The Experimental Determination of the Heat Transfer Coefficient in Thin Channels of a Regenerative Heat Exchanger

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The use of exhaust air heat recovery systems is crucial for modern energy-efficient ventilation. In private homes, space limitations often hinder centralized systems, making decentralized regenerative venti–lation an effective alternative. These systems feature small equivalent diameter cells, requiring refined heat transfer coefficients for accurate efficiency assessments.
An experimental setup was constructed to study heat transfer in tubes with diameters of 3 mm and 5 mm, leading to the development of a Nusselt number formula to determine the convective heat transfer coefficient precisely. To minimize environmental interaction, suitable thermal insulation thickness was applied.
Modeling channels with 3 mm and 9 mm diameters revealed temperature distribution and relationships between the Nusselt and Grashof numbers. A comparative analysis of the Nusselt number formula was conducted, aiding in evaluating heat transfer coefficients for tubes with internal diameters from 3 to 8 mm, along with the construction of a diagram for engineering applications.

Keywords:*heat transfer coefficient, regenerative heat exchanger, energy efficiency, decentralized ventilation, custom library for Scilab.*

1. INTRODUCTION

Energy-efficient technological solutions for forming the microclimate in indoor spaces play a crucial role in modern engineering, science, and technology [1-4]. Their implementation is a requirement of international and local legislation and pertains to various types of buildings with different purposes [5]. Standards regu–lating energy-efficient air exchange in low-rise residential buildings mandate exhaust air heat recovery systems. There are several types of heat exchangers with varying designs, constructions, and operational efficiencies - such as rotary regenerators, plate recupe–rators, and those with intermediate heat transfer fluids [6-9].

Heat exchangers are vital components in modern energy systems, as they ensure the effective use of thermal energy released during processes like cooling and heating. The diversity of heat exchanger designs, particularly the use of spiral and horizontal spiral-coil configurations, significantly impacts their performance and efficiency [10-13]. Research indicates that opti–mizing parameters such as fill mass and tube shape can greatly enhance heat transfer in various environments, including greenhouses and poultry houses [14-15]. Moreover, the introduction of new technologies, such as segmented twisted tapes, can improve heat exchange processes in tubes [16-17]. Special attention should be given to the influence of different tube types on the efficiency of heat exchangers [18-22].

Typically, the installation of ventilation systems in apartments or residential buildings requires dedicated space for the placement of ventilation units and duct–work, as well as provisions for maintenance access to control valves and filters and access to control panels. This increases capital costs during construction or renovation and can disrupt interior design.

Decentralized ventilation systems exist to avoid these issues. An example of such a system with minimal impact on the building's interior is the Twin Fresh regenerative unit by Vents (fig. 1) [23]. Its design includes external and internal face panels or grilles (depending on the modification), a duct axial fan, a filter, and, most importantly, a ceramic heat recovery element. It alternately passes two flowsofwarm and cold air. Due to its porous structure and material properties, heat is absorbed within its mass, allowing for heat/cold accumulation and release.

This type of ventilation equipment should be ins–talled to ensure that air movement occurs in all zones without stagnation. The proper organization of air circulation in spaces for various purposes is a concern for many researchers [24-27]. They have developed a series of recommendations for effective air exchange in rooms [28-31]. Reversible decentralized ventilators must be paired to prevent the intake of untreated outside air and the release of indoor air without heat or cold recovery.

This study focuses on the heat exchange processes occurring within the regenerators of the Twin Fresh by Vents and Blauberg Vento systems.

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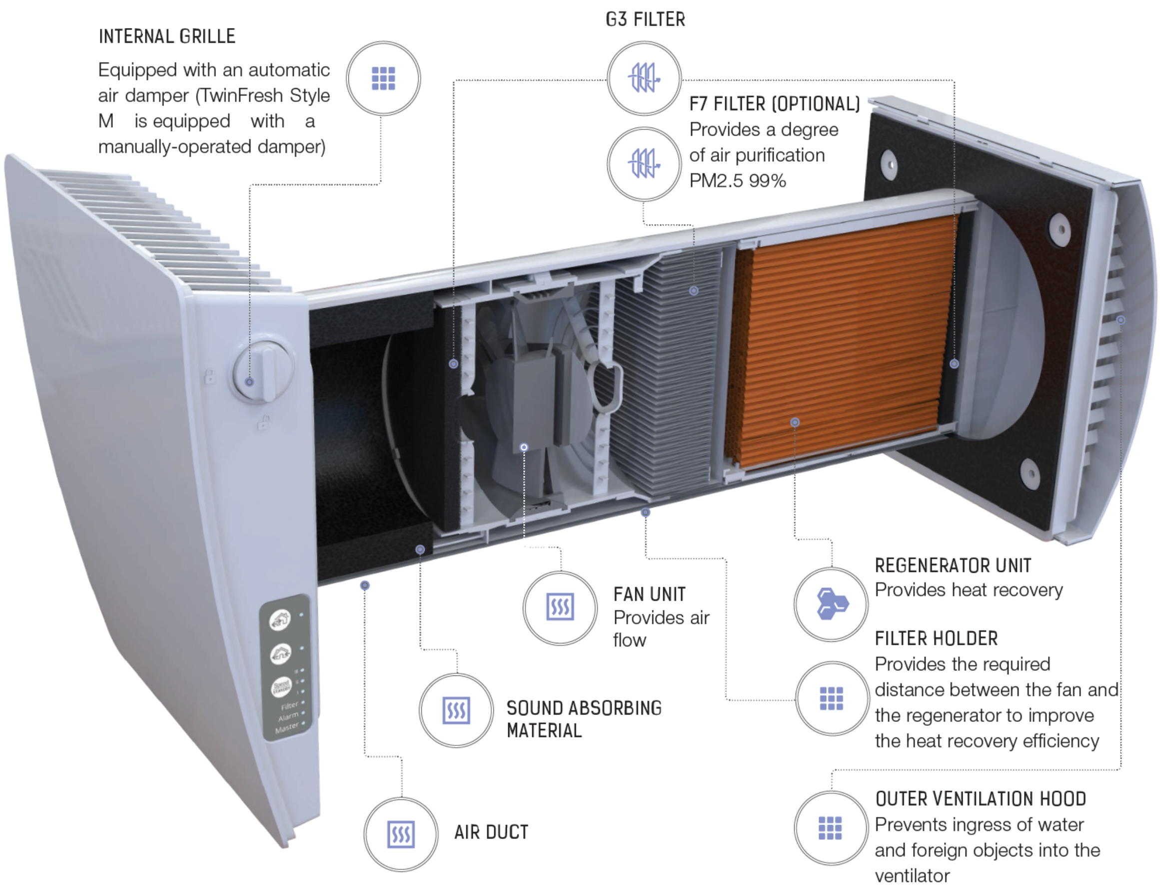


Figure 1. Ventilation unit Twin Fresh by Vents [10]

A distinctive feature of such processes is using thin channels [32] for alternate passing of two laminar airflows of cold and warm air. A significant number of scientific research has been devoted to studying heat and mass transfer processes between laminar flows and the surfaces of tubes with these flows[33-35]. The fluids in such studies include various oils, glycerin, water, kerosene, acetone, benzene, alcohol, and other liquids. Notably, these substances have Prandtl numbers signi–ficantly different fromair, which critically affects the research outcomes.

Before the advent of heat recovery ventilators in ventilation systems, problems associated with airflow through thin channels were not researched. Authors of several well-known studies [34-35] propose formulas for determining the average Nusselt number, taking into account the Reynolds number. However, as established by the authors of this paper [36–37], these formulas do not correspond to the actual operating conditions of the Twin Fresh heat recovery unit.

In the course of previous research and analysis of the operation of these heat regenerators, mathematical modeling of thermal and mass exchange was performed.

The experimental formula by M. O. Mikheev for laminar flow, which includes the Reynolds number, was applied [38-39]:

$$Nu = 0.33 \cdot \mathrm{Re}^{0.5} \cdot \mathrm{Pr}_{air}{}^{0.33} \tag{1}$$

where Re is the Reynolds number; Pr_{air} is the Prandtl number of air.

For a constant surface heat flux, the Nusselt number is given by [40-43]:

$$Nu = 4.36 \tag{2}$$

In the literature [44], a more complex formula of the form (1) is presented, which takes into account the influence of gravitational forces. After transformations, it takes the form:

$$Nu = 1.439 \cdot d^{0.3} \cdot \mathrm{Re}^{0.33} \cdot \mathrm{Pr}_{air}{}^{0.43} \tag{3}$$

where d isthe diameter of the tube[m].

It has been established that there is a significant dis–crepancy in the values of the mean temperature effectiveness coefficient of its operation (ranging from 40% to 95%), which was determined using heat transfer coefficients from various approaches [36-37]. The lite–rature did not provide conditions under which the experimental value of the Nusselt number was deter–mined by the authors of the calculation formulas. Thus, the possibility of analyzing the compliance of the discu–ssed approaches with the operating conditions of the regenerative heat exchanger was absent. To clarify the results, there arose a necessity for experimental investi–gations.

The aim of this researchis the experimental determi–nation of local heat transfer coefficients from the channel surface to the air.

2. CONSTRUCTION EXPERIMENTAL SETUP

Due to the small diameter of the channels in the heat regenerators, the impact of heat loss is significant. Therefore, it is essential to use the correct diameter of thermal insulation. It is consideredto increase the diameter of the thermal insulation significantly. Nevertheless, the critical radiustheory said it would decrease the effectiveness of insulation. An analysis of existing theories regarding the critical horizontal and vertical thermal insulation radius has been conducted [45-46]. As the radius of the thermal insulation increases, the heat transfer rises asymptotically. It is advisable to use not a critical but a practical radius of thermal insulation for each specific task.

Moreover, there is the challenge of measuring the temperatures of the air and the walls of the tube from the inside, as in this case, the sensors would significantly affect the flow.

The feasibility of measuring the temperature on the external surface of a tube with a cross-section of 10x1 mm was analyzed and verified [47] when the air temperature varied from 293.15 to 318.15 K, and the wall surface temperatures ranged from 322.15 to 334.15 K with different diameters of thermal insulation 0.02, 0.05, and 0.1 m.

The experimental plan involved using tubes with diameters corresponding to the cross-sections of existing modifications of the Twin Fresh by Vents regenerators. Specifically, the channels have diameters of 0.003 m and 0.008 m, which necessitated the use of thin thermocouples made from copper and chrome wire with a thickness of 0.2 mm. The authors constructed the thermocouples and calibrated them in a specialized calibration setup (fig. 2, fig.3).

Figure 2. Experimental unit for thermocouple calibration with expedient insulation radius [46]

The vertical vacuum flask (3) was additionally insulated using an appropriate thickness of vertical thermal insulation (4) to achieve a water cooling of less than 0.2 K/h. A thermocouple (1) and reference mercury thermometer (2) were put inside a copper tube submerged in a hot liquid of different temperatures (5). The cold junction of the thermocouples (7) was immersed in a container with a mixture of water and ice (6). After that, they were connected to a voltmeter (8).

The experimental setup (fig. 3, fig. 4) consists of a copper tube (1) resting on a wooden base and featuring a stabilization section (4). This tube is covered with K-Flex thermal insulation with a thickness of 0.05 mm and foil insulation with a thickness of 0.01 mm. Thermocouples (3) are attached to the surface of the tube, along with measuring (7) and power (8) cables.

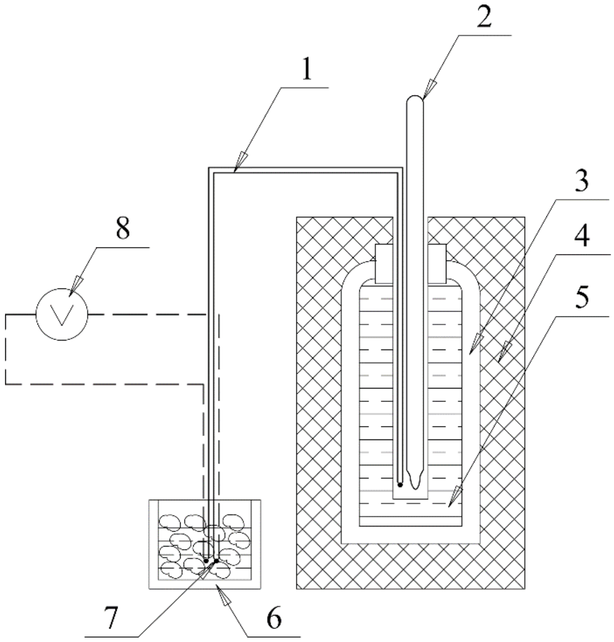


Figure 3. Schematic Diagram of Experimental unit for thermocouple calibration with expedient insulation radius

The power cables connect to a current transformer (10), which is linked to an ampermeter (9) with a toroidal transformer (11). The latter is connected in series to a laboratory autotransformer (12), a voltage stabilizer "Ukraine 3" (13), and a power outlet. The laboratory autotransformer (12) regulates the voltage and current, thus controlling the heating power of the tube.

Using a laboratory multimeter (19), readings from the thermocouples and the voltage across the tube (1) are taken. To ensure the accuracy of the thermocouple data, the junctions (20) of the constantan and chrome wires with copper leads are placed in a thermal mug (22) containing a mixture of water and ice at a temperature of 273.15 K.

The air temperature at the inlet of the tube (1) is measured by a thermometer (5), while at the outlet, it is measured using a thermocouple (2). The air supplied to the tube (1) via an axial fan (6) is filtered through a filter (13). The airflow rate is regulated using a valve (16) and recorded by a membrane flow meter (15) and a video camera (18).

A rotation speed control is incompatible with a multi-cycle gas meter. During each cycle, the resistance of the meter is low. When cycles are switched, an additional energy is necessary to switch the spool valve mechanism. The meter will stop the flow motivated by a reduced-speed fan. If a flow throttle is used, its authority is high during a cycle. When the cycle is switched, the authority will drop to zero, and almost fan energy will be applied to the spool valve mechanism, which will switch rapidly without significant influence on flow. The air duct system has enough large diameter to damper the pulsations. The flow meter is connected to the system with a flexible hose (17).

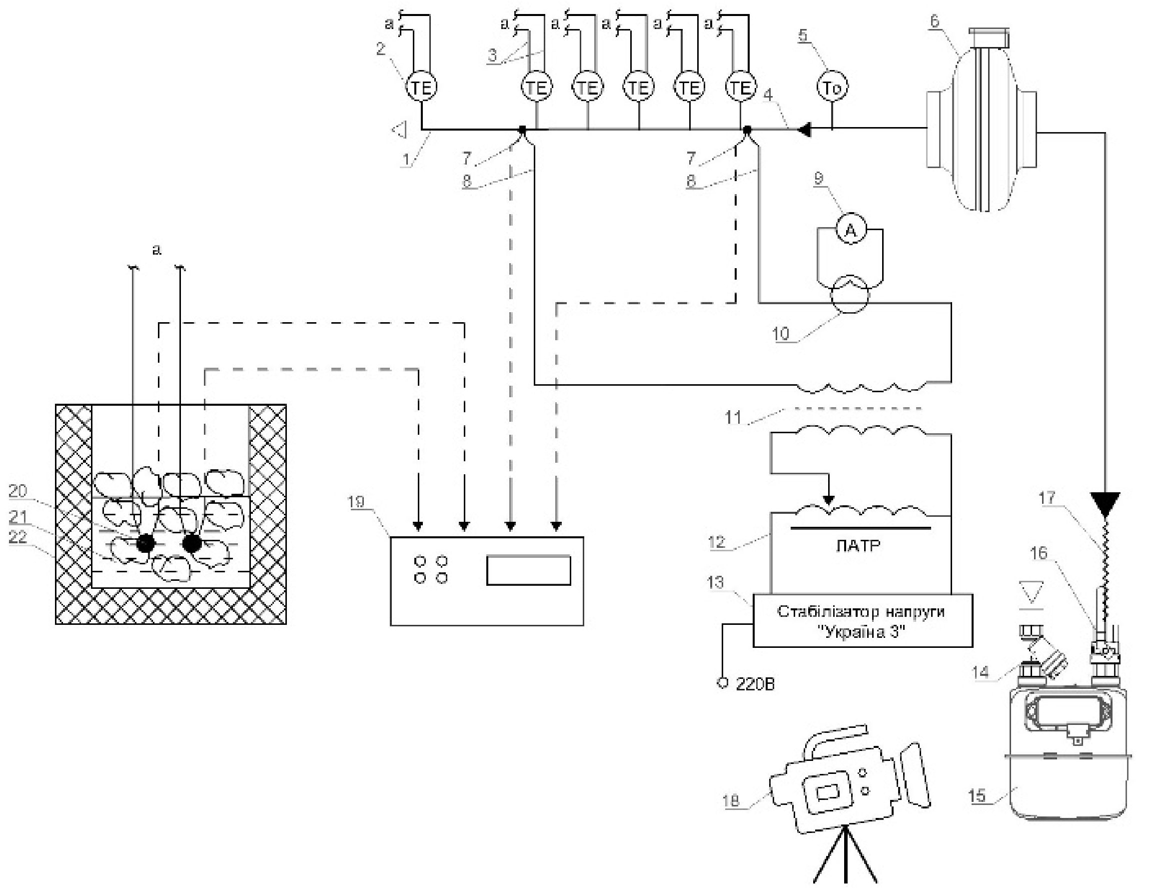


Figure 3. Schematic Diagram of the Experimental Setup for Determining Local Heat Transfer Coefficients from the Channel Surface to Air

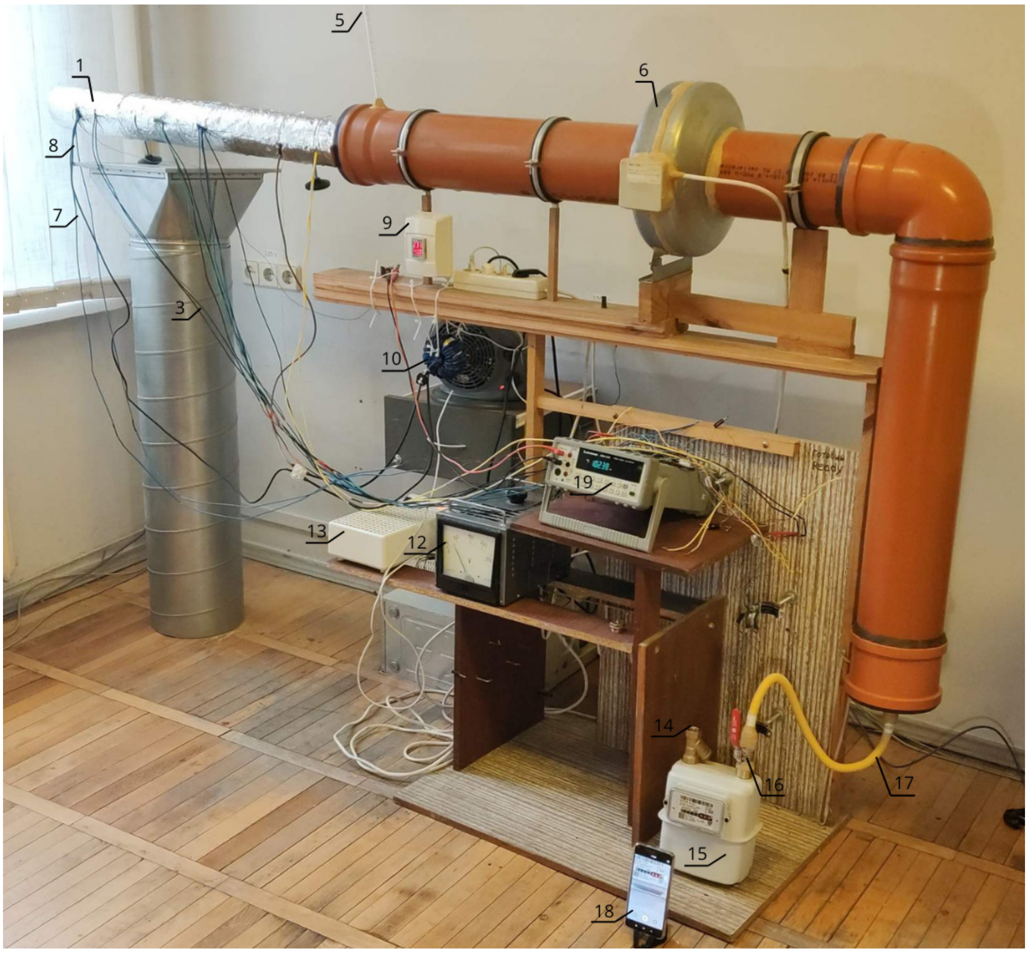


Figure 4. Experimental setup for determining local heat transfer coefficients from the channel surface to air

The heating of the tube and the measurement of its surface temperature occur in a section with a stable temperature profile after a stabilization section of length: $l = 5 \cdot d$ [m].

During the experimental research, significant heat losses through the cables were observed. A composite wire concept was adopted to address this issue. A thin wire is connected to the tube, and after $10 \cdot d$ [m], a thicker wire is attached to it. Since it is practically impossible to calculate these wires, a multi-stranded wire was used, with some strands trimmed to achieve equal temperatures between the tube and the strands. Similar approaches have also been utilized for studying heat exchange in gre–en constructions [48]. This approach is used in the project of a national standard for laboratory research of positive effects in green structures, which is prepared and will be published for public discussion as soon as possible.

As a result of a series of laboratory experiments, dependencies of the temperature changes on the wall surfaces T_w [K] of tubes with diameters of 0.003 m (fig. 5. a and b) and 0.008 m (fig. 6. a and b) were obtained and approximated. In each experiment, parameters such as the heating power of the tube and the velocity of air movement through it were varied.

The task arises to determine the air temperature. Direct measurements are not feasible, so the decision was made to use balance equations.

Let us consider a cross-section of the experimental tube (fig. 7). We will direct an X-axis along the airflow direction. The temperatures of the air and the tube walls increase along the X-axis.

In this case, heat flows from the hotter body (the wall surface) to the cooler environment (the air). These processes can be described by the equation:

$$C_p \cdot G \cdot dT_{air} = \left(q_l - C\right) \cdot dx \tag{4}$$

where C_p is the specific heat of air, taken as $C_p = 1006$ J/(kg·K); G is the mass flow rate of air [kg/s]; $T_{air}$is the air temperature [K]; $q_l$is the specific heat gain, given as $q_l = U \cdot I / l$ [W/m];$U$is the voltage applied to the tube [V]; I is the current [A]; l is the length of the tube [m]; C is the linear specific heat losses [W/m]; x is the distance from the origin along the same axis [m] by the following equation:

$$C = 2 \cdot \bar{C}\left(1 - \frac{T_{\max} - T_{out}}{T_{\max} + T_{\min} - 2 \cdot T_{out}}\right) + \\ + \frac{2 \cdot \bar{C} \cdot x}{l} \cdot \left(2 \cdot \frac{T_{\max} - T_{out}}{T_{\max} + T_{\min} - 2 \cdot T_{out}} - 1\right) \tag{5}$$

T_{max} and T_{min} are the maximum and minimum tempe–ratures of the tube wall surface [K]; T_{out} is the ambient temperature [K]; $\bar{C}$ is the average heat loss along the length, defined as: $\bar{C} = \sum C / l$ [W/m].

By performing simple transformations of (5), we obtain the dependence of air temperature on the coor–dinate x:

$$T_{air}\left(x\right) = T_{air_{in}} + \frac{\left(q_l - C_0 - k \cdot x / 2\right) \cdot x}{C_p \cdot G}\left[K\right] \tag{6}$$

where $T_{air_{in}}$ is the air temperature at the entrance to the experimental section, measured by thermometer (2) in Fig. 3.

The local heat transfer coefficient from the tube to the air is given by:

$$\alpha\left(x\right) = \frac{q_l - C}{\pi \cdot d \cdot \left(T_w - T_{air}\right)}\left[W / \left(m^2 \cdot K\right)\right] \tag{7}$$

The following formula gives the local value of the Nusselt number:

$$Nu = \frac{\alpha\left(x\right) \cdot d}{\lambda_{air}} \tag{8}$$

where λ_{air} is the thermal conductivity of air [W/(m·K)], determined using the formula (3.34) [49].

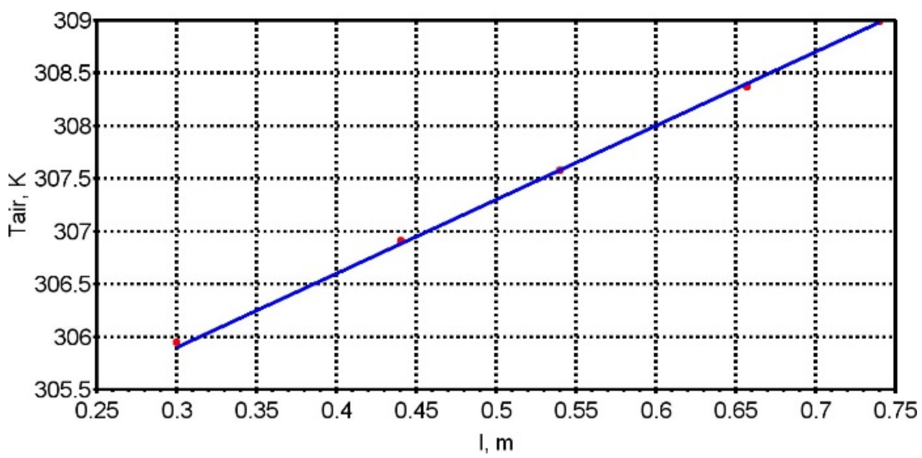
In the work [50], the commonly accepted correlation for the Nusselt number is expressed as:

$$Nu = A \cdot \mathrm{Re}^n \cdot \mathrm{Pr}^m \cdot Gr^k \tag{9}$$

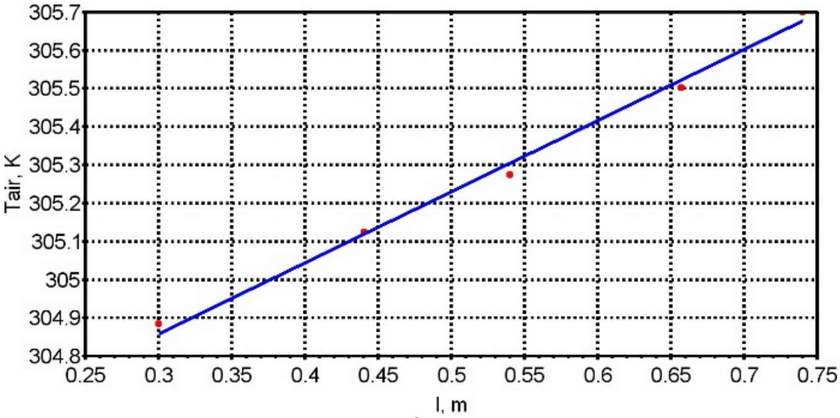
where A, n, m, karethe coefficients depending on the flow regime; Pr is Prandtl number; Gr is Grashof num–ber:

$$Gr = \frac{g \cdot d^3 / \nu^2}{T_{air} \cdot (T_w - T_{air})} \tag{10}$$

where g = 9.80665 – acceleration of gravity, m/s^2; ν – coefficient of kinematic viscosity of air, m^2/s.

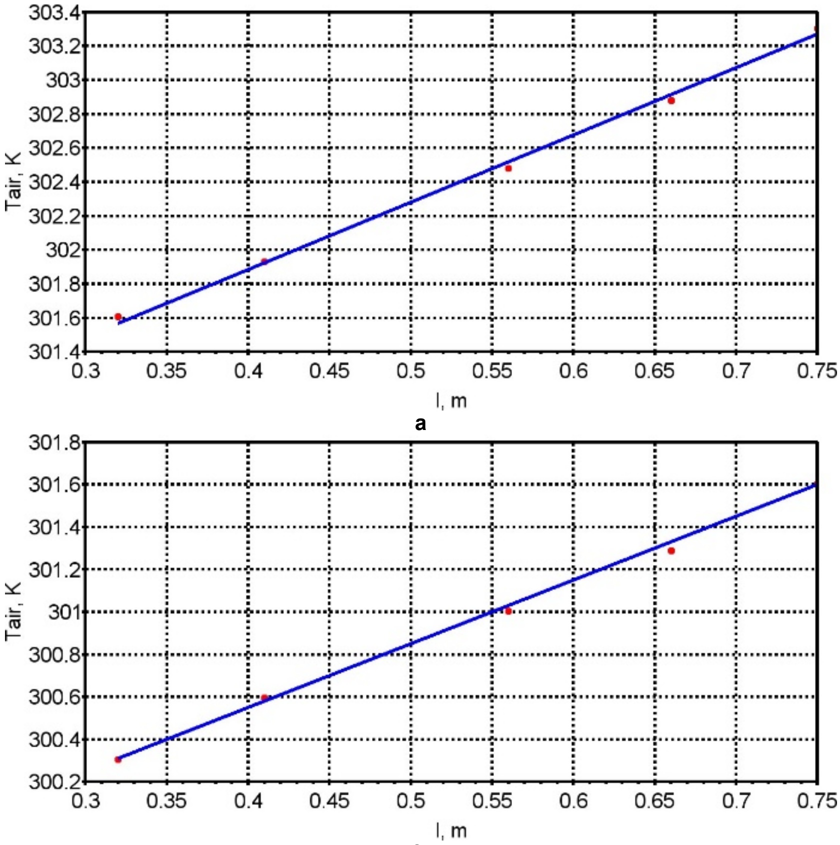


a



b

Figure 5. The dependencies of the wall temperature of the tube with a diameter of 0.003 m relative to its length: red dots represent the actual measured values, while the blue line indicates the approximation



b

Figure 6. The dependencies of the wall temperature of the tube with a diameter of 0.009 m relative to its length: red dots represent the actual measured values, while the blue line indicates the approximation

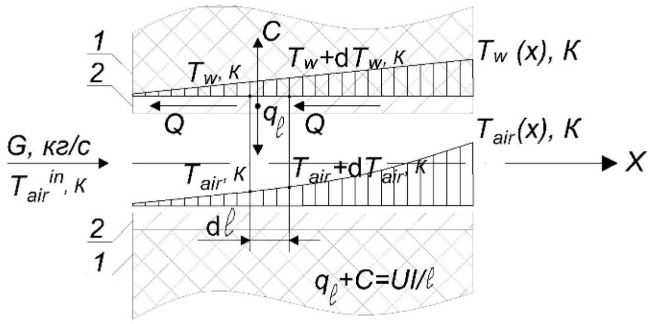


Figure 7. Experimental tube with insulation:1 – thermal insulation K-Flex and foam;2 – copper tube

In the event that the results of (1) cannot be ade–quately described, additional dimensionless factors must be introduced, with the only possible factor being the ratio *x/d*.

We analyzed the obtained local values of the Nus–selt number, considering its dependence on the Rey–nolds and Grashof numbers [36-37, 45-46], while neg–lecting the Prandtl number, which has a deviation of 0.66% from the average value:

$$Nu = A \cdot \mathrm{Re}^n \cdot Gr^m \tag{11}$$

3. RESULTS AND DISCUSSION

During the research, the wall temperature was found to be linearly dependent on length, thus confirming the assumption illustrated in Fig. 7. When considering each infinitesimal partof the tube, we establish a heat balance

- whatever heat Q[W] enters, the same amount exits. Side effects were not taken into account due to effective insulation at the ends.

Due to the highly complex processes involved, selecting operating modes for the installation in advance was impossible using a pre-developed experimental design matrix. Therefore, many experiments were conducted, and the results were processed directly using the least squares method.

The Smart_Refund function from the authors' Decimal.sci library for Scilab was utilized for data processing with rounding in the decimal system. This function automatically removes insignificant factors and optimizes regression coefficients according to any spe–cified rule, in this case, using the least squares method:

$$\sum\left(Nu-\widehat{Nu}\right)\to \min \tag{12}$$

where Nu is the experimental value of the Nusselt number, and $\widehat{Nu}$ is a regression value.

After that, the library rounds the regression coeff–icients to ensure acceptable approximations of the results (in this case, preserving four significant figures of the sum of squared deviations).

As a result of the regression, we obtained:

$$\widehat{Nu}=\frac{500}{\left(\frac{Gr}{100}\right)^{1.92}\cdot\frac{x}{d_2}} \tag{13}$$

Here, 100 is the scaling factor that ensures a good conditionedmodel. Without it, the least squares method (12) yields unstable results.

The deviation of the experimental results of the Nusselt number according to equation (13) ranges from 0.17% to 18.7%. A nomogram has been created based on (13) (fig. 8).

The Reynolds number turned out to be a negligible factor, consistent with the theoretical convection-diffu–sion equation (2). However, unlike this formula, the Grashof number and the dimensionless ratio x/d have a significant impact. The negative influence of the Gras–hof number is explained by the relatively small sizes of the tubes. The vertical convective flows that form within them enhance heat transfer only within their own boundaries. Significant aerodynamic wakes develop be–hind them, where heat exchange is considerably impeded. In tubes of this diameter, this inhibition pro–cess predominates. In 3 mm diameter tubes, convective flows cannot form at all, resulting in a practically negligible influence of the Grashof number (fig. 9)

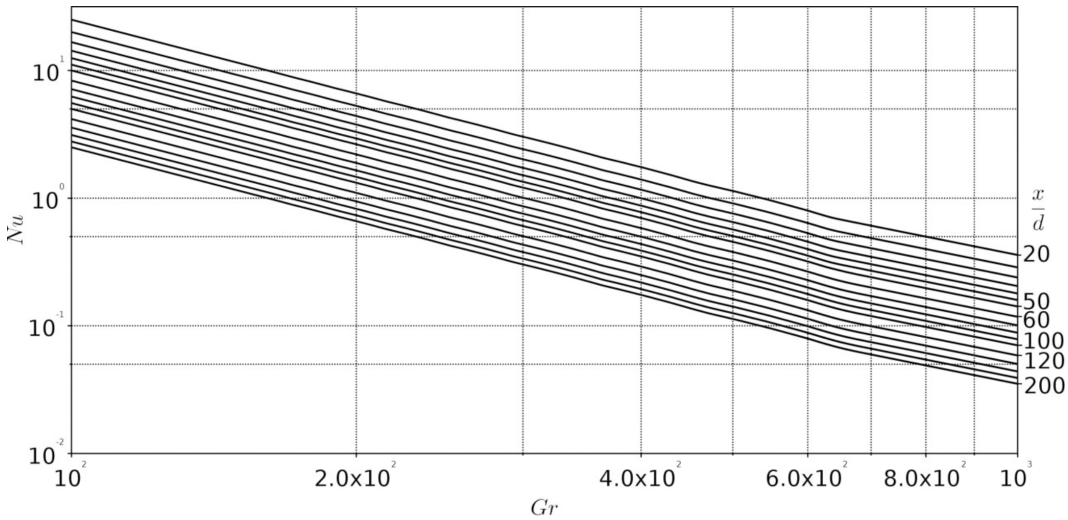


Figure 8. A nomogram of the Nusselt number dependence on the Grashof number for values of *x/d* ranging from 20 to 200

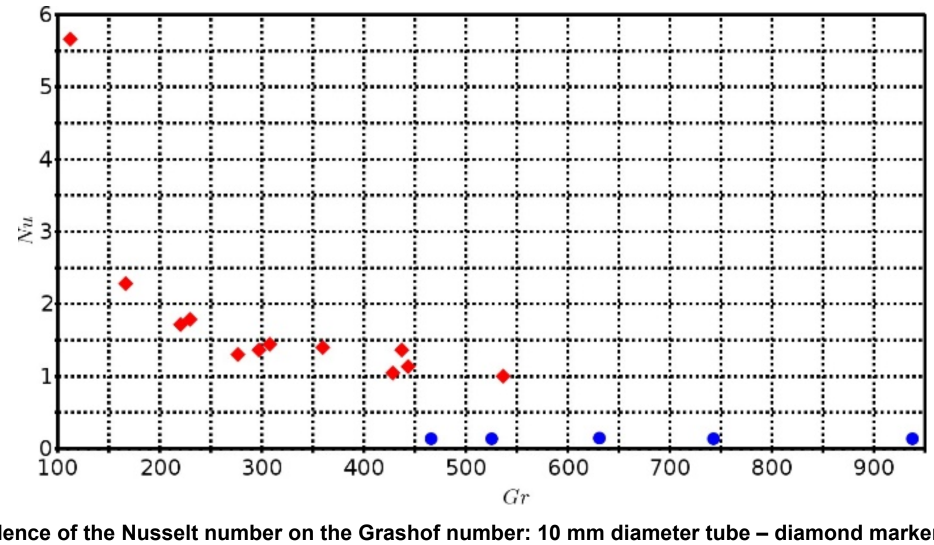


Figure 9. Dependence of the Nusselt number on the Grashof number: 10 mm diameter tube – diamond markers, 3 mm diameter tube – circular markers

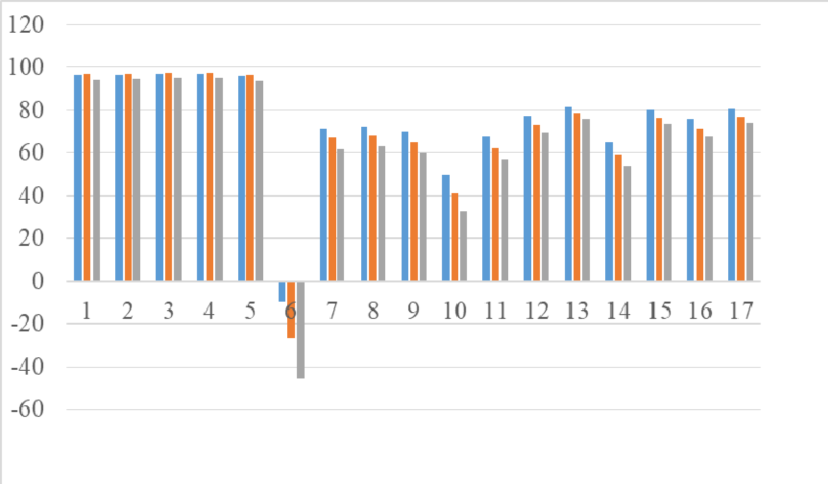


Figure 10. Deviation of formula (11) from formulas: (1) – blue, (2) – orange, (3) – gray

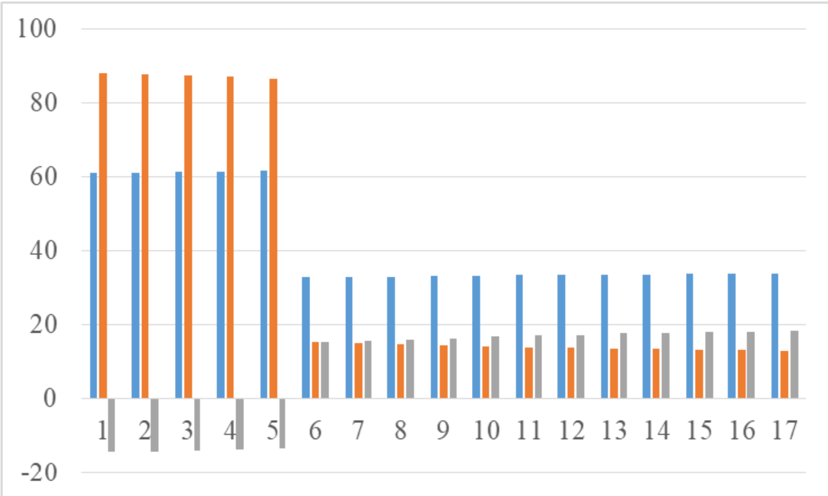


Figure 11. Deviation of formulas (1), (2), and (3) in relation to each other: (1) to (3) – blue, (2) to (3) – orange, (1) to (2) – gray

The obtained result contradicts the form of (1). Compared to the formula that accounts for the Grashof number (3), the Reynolds number does not exert a sig–nificant influence.

The deviations of the results from (11) compared to (1), (2), and (3) are shown in the diagram (fig. 10). Thedeviations among (1), (2), and (3) for comparison are presented in (fig. 11).

As seen in the diagram, the divergence is significant; however, judging by the values, it corresponds to the potential extrapolation error from the ranges provided by the authors of (1), (2), and (3) to the range of this ex–periment.

Thus, for the given range of tube diameters, it is ne–cessary to use (6). This equation is incorporated into the mathematical model of the heat recovery unit Twin Fresh or Blauberg Vento. Based on these equations, the heat exchanger's operation is optimized, and its signi–ficant thermal characteristics are determined.

4. CONCLUSION

In the framework of the experimental study, heat exc–hange processes in tubes with diameters of 3 and 5 mm were thoroughly investigated. As a result, an experimental formula for the Nusselt number was obtained, providing a reliable method for accurately determining the convective heat transfer coefficient from the walls of thin tubes to the air flowing inside them. A comprehensive comparative analysis of the proposed formula with existing empirical data was conducted, confirming its accuracy and applicability.

The findings of this study are crucial for determining the heat transfer coefficient in tubes with internal diameters ranging from 3 to 8 mm. The results are applicable for Reynolds numbers between 150 and 310, Grashof numbers from 110 to 1000, and channel geometric dimension ratios x/d from 20 to 200.

Moreover, the study's outcomes facilitate a robust and precise assessment of the operational efficiency of heat recovery systems constructed from thin tubes. This makes it possible to optimize their design and operation, enhancing energy efficiency and thermal performance across a broad range of practical applications in venti–lation and air conditioning systems. Furthermore, these findings offer critical insights into evaluating the effici–ency of TwinFresh regenerative heat exchangers of various sizes.

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NOMENCLATURE

Nu	Nusselt number
Re	Reynolds number
Pr_{air}	Prandtl number of air
Pr	Prandtl number
Gr	Grashof number
d	diameter of the tube, m
C_p	specific heat capacity of air
G	mass flow rate of air
q_l	specific heat input, given by $q_l = U{\cdot}I / l$
U	voltage applied to the tube
I	current strength
l	length of the tube
$\bar{C}$	average heat loss along the length, defined as: $\bar{C} = \sum C / l$
C	linear specific heat losses
x	distance from the origin along the same axis
T_{air}	air temperature, К
$T_{air\,in}$	air temperature at the entrance to the experimental section
T_{max}	maximum temperature of the tube wall surface
T_{min}	minimum temperature of the tube wall surface
T_{out}	ambient temperature
$A,\ n,\ m,\ k$	coefficients depend on the flow regime

Greek symbols

$\alpha(x)$	local heat transfer coefficient from the tube to the air
λ_{air}	thermal conductivity of air

ЕКСПЕРИМЕНТАЛНО ОДРЕЂИВАЊЕ КОЕФИЦИЈЕНТА ПРЕНОСА ТОПЛОТЕ У ТАНКИМ КАНАЛИМА РЕГЕНЕРАТИВНОГ ИЗМЕЊИВАЧА ТОПЛОТЕ

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Употреба система за рекуперацију топлоте издувног ваздуха је кључна за савремену енергетски ефикасну вентилацију. У приватним кућама, просторна ограничења често ометају централизоване системе, чинећи децентрализовану регенеративну вентилацију ефикасном алтернативом. Ови системи имају мале ћелије еквивалентног пречника, које захтевају рафинисане коефицијенте преноса топлоте за тачне процене ефикасности.

Конструисана је експериментална поставка за проучавање преноса топлоте у цевима пречника 3 мм и 5 мм, што је довело до развоја формуле Нуселтовог броја за прецизно одређивање коефицијента конвективног преноса топлоте. Да би се смањила интеракција са околином, примењена је одговарајућа дебљина топлотне изолације.

Моделирање канала пречника 3 мм и 9 мм открило је дистрибуцију температуре и односе између Нуселт и Грасхоф бројева. Спроведена је компара–тивна анализа формуле Нуселтовог броја која је помогла у процени коефицијената преноса топлоте за цеви унутрашњег пречника од 3 до 8 мм, заједно са конструкцијом дијаграма за инжењерске примене.