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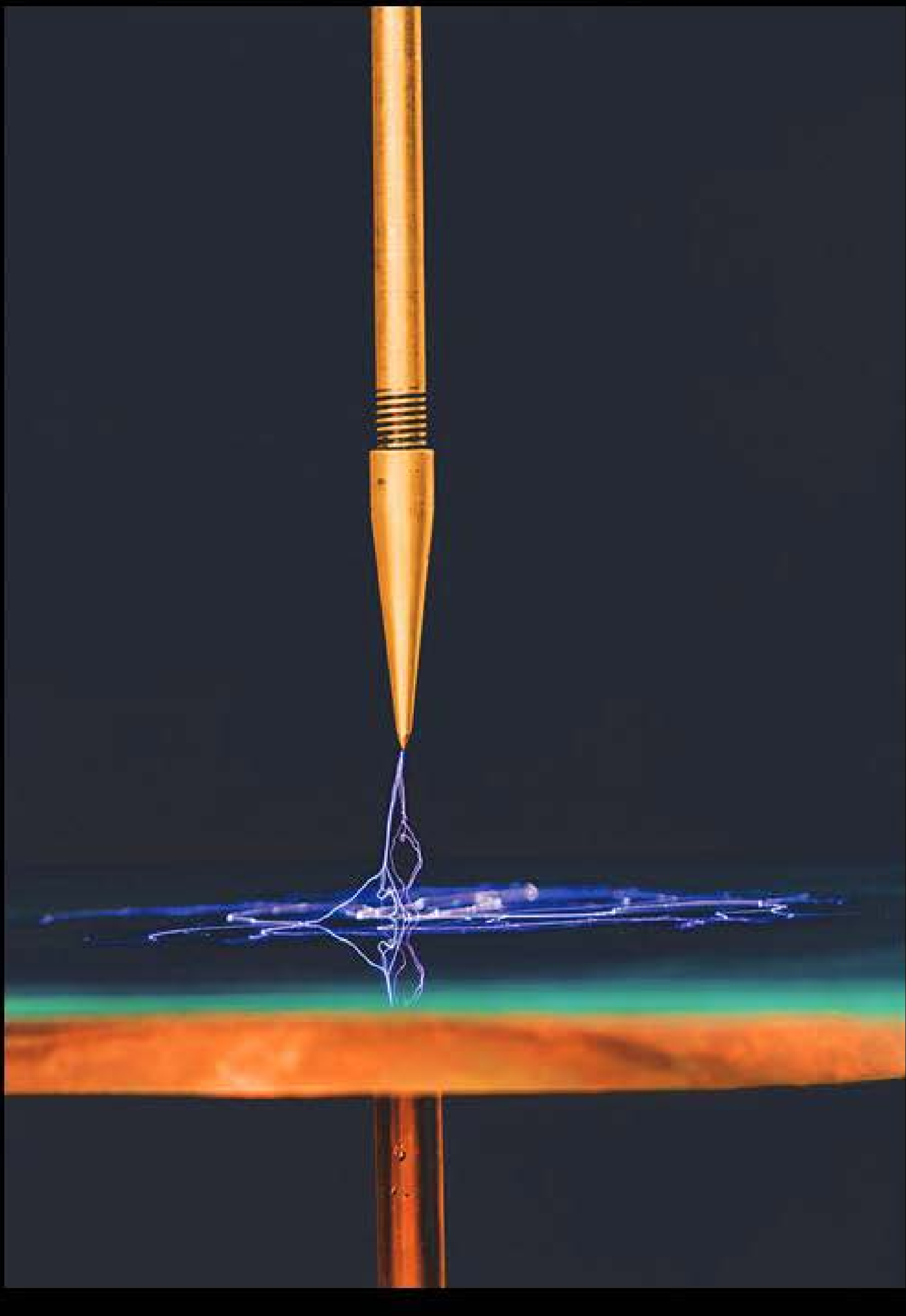

W numerze wybrane artykuły po XV Ogólnopolskim Sympozjum Inżynieria Wysokich Napięć IW-2022

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Spis treści

Vitaly Yaropud, Ihor Kupchuk, Serhiy Burlaka, Julia Poberezhets, Ihor Babyn

Vinnytsia National Agrarian University, (1)
ORCID. 1. 0000-0003-0502-1356, 2. 0000-0002-2973-6914, 3. 0000-0002-4079-4867, 4. 0000-0002-1727-6105,5. 0000-0002-7070-4957

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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 CO2, ammonia NH3, hydrogen sulfide H2S 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

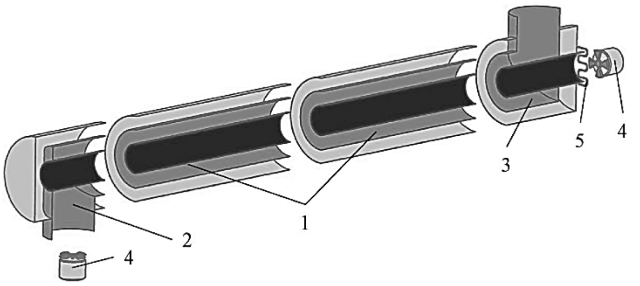
As a result of theoretical studies [21], the mathematical model of heat transfer process in three-pipe concentric heat exchangers was developed, taking into account the phenomenon of condensation therein, which makes it possible to determine the distribution of air flow temperatures along its length, 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 r1, r2 and r3 of air ducts) on volumetric flow rate of air passing through it under condition of highest useful heat power: $L = 14.776 \cdot V + 3.7335$, $r3 = 0.3619 \cdot V + 0.1523$, $r1 = 0.343 \cdot r3$, $r2 = 0.686 \cdot r3$ (at ambient temperature Tc = 0°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 LM =1m, outer tube diameter DM=0.4 m, middle pipe D′M diameter =0.274 m; inner tube diameter D″M =0.138 m and pipe wall thickness δM =0.0005 m.



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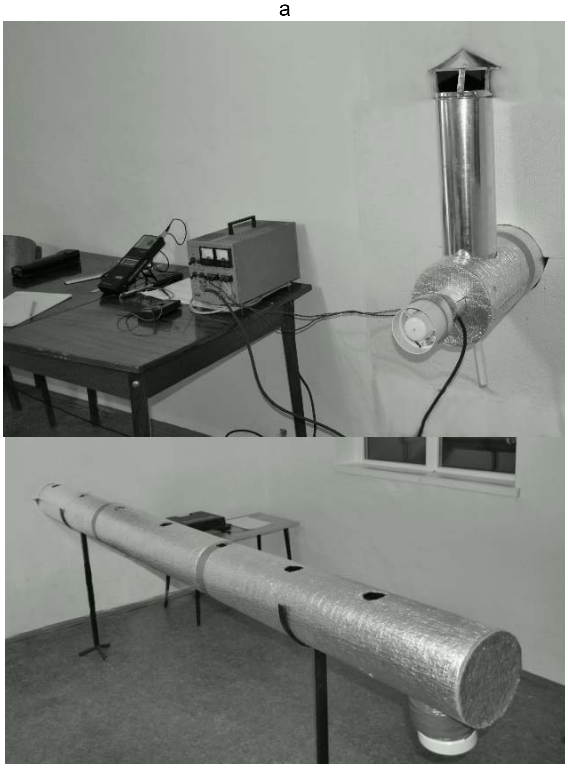
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Fig. 1. Process flow diagram (a) and general view (b) of the experimental stand: 1 – three-pipe module; 2 – corner module of the room;3 – corner module of external environment; 4 – fan; 5 – heating element

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]:

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

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 K, $P_{n.c.}$ = 101325 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 J/(kg·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]:

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

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, °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 ^{0}C, since livestock building should be at this temperature. The upper level of Z_3 factor is assumed to be 8°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:

$$\Delta N = 2849.18 - 172.459Z1 - 190.558Z21 - 965.489Z2 - 514.543Z1Z2 - 2197.57Z22 - 1527.01Z3 - 118.9Z1Z3 - 947.671Z2Z3 - 182.543Z23. \quad (3)$$

For this equation, confidence levels of variance are homogeneous to 95%, the value of Cochran's test G = 0.2942 < G0.05(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 < F0.05(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:

$$\Delta N = 2849.18 - 965.489Z2 - 514.543Z1Z2 - 2197.57Z22 - 1527.01Z3 - 947.671Z2Z3. \quad (4)$$

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

$$\Delta N = -3352.1 + 401.343L - 1029.09\,VL - 35161.2\,V2 + 33529\,V - 947.671\,VTc - 12.1598\,Tc. \quad (5)$$

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:

$$\max\{\Delta N(L = 8\text{ m}; V = 0.36\text{ m3/s}; Tc = 0\text{ °C})\} = 4408\text{ W}. \quad (6)$$

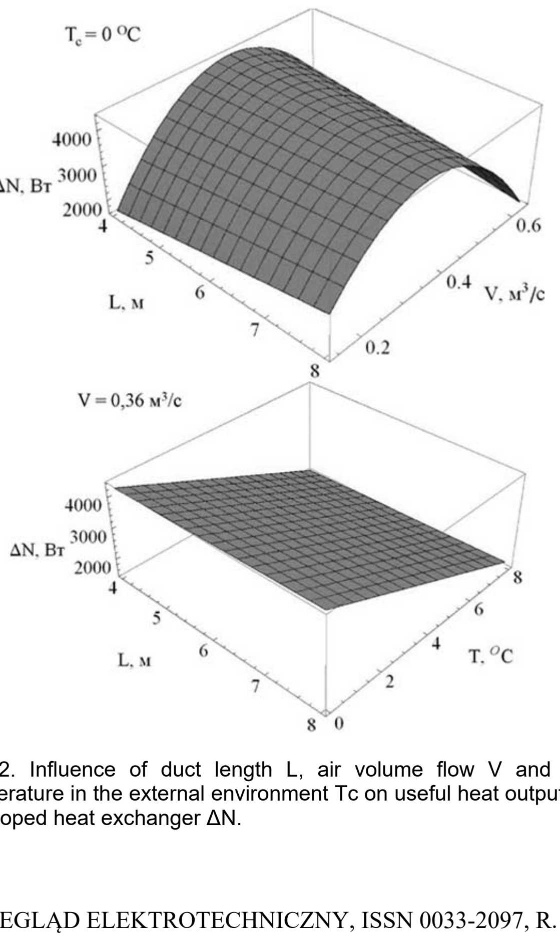

Fig. 2. Influence of duct length L, air volume flow V and air temperature in the external environment Tc on useful heat output of developed heat exchanger ΔN.

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 Tc 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 r3 = 0.2 m, r2 = 0.137 m, r1 = 0.069 m, the optimal values of its structural and technological parameters at which the maximum useful thermal power of the heat exchanger ΔN = 4408 W are L = 8 m, V = 0.36 m3/s, Tc = 0 ^{0}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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