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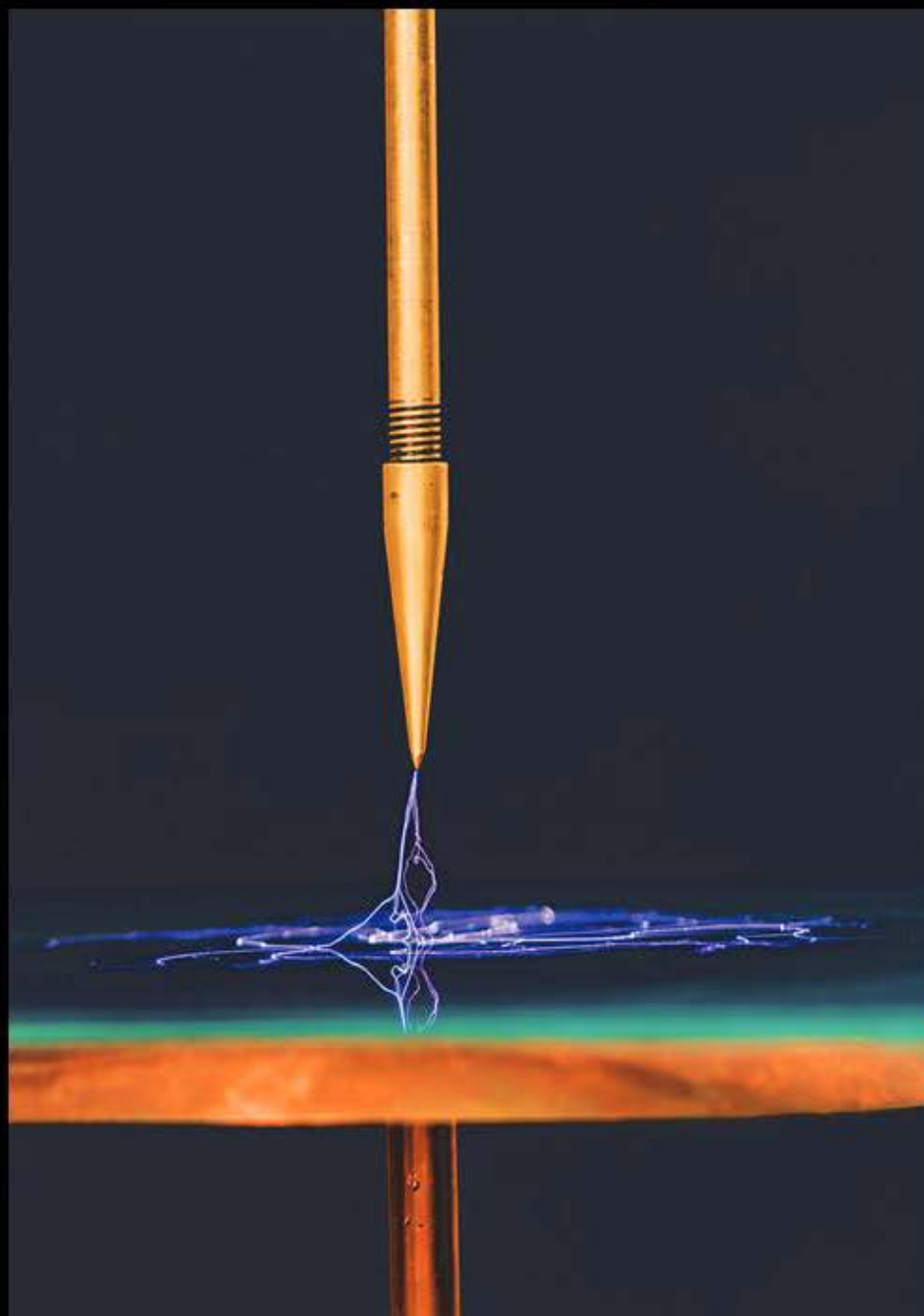
PRZEGLĄD ELEKTROTECHNICZNY

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Contents

01	Stepan KUDRIA, Petr LEZHNIUK, Oleksandr RIEPKIN, Olena RUBANENKO - Hydrogen technologies as a method of compensation for inequality of power generation by renewable energy sources	1
02	Aleksander LULA¹, Krzysztof ZAGRAJEK - A concept of using a private electric vehicle to optimize the costs of electricity in household equipped with a PV power plant	7
03	Paweł TERLIKOWSKI¹, Jolanta GALIŃSKA - Profitability assessment of connecting renewable energy micro-installations in a public utility building in technical and organizational terms	15
04	Paweł KIECZMERSKI, Krzysztof ROGOWSK - Application of SIMATIC S7-1200 controllers for the water level control system in water reservoirs	23
05	Sebastian ŁACHECIŃSKI - Storage of temporal data for the transaction time on the MariaDB platform	28
06	Andrea WEBER, Miklós KUCZMANN - TP Transformation Based Controller and Observer Design of the Inverted Pendulum	34
07	Panduranga Vemula, Rudra Sankar Dhar - Design of 8T SRAM using 14nm FINFET Technology	40
08	Somchai ARUNRUNGRUSMI, Parinda PHANPHECH, Narong MUNGKUNG, Tanes TANITTEERAPAN, Wittawat POONTHONG, Apidat SONGRUK, Khanchai TUNLASAKUN, Tanapon TAMRONGKUNANAN, Surachai THAMMAYARIT, Kongsak ANUNTAHIRUNRAT - Application of a small transformer for designing current detection and overcurrent protection circuit for laboratory tables	44
09	Salah Belkhir, Abderrahmane Ziani, Hakim Azizi, Hocine Moulai - Arc plasma energy evolvement in 60 kV network circuit breakers	51
10	Vitaly Yaropud, Ihor Kupchuk, Serhiy Burlaka, Julia Poberezhets, Ihor Babyn - Experimental studies of design-and-technological parameters of heat exchanger	57
11	Oleg GUBAREVYCH, Svitlana GOLUBIEVA, Inna MELKONOVA - Comparison of the results of simulation modeling of an asynchronous electric motor with the calculated electrodynamic and energy characteristics	61
12	Cekmas CEKDIN, Heni JUNIAR, FADILAH – A program design to form an oscillation curve and reduction with PID control on a heat exchanger	67
13	Kiagus Ahmad Roni, Taufik Barlian, Zulkifli Saleh, Bengaawan Alfaresi, Fadilah, Asri Indah Lestari, Muhammad Ustadzo - Performance of ground fault relay on Feedera incoming substation 15 short circuit in substation 14 Plaju utilities PT. Pertamina (Persero) refinery unit III Plaju – Sungai Gerong	72
14	Suthasinee LAMULTREE, Rut PANTHASA I - Investigation and experiment of a symmetrically unidirectional patterns antenna	76
15	Sastya Hendri Wibowo, Rozali Toyib, Yulia Darnita, Muntahanah - LAN Performance Evaluation Based On Access to Transmission Media	80
16	Mehmet DUMAN, Alp Oral SALMAN - LNA Design and Implementation for Passive Millimeter Wave Imaging System in Ka Band	86
17	Kamil PARFIANOWICZ - Comparative analysis of multilevel voltage source inverters used to supply a five-phase Permanent Magnet Synchronous Motor	90
18	Dominik DUDA, Paweł KUBEK, Krzysztof MAŻNIEWSKI - Analysis of the mechanics of conductors in overhead power lines with the use of computer calculation tools)	95
19	Jarosław KMAK, Wiesław NOWAK, Rafał TARKO - Statistical assessment of shock hazard during earth faults in medium voltage substations	99
20	Konrad KOCHANOWICZ, Wiesław NOWAK, Rafał TARKO - Impact range reduction of the electromagnetic fields generated by power transmission lines	103
21	Tomasz BEDNARCZYK, Wojciech KOŁTUNOWICZ, Michael KRUEGER, Udo RANNINGER - Measurement and localisation of partial discharges in the insulation of power transformers	107
22	Mateusz CYBULSKI, Piotr PRZYBYŁEK - Application of the 3A molecular sieve to improve the condition of the power transformer's insulation system	111
23	Michał LECH, Paweł WĘGIEREK, Damian KOSTYŁA - Laboratory studies of the electrical strength of vacuums obtained on the basis of air, neon and helium	115
24	Kamil LEWANDOWSKI, Hubert MORANDA - Bubble effect phenomenon in transformer solid insulation based on cellulose and aramid (Nomex [®] 926) in the mineral oil and synthetic ester Midel 7131	119
25	Aleksandra SCHÖTT-SZYMCZAK - Overvoltage phenomena in MV power cable lines according to operation of cable screens	124
26	Konrad STRZELECKI - Comparison of initiation and breakdown voltage in esters and mineral oils in the system of insulated electrodes under lightning impulse voltage	128
27	Filip STUCHAŁA - Acceleration voltage of streamers in selected dielectric liquids at lightning impulse voltage	132
28	Dominika SZCZEŚNIAK, Piotr PRZYBYŁEK, Agnieszka MARCINKOWSKA, Aneta LEWANDOWSKA - Modification of natural ester by means of C ₆₀ fullerene	136
29	Jakub GAJZICA, Wiesław NOWAK, Rafał TARKO - Lightning overvoltages analysis of cable outer sheaths in high voltage mixed overhead cable power lines	140
30	Paweł ŻYŁKA - Changes in the distribution of the effective surface charge density in the model gas cavity	144
31	Bartłomiej PASTERNAK, Paweł RÓZGA - Initiation of electrical discharges in the oil wedge system at lightning impulse voltage	148
32	Marek ANDRZEJEWSKI, Wiesław GIL, Wiktor MASŁOWSKI - Measurements quality in on-line bushing monitoring	152
33	Ewa ZAWADZKA, Henryk BRZEZIŃSKI, Krzysztof KOGUT, Patrycja SIWORSKA – ŻUREK - Recycling of electrical power products	157
34	Grzegorz MASŁOWSKI, Robert ZIEMBA - Voltage waveforms induced in power lines by nearby lightning strikes	161
35	Krzysztof KOGUT, Ewa ZAWADZKA, Henryk BRZEZIŃSKI - The solid insulated busbar with the properties to reactive power compensation	165
36	Jarosław GIELNIAK, Magdalena CZERNIAK, Kamil LEWANDOWSKI - Investigation of vibrations of transformers with a rated power of 16, 25 and 40 MVA in terms of assessing their technical condition	169
37	Wiktor KALUS, Jarosław ZYGARLICKI, Łukasz NAGI, Michał KOZIOŁ - The problem of ozone generation in electroadhesion devices	173

Contents

38	Damian KOSTYŁA, Paweł WĘGIEREK, Michał LECH - Fiber optic technology for pressure measurements in high voltage switchgear	177
39	Michał MOLAS, Marcin SZEWCZYK - Measurements of the trajectory of a long spark in three-dimensional space	181
40	Jacek RYBARZ, Sebastian BORUCKI, Michał KUNICKI, Andrzej CICHON - Assessment of the correctness of installation of cable accessories in MV cables with XLPE insulation in terms of partial discharges	185
41	Daria WOTZKA, Mirosław GRYSZPIŃSKI, Ireneusz URBANIEC - Study of UV radiation emitted by partial discharge on the surface of a composite insulator	189
42	Andrzej CICHON, Sebastian BORUCKI, Michał WŁODARZ - The use of selected machine learning classifiers to recognize forms of partial discharge	193
43	Michał KUNICKI - Analysis of the variability in time of signals generated by partial discharges	197
44	Krzysztof WINCENCIK, Jarosław WIATER - Spark-over in high voltage cables used for lightning protection	201
45	Konstanty M. GAWRYLCZYK - Comparison of circuit models used for modeling of frequency analysis response of transformer	205
46	Jacek DZIURA - Liquid transformers of higher thermal classes – a case study	209
47	Andrzej MROZIK - Partial discharge detection in dry medium voltage transformers	212
48	Hubert ŚMIETANKA, Krzysztof WIECZOREK, Przemysław RANACHOWSKI, Zbigniew RANACHOWSKI, Adam BRODECKI - Degradation processes in textolite elements of surge arresters	216
49	Stefan WOLNY - Analysis of the impact of the degree of thermal degradation of NOMEX®910 semi-synthetic insulation impregnated with synthetic ester on the dispersion characteristics of the loss factor in the low and high frequency domain	221
50	Marek ZENKER - Influence of temperature on the dielectric response in the frequency domain of aramid paper impregnated with exploited and regenerated synthetic ester	225
51	Szymon BANASZAK, Michał ZENCZAK, Olgierd MAŁYSZKO, Jakub DOWEJKO, Jarosław JAWORSKI - The selection of installation place of the hydrogen energy buffer in the electric power system	229
52	Michał KOZIOL, Jarosław ZYGARLICKI, Dariusz ZMARZŁY, Elżbieta JANOWSKA-RENKAS – Problems of measuring the electrical parameters of geopolymer concretes	233
53	Aleksandra RAKOWSKA¹, Krzysztof SIODŁA - Sulfurhexafluoride and environmental protection	237
54	Krzysztof A. BOGDANOWICZ - Any-weather solar cells – possibilities and prospective applications	241
55	Tomasz MARCINIAK, Kacper PODBUCKI, Jakub SUDER I - Application of the Nucleo STM32 module in teaching microprocessor techniques in automatic control	245
56	Kinga KONDRACKA, Piotr FIREK, Magdalena JAWOROWSKA, Mariusz SOCHACKI - Technology of modified structures ISFET for detection of avidin	249
57	Maciej SIBIŃSKI, Grażyna BUDZIŃSKA - European Project Semester – The Model of International Interdisciplinary Electronics Education	253
58	Patrycja ŚPIEWAK, Michał WASIAK, Robert P. SARZAŁA - Computer analysis of capacitance phenomena in nitride VCSELs	257
59	Mateusz ZAPART, Cezary WOREK - Simulation analysis of the SOGI-FLL discrete model including behavioral modeling of the microcontroller peripheral	261
60	Przemysław CZARNECKI, Łukasz RUTA, Katarzyna ZNAJDEK, Ewa RAJ, Zbigniew LISIK - Prototype of a hybrid photovoltaic panel with a cooling plate	267
61	Zenon KIDON - Posturography stand for infant assessments	271
62	Michał MOLAS, Marcin SZEWCZYK - Review of current research trends in stochastic methods of simulations of electrical discharges	275
63	Paweł Pawłowski, Adam Dąbrowski, Agnieszka Stankiewicz, Karol Piniarski – Teaching of the basic of electronics in remote and hybrid modes	280
64	Renata Majgier, Ewa Mandowska, Wojciech Grzesiak, Arkadiusz Mandowski - A ceramic thick layer substrate as a material for retrospective dosimetry	284
65	Chai Wankan, Krittanon Prathepha, Worawat Sa-ngiamvibool - A Fully-Balanced Current-Tunable All-Pass Filter with CAPRIO Technique	288
66	Hassen BELILA, Nasserdine BOUDJERDA, Imen BAHRI, Ahsene BOUBAKIR - Robust Hybrid Control Strategy PI-Sliding Mode Control of a STATCOM in the Presence of a Decentralized PV Source	292
67	Jacek KOZYRA, Zbigniew ŁUKASIK, Aldona KUŚMIŃSKA-FIJAŁKOWSKA, Paweł KASZUBA - An analysis of remote voltage measurement in the medium voltage cable networks	299
68	Andrzej Ł. Chojnacki - Impact of ambient temperature on the failure intensity of overhead MV power lines	307
69	Stanisław BEDNAREK - The acoustic pulsar controlled by electromagnets	312

Spis treści

01	Stepan KUDRIA, Petr LEZHNIUK, Oleksandr RIEPKIN, Olena RUBANENKO - Technologie wodorowe jako metoda kompensacji nierówności wytwarzania energii przez odnawialne źródła energii	1
02	Aleksander LULA¹, Krzysztof ZAGRAJEK - Koncepcja wykorzystania prywatnego pojazdu elektrycznego do optymalizacji kosztów zużycia energii elektrycznej dla gospodarstwa domowego wyposażonego w instalację fotowoltaiczną	7
03	Paweł TERLIKOWSKI¹, Jolanta GALIŃSKA - Ocena opłacalności przyłączenia mikroinstalacji OZE w budynku użyteczności publicznej w ujęciu technicznym i organizacyjnym	15
04	Paweł KIECZMERSKI, Krzysztof ROGOWSK - Zastosowanie sterownika SIMATIC S7-1200 do regulacji poziomu wody w zbiornik	23
05	Sebastian ŁACHECIŃSKI - Składowanie danych temporalnych dla wymiaru czasu transakcyjnego na platformie MariaDB	28
06	Andrea WEBER, Miklós KUCZMANN - Transformatorowy sterownik TP i konstrukcja obserwatora odwróconego wahadła	34
07	Panduranga Vemula, Rudra Sankar Dhar - Konstrukcja 8T SRAM przy użyciu technologii 14nm FINFET	40
08	Somchai ARUNRUNGRUSMI, Parinda PHANPHECH, Narong MUNGKUNG, Tanes TANITTEERAPAN, Wittawat POONTHONG, Apidat SONGRUK, Khanchai TUNLASAKUN, Tanapon TAMRONGKUNANAN, Surchai THAMMAYARIT, Kongsak ANUNTAHIRUNRAT - Zastosowanie małego transformatora do projektowania układu detekcji prądu i zabezpieczenia nadprądowego dla stołów laboratoryjnych	44
09	Salah Belkhir, Abderrahmane Ziani, Hakim Azizi, Hocine Moulai - Ocena energii łuku plazmowego w wyłącznikach sieciowych 60 kV	51
10	Vitaly Yaropud, Ihor Kupchuk, Serhiy Burlaka, Julia Poberezhets, Ihor Babyn - Badania eksperymentalne parametrów konstrukcyjno-technologicznych wymiennika ciepła	57
11	Oleg GUBAREVYCH, Svitlana GOLUBIEVA, Inna MELKONOVA - Porównanie wyników modelowania symulacyjnego asynchronicznego silnika elektrycznego z obliczonymi charakterystykami elektrodynamicznymi i energetycznymi	61
12	Cekmas CEKGIN, Heni JUNIAR, FADILAH – Projekt programu do tworzenia krzywej oscylacji i redukcji z regulacją PID na wymienniku ciepła	67
13	Kiagus Ahmad Roni, Taufik Barlian, Zulkifli Saleh, Bengaawan Alfaresi, Fadilah, Asri Indah Lestari, Muhammad Ustadzo - Działanie przekaźnika ziemnozwarciowego na podstacji przychodzącej Feedera 15 Zwarcie w podstacji 14 Plaju media PT. Rafineria Pertamina (Persero) III Plaju – Sungai Gerong	72
14	Suthasinee LAMULTREE, Rut PANTHASA I - Badanie i eksperyment symetrycznie jednokierunkowej anteny pattern	76
15	Sastya Hendri Wibowo, Rozali Toyib, Yulia Darnita, Muntahanah - Ocena wydajności sieci LAN na podstawie dostępu do mediów transmisyjnych	80
16	Mehmet DUMAN, Alp Oral SALMAN - Projekt i wdrożenie LNA dla pasywnego systemu obrazowania fal milimetrowych w paśmie Ka	86
17	Kamil PARFIANOWICZ - Analiza porównawcza wielopoziomowych falowników napięcia stosowanych do zasilania pięciofazowego silnika synchronicznego z magnesami trwałymi PMSM	90
18	Dominik DUDA, Paweł KUBEK, Krzysztof MAŻNIEWSKI - Analiza mechaniki przewodów w napowietrznych liniach elektroenergetycznych z wykorzystaniem komputerowych narzędzi obliczeniowych	95
19	Jarosław KMAK, Wiesław NOWAK, Rafał TARKO - Statystyczna ocena zagrożenia porażeniowego podczas zwarcć doziemnych w stacjach elektroenergetycznych średnich napięć	99
20	Konrad KOCHANOWICZ, Wiesław NOWAK, Rafał TARKO - Ograniczanie zasięgu oddziaływania pól elektromagnetycznych wytwarzanych przez elektroenergetyczne linie przesyłowe	103
21	Tomasz BEDNARCZYK, Wojciech KOŁTUNOWICZ, Michał KRUEGER, Udo RANNINGER - Pomiar i lokalizacja wylądowań niepełnych w izolacji transformatorów energetycznych	107
22	Mateusz CYBULSKI, Piotr PRZYBYŁEK - Zastosowanie siła molekularnego 3A do poprawy stanu układu izolacyjnego transformatora energetycznego	111
23	Michał LECH, Paweł WĘGIEREK, Damian KOSTYŁA - Badania laboratoryjne wytrzymałości elektrycznej próżni uzyskanej na bazie powietrza, neonu i helu	115
24	Kamil LEWANDOWSKI, Hubert MORANDA - Zjawisko bąbelkowania w izolacji stałej transformatorów wykonanej na bazie celulozy oraz aramidu (Nomex [®] 926) zanurzonej w oleju mineralnym i estrze syntetycznym Midel 7131	119
25	Aleksandra SCHÖTT-SZYM CZAK - Zjawiska przejściowe w elektroenergetycznych kablach SN ze szczególnym uwzględnieniem sposobu pracy ich żyły powrotnej	124
26	Konrad STRZELECKI - Porównanie napięcia inicjacji i przebicia w estrach i olejach mineralnych w układzie izolowanych elektrod przy napięciu udarowym	128
27	Filip STUCHAŁA - Napięcie przyspieszenia strimerów w wybranych cieczach dielektrycznych przy napięciu udarowym piorunowym	132
28	Dominika SZCZEŚNIAK, Piotr PRZYBYŁEK, Agnieszka MARCINKOWSKA, Aneta LEWANDOWSKA - Modyfikacja estru naturalnego fulerenem C ₆₀	136
29	Jakub GAJDZICA, Wiesław NOWAK, Rafał TARKO - Analiza narażeń napięciowych powłok kabli w liniach napowietrzno-kablowych wysokich napięć	140
30	Paweł ŻYŁKA - Zmiany rozkładu efektywnej gęstości ładunku powierzchniowego w modelowej wnęce gazowej	144
31	Bartłomiej PASTERNAK, Paweł RÓZGA - Inicjacja wylądowań elektrycznych w układzie z klinem olejowym przy napięciu udarowym	148
32	Marek ANDRZEJEWSKI, Wiesław GIL, Wiktor MASŁOWSKI - Jakość pomiarów w układach monitoringu on-line izolatorów przepustowych	152
33	Ewa ZAWADZKA, Henryk BRZEZIŃSKI, Krzysztof KOGUT, Patrycja SIWORSKA – ŻUREK - Recykling wyrobów elektroenergetycznych	157
34	Grzegorz MASŁOWSKI, Robert ZIEMBA - Fale napięciowe indukowane w liniach elektroenergetycznych pobliskimi wylądowaniami atmosferycznymi	161
35	Krzysztof KOGUT, Ewa ZAWADZKA, Henryk BRZEZIŃSK - Szynoprzewód w izolacji stałej o własności kompensacji mocy biernej	165
36	Jarosław GIELNIAK, Magdalena CZERNIAK, Kamil LEWANDOWSKI - Badanie drgań transformatorów o mocy 10, 16, 25 i 40 MVA pod kątem oceny ich stanu technicznego	169
		173
		1770

PRZEGLĄD ELEKTROTECHNICZNY Vol 2022, Nr 10

Spis treści

37	Wiktoria KALUS, Jarosław ZYGARLICKI, Łukasz NAGI, 4. Michał KOZIOL - Problem generowania ozonu w urządzeniach elektroadhezyjnych	173
38	Damian KOSTYŁA, Paweł WĘGIEREK, Michał LECH - Technika światłowodowa do pomiaru ciśnienia w wysokonapięciowej aparaturze łączeniowej	177
39	Michał MOLAS, Marcin SZEWCZYK - Pomiary trajektorii iskry długiej w przestrzeni trójwymiarowej	181
40	Jacek RYBARZ, Sebastian BORUCKI, Michał KUNICKI, Andrzej CICHON - Ocena poprawności montażu osprzętu kablowego w kablach SN z izolacją XLPE pod kątem występowania wyładowań niepełnych	185
41	Daria WOTZKA, Mirosław GRYSZPIŃSKI, Ireneusz URBANIEC - Badanie promieniowania UV emitowanego przez wyładowania niepełne na powierzchni izolatora kompozytowego	189
42	Andrzej CICHON, Sebastian BORUCKI, Michał WŁODARZ - Zastosowanie wybranych klasyfikatorów uczenia maszynowego do rozpoznawania form wyładowań niepełnych	193
43	Michał KUNICKI - Analiza zmienności w czasie sygnałów generowanych przez wyładowania niepełne	197
44	Krzysztof WINCENCIK, Jarosław WIATER - Przeskoki iskrowe w przewodach o izolacji wysokonapięciowej stosowanych w ochronie odgromowej	201
45	Konstanty M. GAWRYLCZYK - Porównanie modeli obwodowych uzwojeń transformatorów energetycznych	205
46	Jacek DZIURA - Transformatory z izolacją cieczową o wyższej klasie ciepłoodporności – studium przypadku	209
47	Andrzej MROZIK - Detekcja wyładowań niepełnych w transformatorach suchych średniego napięcia	212
48	Hubert ŚMIETANKA, Krzysztof WIECZOREK, Przemysław RANACHOWSKI, Zbigniew RANACHOWSKI, Adam BRODECKI - Procesy degradacji w elementach tekstolitowych ograniczników przepięć	216
49	Stefan WOLNY - Analiza wpływu stopnia termicznej degradacji izolacji półsyntetycznej NOMEX®910 syconej estrem syntetycznym na charakterystyki dyspersyjne współczynnika strat dielektrycznych w dziedzinie niskich i wysokich częstotliwości	221
50	Marek ZENKER - Wpływ temperatury na odpowiedź dielektryczną w dziedzinie częstotliwości papieru aramidowego impregnowanego eksploatacyjnym i zregenerowanym estrem syntetycznym	225
51	Szymon BANASZAK, Michał ZEŃCZAK, Olgierd MAŁYSZKO, Jakub DOWEJKO, Jarosław JAWORSKI - Wybór miejsca zainstalowania wodorowego bufora energetycznego w systemie elektroenergetycznym	229
52	Michał KOZIOL, Jarosław ZYGARLICKI, Dariusz ZMARZŁY, Elżbieta JANOWSKA-RENKAS – Problematyka pomiarów parametrów elektrycznych betonów i geopolimerów	233
53	Aleksandra RAKOWSKA¹, Krzysztof SIODŁA - Sześćfluorek siarki a ochrona środowiska	237
54	Krzysztof A. BOGDANOWICZ - Ogniwa słoneczne na każdą pogodę – możliwości i perspektywy	241
55	Tomasz MARCINIAK, Kacper PODBUCKI, Jakub SUDER I - Zastosowanie modułu Nucleo STM32 w nauczaniu technik mikroprocesorowych w automatyce	245
56	Kinga KONDRACKA, Piotr FIREK, Magdalena JAWOROWSKA, Mariusz SOCHACKI - Technologia i charakterystyka modyfikowanych struktur ISFET na potrzeby detekcji awidyny	249
57	Maciej SIBIŃSKI, Grażyna BUDZIŃSKA - European Project Semester – Model Interdyscyplinarnej Edukacji w Elektronice	253
58	Patrycja ŚPIEWAK, Michał WASIAK, Robert P. SARZAŁA - Komputerowa analiza zjawisk pojemnościowych w azotkowych laserach VCSE	257
59	Mateusz ZAPART, Cezary WOREK - Analiza symulacyjna dyskretnego modelu układu SOGI-FLL wraz z uwzględnieniem behawioralnych modeli układów peryferyjnych mikrokontrolera	261
60	Przemysław CZARNECKI, Łukasz RUTA, Katarzyna ZNAJDEK, Ewa RAJ, Zbigniew LISIK - Prototyp hybrydowego panelu fotowoltaicznego z płytą chłodzącą	267
61	Zenon KIDON - Stabilograficzne stanowisko do badania niemowlą	271
62	Michał MOLAS, Marcin SZEWCZYK - Przegląd aktualnych kierunków badań w zakresie zastosowań modelu fraktalnego w symulacjach wyładowań elektrycznych	275
63	Paweł Pawłowski, Adam Dąbrowski, Agnieszka Stankiewicz, Karol Piniarski – Nauczanie podstaw eelektroniki w trybie zdalnym I hybrydowtm	280
64	Renata Majgier, Ewa Mandowska, Wojciech Grzesiak, Arkadiusz Mandowski - Ceramiczne podłoże grubowarstwowe jako materiał dla dozimetrii retrospektywnej	284
65	Chai Wankan, Krittanon Prathepha, Worawat Sa-ngiamvibool - W pełni zrównoważony, dostrajany prądowo filtr wieloprzepustowy z techniką CAPRIO	288
66	Hassen BELILA, Nasserline BOUDJERDA, Imen BAHRI, Ahsene BOUBAKIR - Odporna strategia sterowania hybrydowego Sterowanie trybem przesuwnym PI STATCOM w obecności zdecentralizowanego źródła PV	292
67	Jacek KOZYRA, Zbigniew ŁUKASIK, Aldona KUŚMIŃSKA-FIJAŁKOWSKA, Paweł KASZUBA - Analiza zdalnego pomiaru napięcia w sieciach kablowych SN	299
68	Andrzej Ł. Chojnacki - Wpływ temperatury otoczenia na intensywność awarii napowietrznych linii elektroenergetycznych średniego napięcia	307
69	Stanisław BEDNAREK - Sterowany elektromagnetycznie pulsator akustyczny	312

Experimental studies of design-and-technological parameters of heat exchanger

Abstract. As a result of experimental studies, a mathematical model of the influence of the length of the air duct, volumetric air flow, air temperature in the external environment on the useful thermal power of the developed heat exchanger for its given geometric parameters was obtained. As a result of experimental studies, the optimal values of the design and technological parameters of the developed heat exchanger have been established, at which its useful thermal power is maximum.

Streszczenie. W wyniku badań eksperymentalnych uzyskano model matematyczny wpływu długości kanału powietrznego, objętościowego przepływu powietrza, temperatury powietrza w środowisku zewnętrznym na użyteczną moc cieplną opracowanego wymiennika ciepła dla zadanych parametrów geometrycznych. W wyniku badań eksperymentalnych ustalono optymalne wartości parametrów konstrukcyjnych i technologicznych opracowanego wymiennika ciepła, przy których jego użyteczna moc cieplna jest maksymalna (**Badania eksperymentalne parametrów konstrukcyjno-technologicznych wymiennika ciepła**)

Keywords: three-pipe module, medium, air, power, air duct, heat, heat exchanger, temperature, pipe.

Słowa kluczowe: moduł trójrurowy, medium, powietrze, moc, kanał powietrzny, ciepło, wymiennik ciepła, temperatura, rura

Introduction

Animal husbandry efficiency directly depends on the conditions in which the animals are kept, wherein ensuring an optimal microclimate is extremely significant. For example, animals' productivity is by 10-30% determined by microclimate of the premises [1, 2]. The main indicators of microclimate include: temperature T ; relative air humidity W , %; chemical composition of air (carbon dioxide CO_2 , ammonia NH_3 , hydrogen sulfide H_2S content); presence of dust (mechanical pollution) and microorganisms (biological contamination) in the air; velocity v , m/s, and direction of air flow; lighting [3]. Air conditions are disturbed by animals' breathing (the release of heat, moisture, carbon dioxide, etc.), and as a result of evaporations from manure [4]. Among the pollution factors mostly affecting animals' breeding are gases (carbon dioxide, ammonia, hydrogen sulfide).

Deviations of microclimate parameters from physiologically determined standards weaken resistance to diseases, cause departure of up to 40 % young animals, milk yield decrease by 10-20%, decrease of up to 30% in fattening weight gain, wool shearing of up to 20%; requires additional costs of feedstock and funds for medical treatment. Microclimate deterioration also shortens the service life of livestock buildings and their technological equipment [5].

Zootechnical and sanitary-hygienic requirements to microclimate generation are reduced to ensuring that all its parameters are within the limits defined by the standards of technological designing of premises, where animals are kept [6-11]. The significance of maintaining stable levels of microclimate parameters should be emphasized. Drastic violations of conditions are especially detrimental. While deviations from optimal standards by one or another indicator are mainly accompanied by decrease in animals' productivity, drastic fluctuations in conditions (for example, temperature conditions) often cause diseases and fall in the number of animals, especially young animals [12-14].

In turn, ensuring optimal microclimate in livestock buildings is associated with significant heating and electricity costs, which make up to 15% of producers' funds. Ever-increasing power costs complicate the situation and aggravate the problem of introduction of power-saving technologies, while also actualizing the economic problem

of reducing specific power consumption for livestock production [4, 15].

To achieve animals' maximum productivity, it would be advisable to provide microclimate in livestock buildings, in power terms, with air heat exchangers, the use of which allows saving power, which is required to heat air in the premises [16].

Taking into account the technological conditions of air in livestock buildings (significant dust content – up to 6 mg/m³, high humidity – up to 80%, high concentration of aggressive components – ammonia up to 20 mg/m³, hydrogen sulfide – up to 10 mg/m³, carbon dioxide – up to 0.28%) and the results of analysis of heat recovery units' design, it was found that by sanitary-hygienic and operational indicators, high power efficiency and low installation costs, heat exchangers of "pipe in pipe" type are the most suitable for the ventilation system [16].

As of today, there exists a huge number of designs of shell-and-tube heat exchangers [16, 17] and respective studies of their design and technological parameters [18-20]. However, in these papers, little attention is paid to optimizing the operational parameters of three-pipe concentric heat recovery units.

Analysis of literary sources and problem statement

As a result of theoretical studies [21], the mathematical model of heat transfer process in three-pipe concentric heat exchangers was developed, taking into account the phenomenon of condensation therein, which makes it possible to determine the distribution of air flow temperatures along its length, and its thermal power. Optimization of theoretical studies' results made it possible to determine dependences of heat exchanger's design parameters (length L and radii r_1 , r_2 and r_3 of air ducts) on volumetric flow rate of air passing through it under condition of highest useful heat power: $L = 14.776 \cdot V + 3.7335$, $r_3 = 0.3619 \cdot V + 0.1523$, $r_1 = 0.343 \cdot r_3$, $r_2 = 0.686 \cdot r_3$ (at ambient temperature $T_c = 0^\circ C$).

Purpose and tasks of research

The purpose of the research is to verify correctness of conclusions made in the course of theoretical studies and to substantiate rational design and operating parameters of three-pipe heat exchanger.

Materials and methods

To perform experimental studies, a universal stand was made, the process flow diagram and general view of which are shown in Figure 1. The universal experimental stand for determination of rational design and operating parameters of three-pipe heat exchanger consists of a set of three-pipe modules, corner modules of the room and external environment, a fan and a heating element. Taking into account theoretical studies [21, 22], the following design parameters of three-pipe module were adopted [23]: length $L_M = 1\text{ m}$, outer tube diameter $D_M = 0.4\text{ m}$, middle pipe D^*M diameter $= 0.274\text{ m}$; inner tube diameter $D^{**}M = 0.138\text{ m}$ and pipe wall thickness $\delta M = 0.0005\text{ m}$.

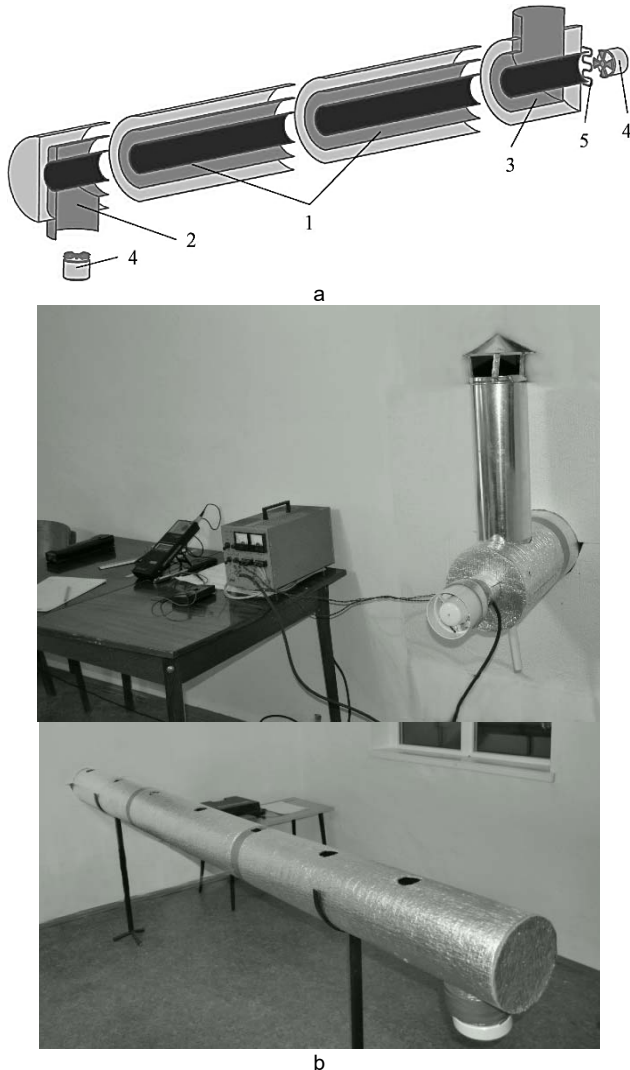


Fig. 1. Process flow diagram (a) and general view (b) of the experimental stand: 1 – three-pipe module; 2 – corner module of the room; 3 – corner module of external environment; 4 – fan; 5 – heating element

When outdoor air passes through developed heat exchanger, it interacts with pipe walls and gets heated up, while taking thermal energy from the air flow coming from the room, therefore useful thermal power was chosen as an optimization criterion, which is determined by the formula [24-26]:

$$(1) \quad \Delta N = N_Q - N_f = V \cdot \rho_{n.c.} \cdot \frac{273}{T_c} C_p (T_3(0) - T_c) - N_f$$

where N_Q is the power used to heat the cold air flow in the heat exchanger, W; N_f is the power required to pump air through the heat exchanger, W; $\rho_{n.c.}$ – air density under normal conditions ($T_{n.c.} = 273\text{ K}$, $P_{n.c.} = 101325\text{ Pa}$), $\rho_{n.c.} =$

1.293 kg/m^3 [27, 28]; V – volumetric flow rates of air in the duct (fan), m^3/s ; C_p is air's specific heat capacity, $C_p = 1006\text{ J/(kg}\cdot\text{K)}$ [24, 28]; T_c , $T_3(0)$ – air flow temperature established in the external environment and at the heat exchanger's outlet, respectively, K.

Volumetric air flow rates in the duct may be determined using the formula [24]:

$$(2) \quad V = v A_i,$$

where v – air velocity in the duct, m/s ;

A_i is the duct's cross-sectional area, m^2 .

Air velocity in the duct v was measured using "Solomat MPM 500E" multifunctional measuring device [27, 29]. Required velocity in the air duct v was set using FL FS1.6 fan performance controllers [28, 30, 31]. The power required to pump air through the heat exchanger was determined experimentally using CO-EA05 electric energy meters installed on both fans [25, 32]. Temperatures were measured using "Solomat MPM 500E" multifunctional measuring device [33, 34]. Ambient temperature was set using NK-125-0.6-1 duct electric heater [35].

The methodology of experimental research is based on mathematical planning of experiments [22, 36, 37].

Let us assume: heat exchanger length, L , m (Z_1), volumetric air flow rates (fan performance V , m^3/s) (Z_2), ambient air temperature T_c , $^{\circ}\text{C}$ (Z_3), as variation factors.

As the lower level factor Z_1 , let us assume the heat exchanger's length equal to 4 m, which corresponds, according to developed universal experimental stand, to four three-pipe modules. As the top level factor, let us assume the length equal to 8 m (eight three-pipe modules). The lower level of Z_2 factor is assumed to be 0.14 m^3/s , the upper level being 0.64 m^3/s , which corresponds to maximum air supply provided by the fan. Let us choose the lower level of factor Z_3 as equaling to 0 $^{\circ}\text{C}$, since livestock building should be at this temperature. The upper level of Z_3 factor is assumed to be 8 $^{\circ}\text{C}$, which objectively depends on climatic conditions and ambient air temperature [6, 38].

The experiments were carried out according to the plan for full three-factor experiment PFE 33, and when the factors were variable (table 1), Box-Benkin experiment planning matrix was used [22, 30, 39]. The experiments were carried out in triplicate [13, 36].

Table 1. Matrix for carrying out experimental studies of a three-pipe heat exchanger

№	Factors		
	Z1 (L, m)	Z2 (V, m ³ /s)	Z3 (T _c , °C)
1	8	0,64	4
2	8	0,14	4
3	4	0,64	4
4	4	0,14	4
5	6	0,39	4
6	8	0,39	8
7	8	0,39	0
8	4	0,39	8
9	4	0,39	0
10	6	0,39	4
11	6	0,64	8
12	6	0,64	0
13	6	0,14	8
14	6	0,14	0
15	6	0,39	4

Main Results of the Study

In accordance with PFE 33 experiment plan, 15 options of three-factor combinations in the experimental unit's design were implemented. Analysis of research results, according to accepted planning matrix, made it possible to obtain a regression model of studied factors' influence on useful thermal power of developed heat exchanger.

As a result of calculations of regression coefficients, the mathematical model was obtained in an encoded form of studied factors' influence on useful thermal power of developed heat exchanger:

$$(3) \quad \Delta N = 2849.18 - 172.459Z_1 - 190.558Z_2 - 965.489Z_2^2 - 514.543Z_1Z_2 - 2197.57Z_2^2 - 1527.01Z_3 - 118.9Z_1Z_3 - 947.671Z_2Z_3 - 182.543Z_2^2Z_3.$$

For this equation, confidence levels of variance are homogeneous to 95%, the value of Cochran's test $G = 0.2942 < G_{0.05}(2, 15) = 0.3346$. The variance in the adequacy of mathematical model $S_{ad}^2 = 573501$; the variance of experimental error $S_y^2 = 267607$; the value of Fisher's variance ratio $F = 2.14 < F_{0.05}(9, 30) = 2.21$; the model is adequate at any level of confidence.

According to calculated values of correlation coefficients and Student's test $t_{0.05}(30) = 2.04$ at a confidence level of more than 95%, the ratios for such equation terms: $Z_1, Z_1Z_2, Z_2^2, Z_3, Z_2Z_3$ are significant.

Based thereon, regression equation (3) will be as follows:

$$(4) \quad \Delta N = 2849.18 - 965.489Z_2 - 514.543Z_1Z_2 - 2197.57Z_2^2 - 1527.01Z_3 - 947.671Z_2Z_3.$$

In decoded form, model (4) will be as follows:

$$(5) \quad \Delta N = -3352.1 + 401.343L - 1029.09V - 35161.2V^2 + 33529V - 947.671VT_c - 12.1598T_c.$$

When analyzing equation (5), one can argue that useful thermal power of developed heat exchanger is affected by all above-mentioned factors (Figure 2). At the same time, with the increase of the air duct length and decrease of the air temperature in the external environment, useful thermal power also increases, and when the values of volumetric air flow vary, useful thermal power has the following optimum:

$$(6) \quad \max \{ \Delta N(L=8 \text{ m}; V=0.36 \text{ m}^3/\text{s}; T_c=0 \text{ }^\circ\text{C}) \} = 4408 \text{ W}.$$

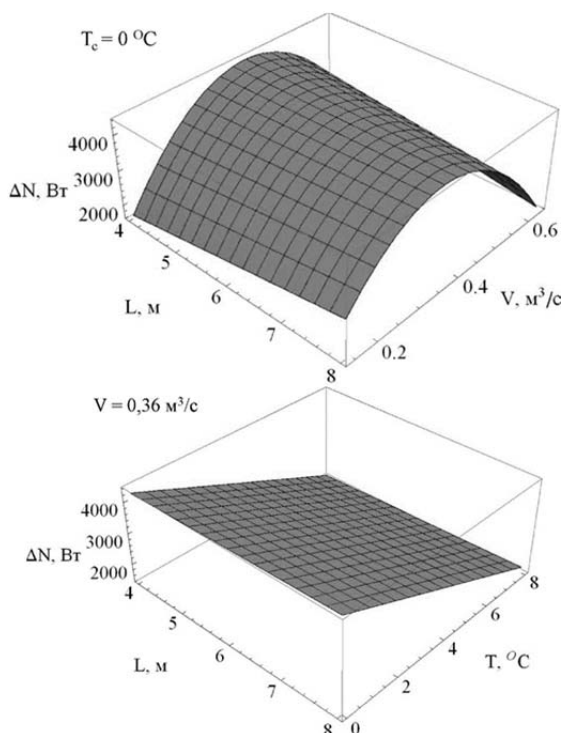


Fig. 2. Influence of duct length L , air volume flow V and air temperature in the external environment T_c on useful heat output of developed heat exchanger ΔN .

Conclusions

1. As a result of experimental studies, a mathematical model of the influence of the duct length L , volumetric air flow V , air temperature in the external environment T_c on the useful thermal power of the developed heat exchanger ΔN for its given geometric parameters was obtained.

2. As a result of experimental studies, it has been established that for given radii of the heat exchanger air duct $r_3 = 0.2 \text{ m}$, $r_2 = 0.137 \text{ m}$, $r_1 = 0.069 \text{ m}$, the optimal values of its structural and technological parameters at which the maximum useful thermal power of the heat exchanger $\Delta N = 4408 \text{ W}$ are $L = 8 \text{ m}$, $V = 0.36 \text{ m}^3/\text{s}$, $T_c = 0 \text{ }^\circ\text{C}$.

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Authors: YAROPUD Vitaly – PhD in Engineering, Associate Professor, Dean of the Faculty of Engineering and Technology, Vinnytsia National Agrarian University (21008, 3 Sonyachna str., Vinnytsia, Ukraine), e-mail: yaropud77@gmail.com; KUPCHUK Ihor – PhD in Engineering, Associate Professor, Deputy Dean for Science, Faculty of Engineering and Technology, Vinnytsia National Agrarian University (21008, 3 Sonyachna str., Vinnytsia, Ukraine), e-mail: kupchuk.igor@i.ua; BURLAKA Serhiy – PhD in Engineering, Senior Lecturer, Faculty of Engineering and Technology, Vinnytsia National Agrarian University (21008, 3 Sonyachna str., Vinnytsia, Ukraine), e-mail: ipserhiy@gmail.com; POBEREZHETS Julia – PhD in Agricultural Science, Associate Professor, Faculty of Production Technology and Processing of Livestock Products and Veterinary, Vinnytsia National Agrarian University (21008, 3 Sonyachna str., Vinnytsia, Ukraine), e-mail: julia.p08@ukr.net; BABYN Ihor – PhD in Engineering, Associate Professor, Faculty of Engineering and Technology, Vinnytsia National Agrarian University (21008, 3 Sonyachna str., Vinnytsia, Ukraine), e-mail: ihorbabyn@gmail.com.

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