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EXPERIMENTAL INVESTIGATION OF HYDRODYNAMIC AND HEAT TRANSFER PROCESSES IN THE HEAT EXCHANGER OF AN AIR COOLING SYSTEM

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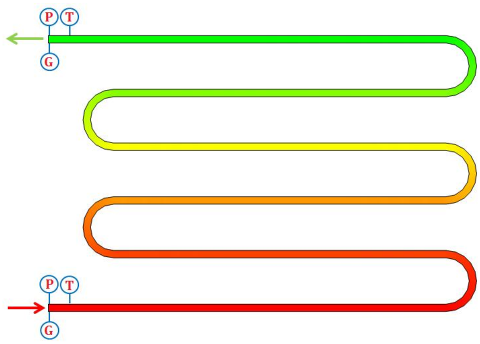
Abstract. This paper presents an experimental investigation of hydrodynamic and heat transfer processes in the heat exchanger of an air cooling system under different air flow velocities and temperatures. When the air velocity varied from 2.5 to 4.0 m/s, the Reynolds number ranged from 4700 to 8300, indicating a fully turbulent flow regime. The total pressure drop increased from 213.3 to 1158.7 Pa, while the required fan power varied between 12.2 and 106.4 W. The heat transfer analysis showed that the Nusselt number increased from 16.2 to 23.6, and the heat transfer coefficient rose from 13.5 to 20.1 $W/(m^2 \cdot K)$. Based on the experimental results, an empirical correlation, $Nu = 0.018 Re^{0,78}$, was proposed with a prediction error not exceeding ±1.6%.

Keywords: air cooling system; heat exchanger; friction resistance; Reynolds number; Nusselt number; heat transfer coefficient; pressure drop; energy efficiency.

Introduction. In recent years, the limited availability of energy resources, the continuous increase in electricity prices, and the challenges associated with climate change have made the improvement of energy efficiency in building cooling systems a highly relevant issue worldwide. In particular, in regions with hot and arid climatic conditions, cooling buildings using conventional compressor-based air conditioning systems requires significant electrical energy consumption and has a negative impact on the environment [1, 2]. Therefore, the development and investigation of alternative, low-energy cooling technologies based on the effective utilization of ambient environmental conditions are of great scientific and practical importance. Air cooling systems (ACS) based on natural and semi-natural cooling methods have attracted increasing attention from researchers in recent years. The main advantages of such systems include the utilization of the thermal potential of outdoor air, reduced electricity consumption, and lower operational costs. In particular, the application of water-cooled heat exchangers enables effective cooling of outdoor air before it is supplied to indoor spaces [3, 4]. However, the actual performance of these systems strongly depends on the hydrodynamic and heat transfer characteristics of the air flow, which necessitates comprehensive experimental and theoretical investigations. Pressure losses and friction resistance of the air flow passing through the heat exchanger are among the key factors determining the overall energy efficiency of the system. Although an increase in air flow velocity enhances heat transfer intensity, it simultaneously leads to a significant rise in pressure losses caused by wall friction and local resistances within the ducts. This, in turn, results in higher fan power requirements and increased energy consumption of the system. Therefore, determining the friction resistance coefficient as a function of air velocity, temperature, and heat exchanger geometric parameters represents an important scientific task. Moreover, the evaluation of air cooling system performance requires accurate determination of the heat transfer coefficient and the Nusselt number. Under practical operating conditions, heat transfer processes become more complex due to the combined influence of flow regime, thermophysical properties of air and the cooling medium, as well as the surface temperature and structure of the heat exchanger. Most theoretical correlations available in the literature are derived for idealized conditions, making it necessary to experimentally verify their applicability to real systems. From this perspective, the present study experimentally investigates hydrodynamic and heat transfer processes in the heat exchanger of an air cooling system installed in a test building under various air flow velocities and temperatures. During the experiments, the Reynolds number, friction resistance coefficient, total pressure drop, required fan power, and heat transfer coefficient were determined. The experimental results were compared with existing theoretical models, and an empirical correlation for the heat transfer coefficient was proposed. The findings of this study provide a scientific basis for selecting optimal air flow velocities, sizing fan power, and improving the energy efficiency of heat exchangers in air cooling systems. Furthermore, the proposed empirical correlations can serve as a practical guideline for the implementation of low-energy cooling technologies in hot climate conditions.

Results and discussion. Experimental investigations to determine the friction resistance coefficient in the heat exchanger of the ACS under various air velocities and temperatures were carried out according to the measurement and control scheme shown in Fig. 1. Based on the experimental studies conducted in the test building, it was determined that when the air supply velocity to the room was 0.5, 1.0, 1.5, and 2.0 m/s, the corresponding air outlet velocities from the heat exchanger were 1.5, 2.0, 2.6, and 3.2 m/s, respectively. To achieve these outlet velocities, the required inlet air velocities to the heat exchanger were experimentally determined to be approximately 2.5, 3.0, 3.5, and 4.0 m/s. The experimental investigations for determining the friction resistance coefficient were carried out within the following ranges of the main parameters: inlet outdoor air velocity of u=2.5−4.0 m/s, inlet outdoor air temperature of 25-40°C, average cooling water temperature of 21°C, and average surface temperature of the heat exchanger of 21°C.



T-temperature; G-volumetric flow rate; P-pressure

Fig. 1. Measurement and control scheme for determining friction resistance in the heat exchanger

To determine the flow regime of air moving inside the tube, the Reynolds number was calculated as follows:

$$Re=\frac{ud}{\nu} \quad (1)$$

where u is the inlet air velocity (m/s), d is the inner diameter of the heat exchanger tube (m), and ν is the kinematic viscosity of air (m^2/s).

The calculation results based on Eq. (1) indicate that the investigated air flow regime is turbulent under all considered operating conditions (Fig. 2a). As the air velocity increases within the range of 2.5-4.0 m/s, the Reynolds number rises almost linearly, varying from 4706 to 8258. This behavior can be attributed to the increasing dominance of inertial forces over viscous forces as the flow velocity increases.

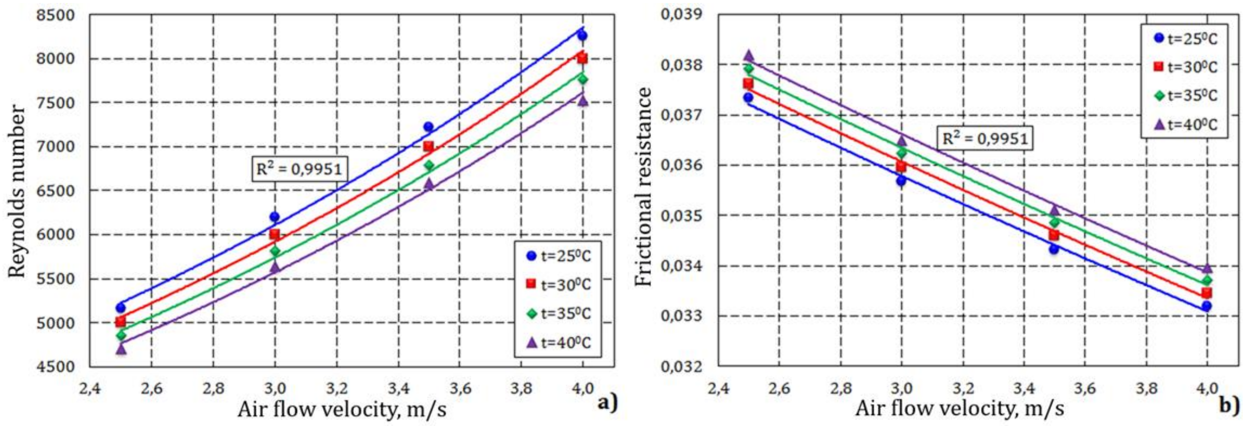


Fig. 2. Dependence of the Reynolds number (a) and the friction resistance coefficient (b) on air flow velocity

In addition, increasing the outdoor air temperature from 25°C to 40°C leads to a decrease in the Reynolds number. This effect is associated with the increase in the kinematic viscosity of air as its temperature rises, which enhances the viscous characteristics of the flow. The high coefficient of determination observed in the graphs ($R^2 \approx 0.995$) confirms the stability and reliability of the calculated results. Figure 2b illustrates the dependence of the friction resistance coefficient on air flow velocity. As can be seen from the figure, the friction resistance coefficient exhibits a decreasing trend with increasing flow velocity. This behavior can be explained by the reduction of relative friction losses in the boundary layer as the Reynolds number increases. At the same time, an increase in temperature causes a slight rise in the friction resistance coefficient, which is attributed to the increase in air viscosity.

The total pressure drop in the heat exchanger tube due to friction and local resistances is determined as follows:

$$\Delta P_{tot}=\Delta P_f+\Delta P_l=\xi\frac{l}{d}\frac{\rho u^2}{2}+\zeta_{tot}\frac{\rho u^2}{2} \qquad (2)$$

where l is the tube length (m) and ρ is the air density (kg/m^3).

The required fan power is calculated using the following expression:

$$N=\frac{G\Delta P}{\eta} \qquad (3)$$

where G is the volumetric air flow rate (m^3/s), ΔP is the pressure drop (Pa), and η is the fan efficiency, which is typically assumed to be $\eta=0.85$.

Under conditions of varying inlet air velocities and temperatures, the total pressure drop and the required fan power in the heat exchanger system were determined. The obtained results are presented in Fig. 3. As shown in Fig. 3a, the total pressure drop in the heat exchanger tube increases significantly with increasing air flow velocity. This behavior is attributed to the intensification of friction losses along the tube walls as the flow velocity rises. At the same time, the effect of increasing inlet air temperature on the total pressure drop was found to be relatively small. When the air flow velocity varied from 2.5 to 4.0 m/s, the total pressure drop in the heat

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exchanger ranged from 213.3 to 1158.7 *Pa*. A high level of agreement between the experimental results and the mathematical model was achieved, with a coefficient of determination of R^2=0.9903.

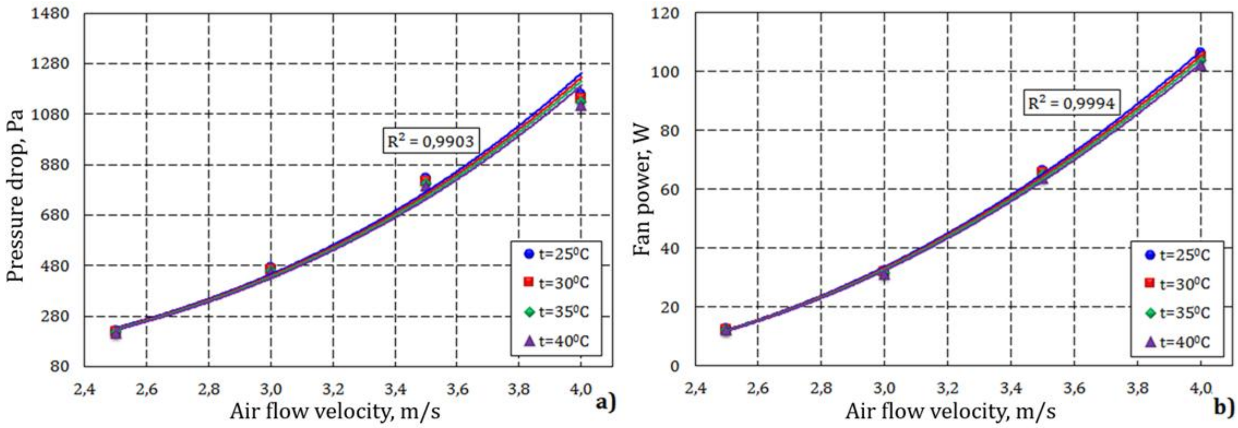

Fig. 3. Dependence of the total pressure drop (a) and fan power (b) on air flow velocity

Fig. 3b shows the dependence of the fan power on air flow velocity and temperature. The results indicate that the required fan power increases sharply with increasing inlet air velocity. In addition, an increase in outdoor air temperature leads to a slight rise in fan power, which is associated with increased pressure losses resulting from changes in the thermophysical properties of air. According to the experimental results, when the air flow velocity varies in the range of 2.5-4.0 m/s, the fan power increases from 12.2 to 106.4 W. The high level of agreement between the experimental and model results is confirmed by a coefficient of determination of R^2=0.9994. Analysis of the experimental and theoretical results indicates that the selected fan with a rated power of 120 W is capable of providing the required air flow rate for the cooling system of the test building.

Experimental investigations of heat transfer processes were carried out according to the scheme shown in Fig. 4 under various inlet air velocities and temperatures at the heat exchanger of the air cooling system. The experiments aimed at determining the heat transfer coefficient were conducted within the following ranges of the main parameters: inlet air velocity of u=2.5−4.0 m/s, inlet air temperature of 25-40°C, average cooling water temperature of 21°C, average surface temperature of the heat exchanger of 21°C, and Reynolds number in the range of 4700-8300. For turbulent air flow inside the heat exchanger tube, the heat transfer coefficient can be evaluated using the correlation proposed by Academician M. A. Mikheev [5]:

$$Nu=0{,}021Re^{0{,}8}Pr^{0{,}43}\left(\frac{Pr_a}{Pr_w}\right)^{0{,}25} \quad (4)$$

For air, this expression can be simplified assuming $Pr_a \approx 0{,}7$, yielding:

$$Nu=0{,}018Re^{0{,}8} \quad (5)$$

where Re is the Reynolds number determined using Eq. (1), with the corresponding values presented in Fig. 2a.

The heat transfer coefficient from the outer surface of the heat exchanger tube to the cooling water in the fountain basin is determined as follows:

$$\alpha=\frac{Nu\lambda}{d_{out}} \quad (6)$$

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where λ is the thermal conductivity of air ($W/(m{\cdot}°C)$) and d_{out} is the outer diameter of the tube (m).

The results obtained using Eqs. (5) and (6) are presented in Fig. 4. As shown in Fig. 4a, the Nusselt number increases with increasing air flow velocity. Specifically, when the inlet air velocity was 2.5 m/s, the Nusselt number varied in the range of 15.6-16.8, with an average value of 16.2. At an air velocity of 3.0 m/s, it ranged from 18.1 to 19.4, yielding an average of 18.7. Further increasing the velocity to 3.5 m/s resulted in Nusselt numbers between 20.4 and 22.0, with an average value of 21.2, while at 4.0 m/s the Nusselt number varied from 22.7 to 24.5, with an average of 23.6. According to the results presented in Fig. 4b, the heat transfer coefficient from the outer surface of the heat exchanger tube increases with increasing air flow velocity. Variations in inlet air temperature were found to have only a minor effect on the heat transfer coefficient. When the inlet air velocity was 2.5 m/s, the heat transfer coefficient ranged from 13.5 to 13.8 $W/(m^2{\cdot}°C)$; at 3.0 m/s, it varied between 15.6 and 16.0 $W/(m^2{\cdot}°C)$; at 3.5 m/s, between 17.6 and 18.1 $W/(m^2{\cdot}°C)$; and at 4.0 m/s, between 19.6 and 20.1 $W/(m^2{\cdot}°C)$. The close agreement between the experimental data and the model predictions is confirmed by a high coefficient of determination of R^2=0.9951.

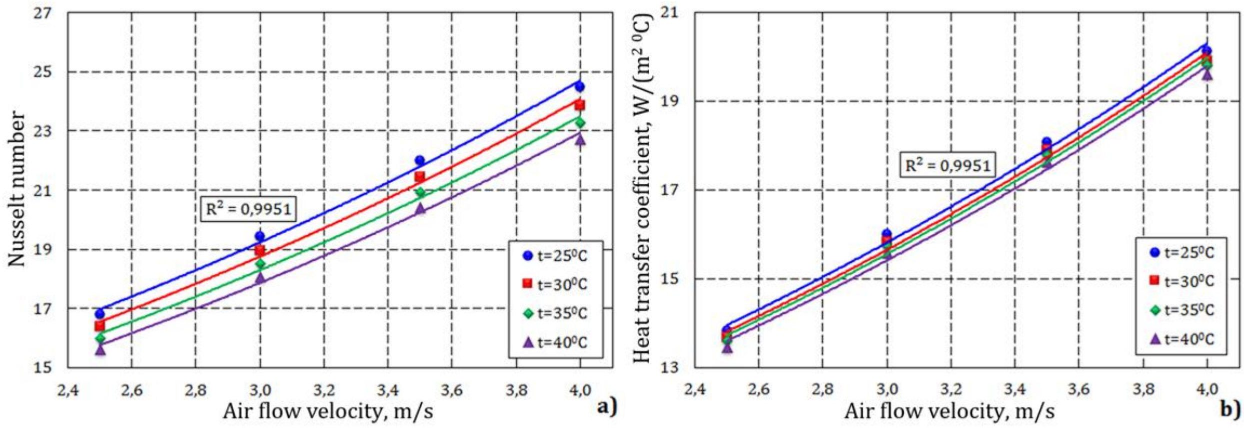

Figure 4. Dependence of the Nusselt number (a) and the heat transfer coefficient (b) on air flow velocity

The results of the outlet air temperature from the heat exchanger tube and the amount of heat transferred from the outer surface of the tube to the cooling water are presented in Fig. 5.

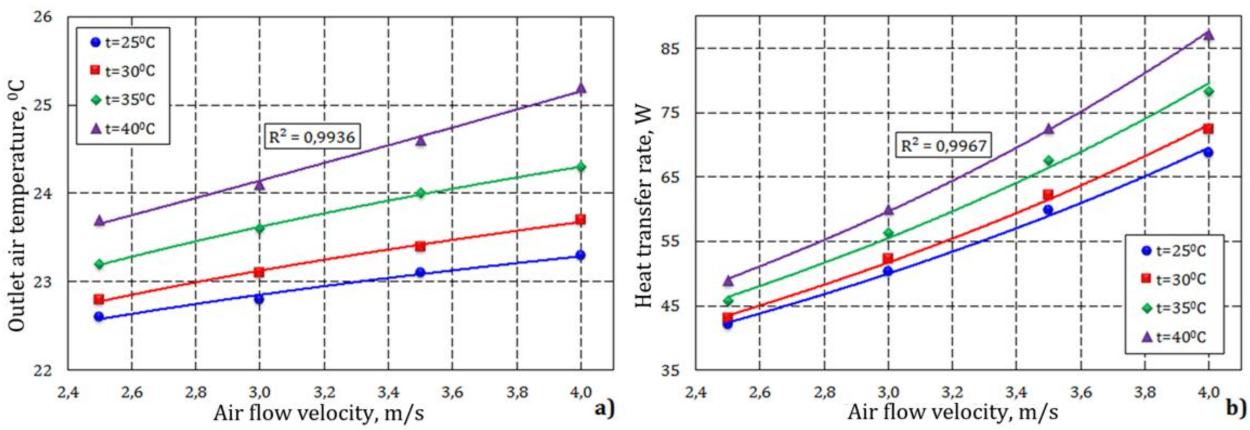

Figure 5. Dependence of the outlet air temperature (a) and the heat transfer rate (b) on air flow velocity

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As shown in Fig. 5a, an increase in both the inlet air velocity and temperature leads to a rise in the outlet air temperature. When the inlet air temperature was 25°C and the air velocity varied from 2.5 to 4.0 m/s, the outlet air temperature ranged from 22.6 to 23.7°C. At an inlet air temperature of 30°C and the same velocity range, the outlet air temperature increased from 22.8 to 24.1°C. For an inlet air temperature of 35°C, the outlet air temperature varied between 23.1 and 24.6°C, while at 40°C it ranged from 23.3 to 25.2°C as the air velocity increased from 2.5 to 4.0 m/s. A high level of agreement between the experimental and model results was achieved, with a coefficient of determination of R^2=0.9936. According to the results presented in Fig. 5b, the amount of heat transferred to the cooling water increases with increasing air flow velocity and inlet air temperature. When the inlet air temperature was 25°C and the air velocity varied from 2.5 to 4.0 m/s, the heat transfer rate ranged from 42.0 to 48.9 W. At inlet air temperatures of 30°C, 35°C, and 40°C, the corresponding heat transfer rates increased to 50.3-60.0 W, 59.8-72.7 W, and 68.8-87.2 W, respectively. The close agreement between the experimental data and the model predictions is confirmed by a coefficient of determination of R^2=0.9967.

The experimental results obtained for the heat transfer coefficient were generalized and reprocessed, resulting in the following empirical correlation:

$$Nu=0{,}018 \cdot Re^{0{,}78} \qquad (7)$$

This empirical correlation is valid within the following ranges of the main parameters: inlet air velocity of 2.5-4.0 m/s, inlet air temperature of 25-40°C, and Reynolds number in the range of 4700-8300. The prediction error does not exceed ±1.6%. Analysis of the experimental results indicates that the proposed air cooling system is capable of maintaining the indoor air temperature within the range of 22.8-25.2°C when the outdoor air temperature varies between 25 and 40°C.

Conclusions. Based on the experimental investigations, when the inlet air velocity to the heat exchanger ranged from 2.5 to 4.0 m/s, the Reynolds number varied between 4706 and 8258, indicating that the air flow was fully turbulent under all considered operating conditions. An increase in outdoor air temperature from 25°C to 40°C resulted in a decrease in the Reynolds number, which can be explained by the increase in the kinematic viscosity of air with temperature.

With increasing air flow velocity in the heat exchanger tube, the total pressure drop increased significantly, ranging from 213.3 to 1158.7 Pa. Under these conditions, the required fan power varied between 12.2 and 106.4 W. Based on the experimental results, it was confirmed that the selected fan with a rated power of 120 W is capable of reliably providing the required air flow rate for the air cooling system.

Analysis of the heat transfer processes showed that the average Nusselt number increased from 16.2 at an inlet air velocity of 2.5 m/s to 18.7 at 3.0 m/s, 21.2 at 3.5 m/s, and 23.6 at 4.0 m/s. Correspondingly, the heat transfer coefficient increased within the ranges of 13.5-13.8, 15.6-16.0, 17.6-18.1, and 19.6-20.1 $W/(m^2 \cdot °C)$. A high level of agreement between the experimental data and theoretical model predictions was achieved, with a coefficient of determination of R^2≈0.995.

Based on the experimental data, an empirical correlation for the heat transfer process, $Nu=0.018Re^{0{,}78}$, was proposed and validated for Reynolds numbers in the range of 4700-8300, with a prediction error not exceeding ±1.6%. The results demonstrate that, under outdoor air temperatures of 25-40°C, the proposed air cooling system is capable of maintaining the indoor

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air temperature within the range of 22.8-25.2℃ and increasing the heat transfer rate to the cooling water up to 42.0-87.2 W.

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