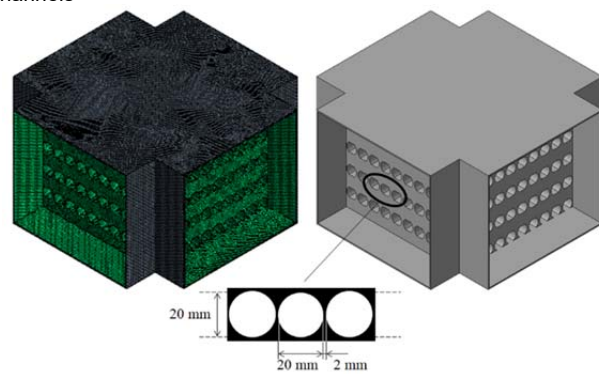


Fig. 5. Layout diagram of the channels and general view of calculated grid of side-evaporative heat exchanger with circular channels

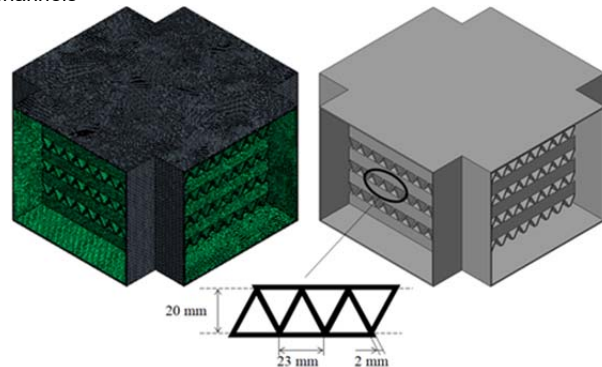


Fig. 6. Layout diagram of the channels and general view of calculated grid of side-evaporative heat exchanger with triangular channels

The following were chosen as physical models of dry and wet channels: three-dimensional, Eulerian multiphase model, method of separated flow and volume liquid VOF, model of phase interaction and model of separated multiphase temperature. The current flow was subjected to the Navier-Stokes equation and k-ε model of turbulence. The Euler phases were air and water. The air phase was subjected to MASVP-PR97 real gas (vapor) and turbulent flow models. The water phase was subjected to the Van der Waals real gas and turbulent flow models.

The following were chosen as physical models of the heat exchanger's walls: three-dimensional model of solid

body material, constant density, model of power of separated solid body.

Physiomechanical properties of all simulation phases are summarized in Table 1.

Table 1. Physiomechanical properties of modeling phases.

Property	Air phase	Water phase	Wall
Dynamic viscosity, Pa·s	$1.85508 \cdot 10^{-5}$	$1.26765 \cdot 10^{-5}$	-
Molecular weight, kg/mol	28.9664	18.0153	-
Thermal conductivity ratio, $V/(m \cdot K)$	0.0260305	0.0253325	0.44
Specific heat capacity, $J/(kg \cdot K)$	1003.62	1938,19	1700.0

Selected solver is of a stationary one. The number of internal inertias was equal to 10.

The dimensions of the side-evaporative heat exchanger and boundary conditions of modeling are shown in fig. 7. At the inlet to the heat exchanger, air flow was equal to $Q_1 = Q_2 = 100$ cu m/h, the temperature was $t_1 = 30$ °C, $t_2 = 0$ °C, absolute humidity – $x_1 = x_2 = 5$ g/kg. Heat insulation is mounted around the heat exchanger, that is, heat exchange with the environment does not occur.

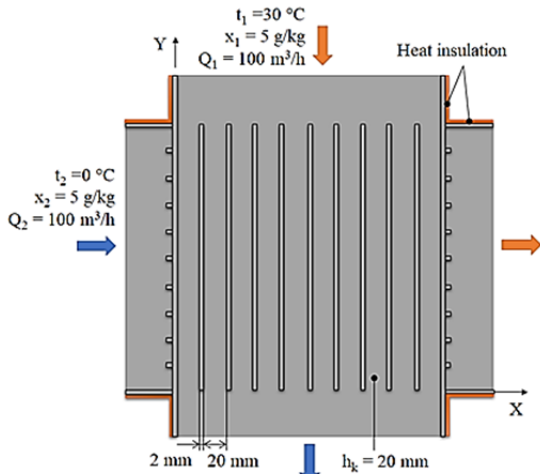


Fig. 7. Dimensions of the heat exchanger of side-evaporative cycle type and boundary conditions of modeling

Results and Discussion

As a result of modeling, distribution of the temperature field in the heat exchanger with different channel shapes was obtained (see Fig. 8-13). Transverse distribution of the temperature field is characterized by formation of cylindrical velocity distribution in the middle of the channel, regardless of its shape. In turn, longitudinal distribution of the temperature field in side-evaporative heat exchanger shows the interaction between two air flows, which change their temperature due to heat conduction through the walls.

For square ducts, the temperature of the cold air flow varies on average from 0°C to 10.3 °C, that of warm air – from 0°C to 17.7°C. Thermal efficiency ratio of the heat exchanger with square channels is $\eta_t = 0.84$.

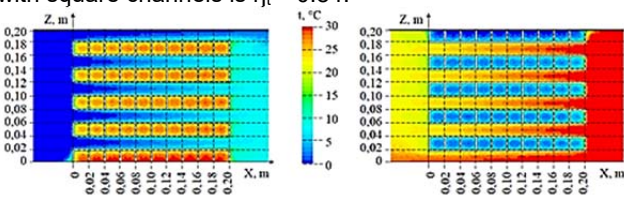


Fig. 8. Transverse distribution of the temperature field in side-evaporative heat exchanger with square channels

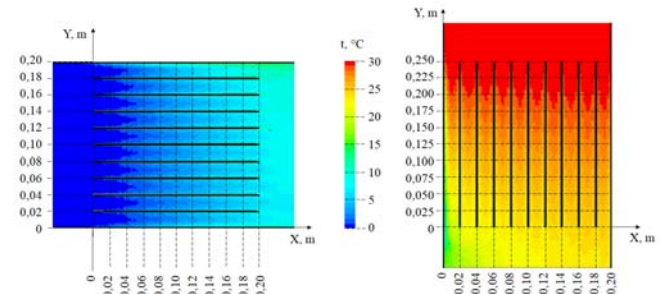


Fig. 9. Longitudinal distribution of the temperature field in side-evaporative heat exchanger with square channels

For round ducts, the temperature of cold air flow varies on the average from 0°C to 9.4°C, that of warm air – from 0°C to 17.1°C. Thermal efficiency ratio of the heat exchanger with circular channels is $\eta_t = 0.73$.

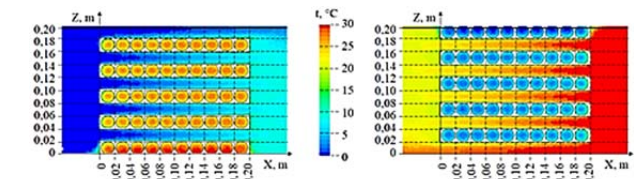


Fig. 10. Transverse distribution of temperature field in side-evaporative heat exchanger with circular channels

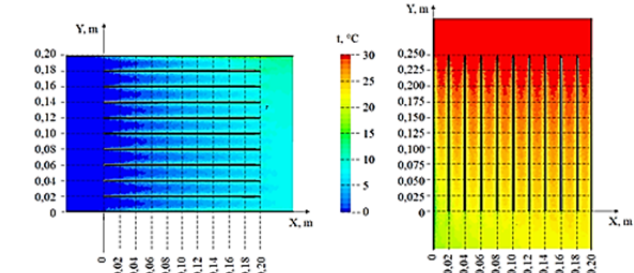


Fig. 11. Longitudinal distribution of temperature field in side-evaporative heat exchanger with circular channels

For triangular-shaped channels, the temperature of cold air flow varies on the average from 0°C to 11.5°C, that of warm air – from 0°C to 18.2°C. Thermal efficiency ratio of the heat exchanger with triangular channels is $\eta_t = 0.97$.

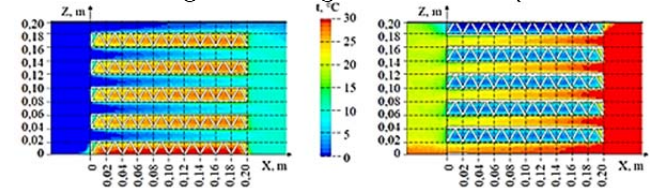


Fig. 12. Transverse distribution of temperature field in side-evaporative heat exchanger with triangular channels

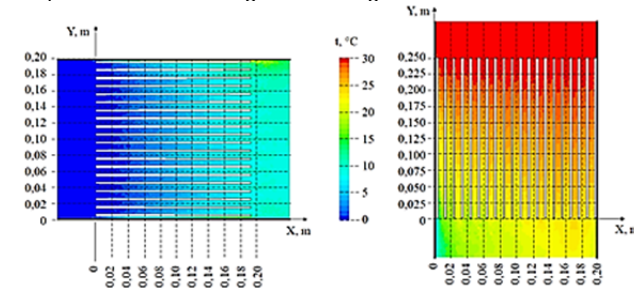


Fig. 13. Longitudinal distribution of temperature field in side-evaporative heat exchanger with triangular channels

To visualize the process of air flow movement through the channels of the heat exchanger, a vector field of velocities was constructed (see Fig. 14-16). Comparing the air flow distributions in all channels depending on their shape, we can conclude that for triangular channels, a more uniform air flow is observed over the entire cross-sectional area. The average speed is equal to $V = 0.042$ m/s, with the ratio of its variation $\delta_V = 0.88$. At the same time, for square channels – $V = 0.039$ m/s, and the ratio of variation $\delta_V = 0.84$, while for round channels – $V = 0.046$ m/s, and the ratio of variation $\delta_V = 0.81$.

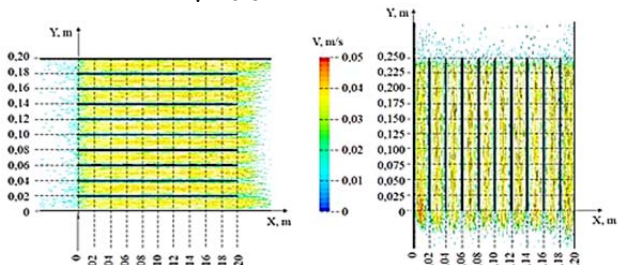


Fig. 14. Distribution of the velocity vector field in the side-evaporative heat exchanger with square channels

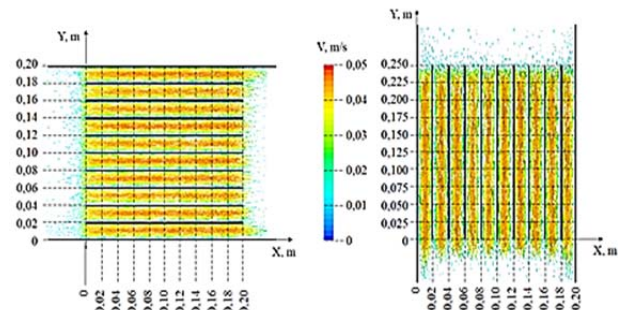


Fig. 15. Distribution of the velocity vector field in the side-evaporative heat exchanger with round channels

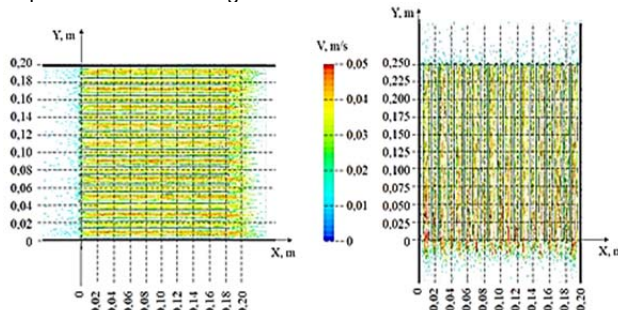


Fig. 16. Distribution of the velocity vector field in the side-evaporative heat exchanger with triangular channels

The change in air flow humidity during its movement is shown in fig. 17-19. During thermal air flow movement, a significant decrease in absolute humidity is observed to almost 0.01 g/kg. This is explained by the condensation phenomenon. Thus, cold air cools the walls of the heat exchanger. In turn, during warm air movement, the moisture contained in the flow turns into a liquid and settles on the cold walls of the heat exchanger. This phenomenon leads to decrease in the absolute humidity of warm air.

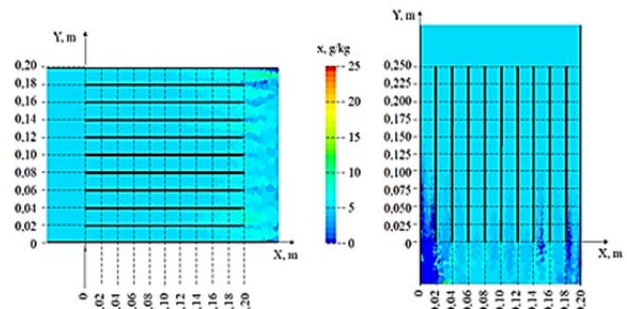


Fig. 17. Distribution of absolute air humidity in the side-evaporative heat exchanger with square channels

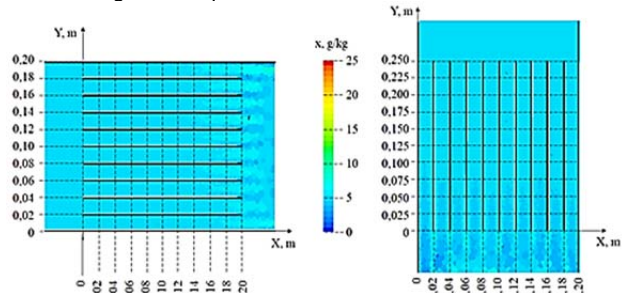


Fig. 18. Distribution of absolute air humidity in the side-evaporative heat exchanger with circular channels

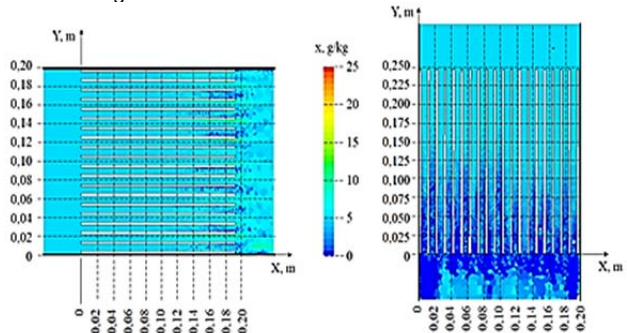


Fig. 19. Distribution of absolute air humidity in the side-evaporative heat exchanger with triangular channels

By comparing humidity distribution in channels of different shapes, one can state that decrease in the absolute humidity of thermal air flow sooner occurs in the side-evaporative type heat exchanger with triangular channels. This is evidenced by the data shown in Figure 19.

Conclusion

As a result of analytical studies of pressure losses of an indirect-evaporative air heat exchanger, the dependences of pressure losses Δp_T and power N_T on the number of channels N_k , the cross-sectional area of channels F_k , the average air velocity in the channels under the condition $w_T = w_T$ and their shape (square, equilateral triangle, circle). It was established that the most effective form of channels in terms of pneumatic losses (reduction of pressure losses by 23%) has channels in the form of an equilateral triangle.

As a result of the numerical simulation of the indirect-evaporative air heat exchanger in the Star CCM+ software package, the distribution of the temperature field, the vector field of velocities and the absolute air humidity in channels of different shapes (square, equilateral triangle, circle) was established.

The calculated coefficient of thermal efficiency of the heat exchanger with triangular channels is the biggest $\eta_t = 0.97$. For a square shape, it is $\eta_t = 0.84$, and for round – $\eta_t = 0.73$.

The average speed of the air flow in the triangular channels is $V = 0.042$ m/s, and the coefficient of its variation is $\delta_v = 0.88$. In turn, for square channels – $V = 0.039$ m/s, coefficient of variation $\delta_v = 0.84$, for round channels – $V = 0.046$ m/s, coefficient of variation $\delta_v = 0.81$.

Comparing the humidity distribution in channels of different shapes, it can be stated that the decrease in the absolute humidity of the thermal air flow from 0.05 g/kg to 0.01 g/kg occurs earlier in the heat exchanger of the indirect-evaporative type with triangular channels.

The above statements make it possible to conclude that the indirect-evaporative air heat exchanger with triangular channels is the most efficient.

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