

Effect of airflow velocity on the thermal performance of an inline tube heat exchanger in a biomass-fired fish drying system

Rifqi Dzaki Yenda^{1*}, Eddy Elfiano¹, Sehat Abdi Saragih¹,
Dody Yulianto¹, Sufizar Ahmad²

¹Department of Mechanical Engineering, Universitas Islam Riau,
Pekanbaru, Indonesia

²Faculty of Mechanical and Manufacturing Engineering, University Tun Hussein
Onn Malaysia, 86400 Parit Raja, Batu Pahat, Johor, Malaysia
rifqidzakiyenda@student.uir.ac.id

ABSTRACT

This study evaluates the effect of airflow velocity on the thermal performance of an inline tube heat exchanger integrated into an indirect biomass-fired fish drying system. Experiments used wood biomass as the heat source and three airflow velocities of 0.8, 1.1, and 1.3 m/s. Temperatures were recorded at 5, 10, and 15 min, and the convective heat transfer coefficient, air mass flow rate, logarithmic mean temperature difference (LMTD), heat transfer rate, and calculated thermal efficiency were evaluated using established heat-transfer relations. Increasing airflow velocity from 0.8 to 1.3 m/s increased the average convective heat transfer coefficient from 23.936 to 33.041 W/m²·K, a 38% increase. The highest reported heat transfer rate, 9.86 W, occurred at 1.3 m/s, whereas the highest calculated thermal efficiency, 54.18%, occurred at 0.8 m/s. Under the tested conditions, higher airflow enhanced convective heat transfer and mass flow rate, while lower airflow produced a higher calculated efficiency. The results indicate a trade-off between heat-transfer capacity and calculated thermal efficiency. Verify the reported heat-transfer-rate unit and efficiency basis against the complete energy balance before final publication.

Keywords: Airflow velocity, biomass, fish dryer, heat exchanger, inline tube

INTRODUCTION

Fish drying is widely used to extend shelf life by reducing moisture content and limiting microbial activity. Conventional open-air drying, however, depends heavily on weather conditions and may expose fish to dust, insects, and uncontrolled temperature and humidity. Controlled forced-convection drying systems can reduce these limitations by providing a more stable thermal environment

and more reproducible drying conditions (Jain, 2006; Khan, 2021).

Biomass is a practical renewable heat source for drying applications because agricultural and woody residues can be converted directly into thermal energy. Hybrid and biomass-assisted fish dryers have demonstrated the feasibility of using biomass heat to support controlled drying while reducing dependence on conventional fuels (Hamdani et al.,

2018; Budi et al., 2023). In an indirect configuration, combustion gases are separated from the drying air, which helps prevent direct product contamination.

The heat exchanger is therefore a key component of an indirect biomass dryer because it transfers thermal energy from the combustion side to the drying-air stream. For a tube-bank heat exchanger, airflow velocity influences the Reynolds number, Nusselt number, convective heat transfer coefficient, mass flow rate, and drying-air temperature rise. An inline tube arrangement is attractive for compact drying equipment because of its simple geometry and predictable flow path; nevertheless, its performance depends on tube-bank geometry and operating conditions. Previous studies have investigated forced-convection fish drying, solar-biomass dryers, and biomass-heated drying equipment (Jain, 2006; Hamdani et al., 2018; Khan, 2021; Budi et al., 2023). However, the present manuscript focuses specifically on how airflow velocity changes the coupled thermal parameters of an inline tube heat exchanger integrated into an indirect biomass-fired fish dryer. This focus provides the practical research gap addressed in the present work.

Accordingly, this study evaluates the effect of airflow velocities of 0.8, 1.1, and 1.3 m/s on the convective heat transfer coefficient, air mass flow rate, LMTD, heat transfer rate, and calculated thermal efficiency. The objective is not to establish a universal optimum velocity, but to identify the best-performing condition among the tested velocities for heat-transfer capacity and

calculated efficiency, thereby providing a basis for subsequent optimisation of biomass-powered fish drying systems.

MATERIALS AND METHODS

Experimental Setup

The experimental study was conducted at the Mechanical Engineering Laboratory, Faculty of Engineering, Universitas Islam Riau, Indonesia. The test rig consisted of an indirect biomass-fired fish drying system equipped with an inline tube heat exchanger. The heat exchanger transferred thermal energy from biomass combustion to the drying-air stream so combustion gases did not directly mix with the air entering the drying chamber.

The apparatus consisted of a combustion chamber, an inline tube heat exchanger, an electric blower, a drying chamber, and an exhaust chimney. Wood biomass was used as the heat source. Air supplied by the blower passed through the heat-exchanger flow path, absorbed thermal energy, and subsequently entered the drying chamber. The heat-exchanger geometry was kept unchanged for all tests.

Wood biomass served as the fuel source throughout the experiments. The measuring system consisted of the inline tube heat exchanger, biomass combustion furnace, electric blower, thermocouples, infrared thermometer, anemometer, tachometer, and speed controller (dimmer).

Wood biomass served as the fuel source throughout the experiments. The following measuring instruments were used during data collection:

- Inline tube heat exchanger
- Biomass combustion furnace
- Electric blower
- Thermocouple
- Infrared thermometer
- Anemometer
- Tachometer
- Speed controller (dimmer)

Thermocouples were used to measure temperatures at selected locations in the system, while an infrared thermometer was used to monitor heat-exchanger surface temperature. Air velocity was measured using an anemometer, blower rotational speed using a tachometer, and the speed controller was used to set the required airflow velocity. Instrument model, range, resolution, and measurement uncertainty should be reported in the final manuscript.

The thermocouple was installed to measure the temperatures at several locations along the heat exchanger, while the infrared thermometer was used to monitor the surface temperature of the heat exchanger. Air velocity was measured using an anemometer, whereas blower rotational speed was determined using a tachometer. A speed controller was employed to adjust the airflow velocity during testing.

Experimental Procedure

Before testing, all experimental equipment was inspected to ensure proper operation. The combustion chamber was then filled with wood biomass and ignited until stable combustion was achieved. The blower was adjusted to produce three airflow velocity variations:

- 0.8 m/s
- 1.1 m/s

- 1.3 m/s

For each airflow condition, temperatures were recorded after:

- 5 minutes
- 10 minutes
- 15 minutes

Temperature measurements included:

- combustion chamber temperature,
- heat exchanger surface temperature,
- inlet air temperature,

Each experiment was repeated under identical operating conditions to ensure measurement consistency.

Data Analysis

Experimental data were analyzed using standard forced-convection and energy-balance relations. For an inline tube bank, the maximum velocity between adjacent tubes is determined from the inlet velocity and tube-bank geometry. The exact relation used in this study must be stated together with its geometric assumptions.

Experimental data were analyzed using fundamental heat transfer equations. The maximum airflow velocity inside the inline tube bank was determined as

$$V_{max} = \frac{ST}{SL - D} v$$

where ST is the transverse tube pitch (m), D is the tube outside diameter (m), and V is the approach air velocity (m/s).

The Reynolds number was evaluated from $ReD = \rho V_{max} D / \mu$, where ρ is air density (kg/m^3), V_{max} is the maximum flow velocity (m/s), D is tube diameter (m), and μ is dynamic viscosity ($\text{Pa}\cdot\text{s}$).

Thermophysical properties should be evaluated at a stated reference or film temperature.

The Reynolds number was calculated as

$$Re_D = \frac{\rho V_{max} D}{\mu}$$

We obtained the Nusselt number for the inline tube bank from an empirical tube-bank correlation. In the final manuscript, the complete correlation $Nu_D = C Re_D^m Pr^n$ (including any row correction factor), the values of C , m , and n , and the applicable Reynolds- and Prandtl-number ranges must be stated and referenced to the heat-transfer source used by the authors.

The convective heat transfer coefficient was calculated as $h = Nu_D k / D$, where k is the thermal conductivity of air ($W/m \cdot K$).

The convective heat transfer coefficient was calculated using

$$h = \frac{Nu_{D,N_L} k}{D}$$

The air mass flow rate was determined from $\dot{m} = \rho A V$, where A is the effective airflow cross-sectional area (m^2). Because the reported mass flow rates are relatively high for a laboratory dryer, report the value of A and the density used in this calculation, and verify them.

The air mass flow rate was determined from

$$\dot{m} = \rho_1 V N_T S_T L$$

The logarithmic mean temperature difference (LMTD) was calculated from the measured hot- and cold-side temperature differences. The final manuscript must identify the actual flow configuration (e.g., cross-flow,

parallel flow, or counter-flow) and state whether an LMTD correction factor was required.

The logarithmic mean temperature difference (LMTD) was calculated as

$$\Delta T_{lm} = \frac{(T_s - T_e) - (T_s - T_i)}{\ln(T_s - T_e) - (T_s - T_i)}$$

Evaluate the heat transfer rate consistently from the selected energy balance; for example, $\dot{Q} = \dot{m} c_p (T_{out} - T_{in})$ for the air side, or $\dot{Q} = UA \Delta T_{lm}$ when using an overall heat-transfer coefficient. The manuscript must state which relation generated the reported values.

The heat transfer rate was calculated using

$$\dot{Q} = \dot{m} c_p (T_e - T_i),$$

Define thermal efficiency using an explicit energy boundary. If efficiency is based on biomass energy input, the denominator should include the biomass consumption rate and lower heating value; if you used another effectiveness/efficiency definition, report that equation and all variables. The values reported in this manuscript are therefore referred to as calculated thermal efficiency until the energy boundary is fully documented. Finally, the thermal efficiency of the heat exchanger was determined by All symbols, units, property values, geometric parameters, and correlation validity ranges used in the calculations should be provided immediately after the corresponding equations.

Where:

$$= \frac{\dot{m} c_p (T_e - T_i)}{\dot{m} c_p (T_s - T_i)} \times 100\%$$

These equations were applied to evaluate the influence of airflow

velocity on the thermal performance of the inline tube heat exchanger.

2.5 Research Variables

The independent variable investigated in this study was the airflow velocity supplied by the blower, consisting of three levels:

- 0.8 m/s
- 1.1 m/s
- 1.3 m/s

The dependent variables included:

- A. Convective heat transfer coefficient (h)
- B. Air mass flow rate ($\dot{m}$)
- C. Logarithmic Mean Temperature Difference (LMTD)
- D. Heat transfer rate (Q)
- E. Thermal efficiency (η)

All experiments used the same heat exchanger geometry, identical biomass fuel, and the same test duration to ensure comparable operating conditions.

RESULTS AND DISCUSSION

Effect of Air Velocity on Convective Heat Transfer Coefficient

The convective heat transfer coefficient increased with air velocity. At an air velocity of 0.8 m/s, the heat transfer coefficient ranged from 23.867 to 24.016 W/m²·°C. When the air velocity increased to 1.1 m/s, the coefficient increased to 28.567–28.730 W/m²·°C. The highest values were obtained at 1.3 m/s, ranging from 32.909 to 33.173 W/m²·°C.

Increasing airflow velocity from 0.8 to 1.3 m/s increased the average heat transfer coefficient by approximately 38%. This trend is physically consistent with forced convection because a higher

Reynolds number promotes stronger mixing and a thinner thermal boundary layer around the tubes. The magnitude of the coefficient should nevertheless be interpreted together with the exact tube-bank correlation and geometry reported in the revised Method section.

The results indicate that increasing air velocity from 0.8 m/s to 1.3 m/s increased the average heat transfer coefficient by approximately 38%, demonstrating the strong influence of airflow on heat exchanger performance.

Effect of Air Velocity on Air Mass Flow Rate

The air mass flow rate also increased proportionally with airflow velocity. At 0.8 m/s, the mass flow rate remained relatively constant at 0.552–0.554 kg/s. Increasing the velocity to 1.1 m/s increased the mass flow rate to approximately 0.757–0.760 kg/s, while the highest airflow velocity (1.3 m/s) produced mass flow rates between 0.891 and 0.895 kg/s. The reported heat transfer rate increased with airflow velocity.

The heat transfer rate increased significantly with increasing airflow velocity. At 0.8 m/s, the heat transfer rate ranged from 7.70 to 8.02 W. At 1.1 m/s, the heat transfer rate increased to 8.93–9.58 W. The highest heat transfer rate occurred at 1.3 m/s, reaching 9.28–9.86 W.

3.3 Effect of Air Velocity on Thermal Efficiency

Unlike the heat transfer rate, thermal efficiency decreased as airflow velocity increased.

The decrease in calculated thermal efficiency with increasing airflow

velocity may reflect a trade-off between convective enhancement and the temperature rise achieved per unit mass of air. Increasing velocity raises Reynolds number, Nusselt number, h , and mass flow rate, but also reduces the characteristic residence time and may decrease the air temperature rise. Therefore, the efficiency trend should not be attributed to residence time alone; it should be interpreted using the complete energy balance and the efficiency definition stated in the revised Method section. At 1.1 m/s, the efficiency decreased to 50.30–51.64%. The lowest efficiency occurred at 1.3 m/s, ranging from 47.39% to 48.30%.

This phenomenon can be explained by the air residence time inside the heat exchanger. At lower airflow velocities, air remains in the tube bank longer, allowing greater heat absorption from the tube surface. Conversely, at higher airflow velocities, the residence time becomes shorter, reducing the amount of heat absorbed by each unit mass of air

Overall Thermal Performance

The experimental results clearly indicate that airflow velocity strongly affects the thermal characteristics of the inline tube heat exchanger.

Increasing the airflow velocity from 0.8 m/s to 1.3 m/s produced:

- an increase in the Reynolds number,
- an increase in the Nusselt number,
- higher convective heat transfer coefficients,
- higher air mass flow rates,
- Among the tested conditions, 1.3 m/s produced the highest

reported heat transfer coefficient ($33.173 \text{ W/m}^2\cdot\text{K}$) and heat transfer rate (9.86 W), whereas 0.8 m/s produced the highest calculated thermal efficiency (54.18%). These conditions should not be described as universal optima because only three airflow velocities were investigated.

The results therefore indicate a trade-off between convective heat-transfer enhancement and calculated thermal efficiency within the investigated range. Higher airflow velocity favours the convective coefficient and mass throughput, whereas lower airflow velocity yields the highest reported efficiency. Future optimisation should include a wider airflow range, pressure-drop or blower-power measurements, and drying-performance indicators such as moisture ratio, drying rate, and specific energy consumption.

Therefore, the optimum operating condition depends on the desired application. If the objective is to maximise heat transfer capacity, an airflow velocity of 1.3 m/s is preferable because it produced the highest heat transfer coefficient ($33.173 \text{ W/m}^2 \text{ }^\circ\text{C}$) and heat transfer rate (9.86 W). Conversely, if the objective is to maximise thermal efficiency, the airflow velocity of 0.8 m/s is more suitable because it achieved the highest efficiency (54.18%).

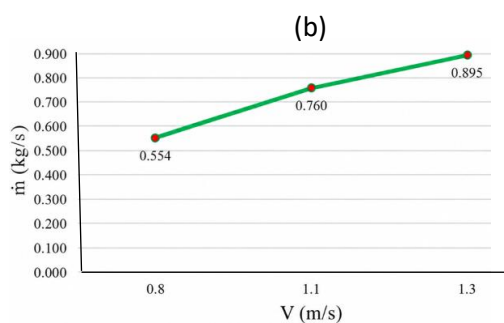
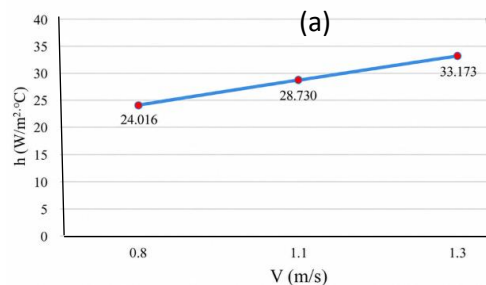
These findings indicate a trade-off between heat transfer enhancement and thermal efficiency in biomass-fired inline tube heat exchangers. Consequently, selecting the appropriate airflow velocity should consider the specific

operational requirements of the fish drying system

Table 1 Heat Transfer Calculation Results at 5 Minutes for Air Velocities of 0.8, 1.1, and 1.3 m/s.

V (m/s)	V_{max} (m/s)	Re_D	Nu_D	h (W/m ² °C)
0,8	1,6	1292,3	21,38	24,016
1,1	2,2	1907,5	26,50	28,730
1,3	2,6	2354,4	31,27	33,173

$\dot{m}$ (kg/s)	ΔT_{lm} (°C)	$\dot{Q}$ (W)	η (%)
0,554	174	$7,70 \times 10^4$	54,03
0,760	164,3	$8,93 \times 10^4$	50,30
0,895	158,1	$9,28 \times 10^4$	47,39



(c)

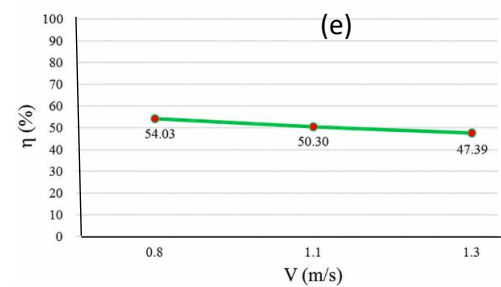
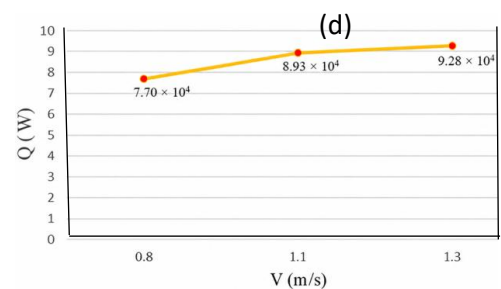
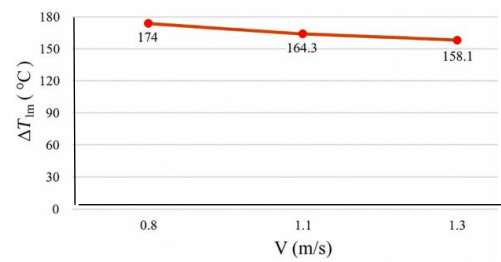


Figure 1 (a) V vs h , (b) V vs $\dot{m}$, (c) V vs ΔT , (d) V vs Q , (e) V vs η to confirms a monotonic increase in the convective heat transfer coefficient with airflow velocity: the average value increased from 23.936 W/m²·K at 0.8 m/s to 28.648 W/m²·K at 1.1 m/s and 33.041 W/m²·K at 1.3 m/s.

The trend is consistent with the increase in Reynolds and Nusselt numbers expected in forced convection over a tube bank. Within the tested range, 1.3 m/s, it therefore provided the strongest convective heat-transfer response. This conclusion applies only to the investigated velocities and should be interpreted after the tube-bank correlation and geometric parameters are fully reported.

Figure 1 shows the effect of air velocity on the convective heat transfer coefficient of the inline tube heat exchanger. The convective heat transfer coefficient increased as the airflow velocity increased from 0.8 m/s to 1.3 m/s. The average heat transfer coefficient increased from 23.936 W/m²·°C at 0.8 m/s to 28.648 W/m²·°C at 1.1 m/s, and reached 33.041 W/m²·°C at 1.3 m/s.

Overall, Figure 1 shows that airflow velocity is a dominant factor influencing the convective heat transfer performance of the inline tube heat exchanger. The highest average heat transfer coefficient of 33.041 W/m²·K was achieved at an airflow velocity of 1.3 m/s, indicating the best convective heat transfer performance among the tested operating conditions

Figure 2 shows that the reported air mass flow rate increased with airflow velocity, from an average of 0.553 kg/s at 0.8 m/s to 0.759 kg/s at 1.1 m/s and 0.893 kg/s at 1.3 m/s. Because $\dot{m} = \rho AV$, the revised manuscript should provide the effective flow area and air density used to verify these values. Analysis of Figure 2. Effect of Air Velocity on the Air Mass Flow Rate

Figure 2 illustrates the relationship between airflow velocity and the air mass flow rate through the inline tube heat exchanger. The results indicate that the air mass flow rate increased as airflow velocity increased. At an airflow velocity of 0.8 m/s, the average mass flow rate was 0.553 kg/s. Increasing the airflow velocity to 1.1 m/s raised the average mass flow rate to 0.759 kg/s, while the highest average value of 0.893 kg/s was obtained at 1.3 m/s.

Figure 3 indicates that LMTD decreased as airflow velocity increased. This suggests that the terminal temperature differences changed with airflow rate; however, the physical interpretation requires the actual hot- and cold-side temperature data and the correct flow configuration/correction factor used in the LMTD calculation. Analysis of Figure 3. Effect of Air Velocity on the Logarithmic Mean Temperature Difference (LMTD)

Figure 3 illustrates the effect of airflow velocity on the logarithmic mean temperature difference (LMTD) of the inline tube heat exchanger. The results indicate that the LMTD tends to decrease as the airflow velocity increases from 0.8 m/s to 1.3 m/s. This trend suggests that the temperature difference between the hot and cold fluids becomes smaller at higher airflow velocities.

Figure 4 shows an increase in the reported heat transfer rate from an average of approximately 7.86 W at 0.8 m/s to 9.25 W at 1.1 m/s and 9.57 W at 1.3 m/s. This trend is consistent with increasing mass throughput, but the absolute values and units must be rechecked against the raw temperature data and energy balance before publication. Analysis of Figure 4. Effect of Air Velocity on the Heat Transfer Rate

Figure 4 presents the effect of airflow velocity on the heat transfer rate of the inline tube heat exchanger. The experimental results show that the heat transfer rate increased as the airflow velocity was raised from 0.8 m/s to 1.3 m/s. The average heat transfer rate increased from approximately 7.86 W at 0.8 m/s to

9.25 W at 1.1 m/s, reaching a maximum of 9.57 W at 1.3 m/s.

Figure 5 shows that the calculated thermal efficiency decreased from an average of 54.11% at 0.8 m/s to 50.97% at 1.1 m/s and 47.85% at 1.3 m/s. The revised manuscript should interpret this result using the explicitly defined efficiency equation rather than residence time alone. Analysis of Figure 5. Effect of Air Velocity on Thermal Efficiency

CALCULATION EQUATIONS AND REQUIRED REPORTING

1. Maximum velocity through an inline tube bank: state the exact equation used and define ST, D, and V.

EQUATIONS

2. Reynolds number: $Re_D = \rho V_{max} D / \mu$.

$$V_{max} = \frac{ST}{SL - D} v$$

3. Nusselt number: report the complete inline tube-bank correlation, constants, validity range, and source.

$$Re_D = \frac{\rho V_{max} D}{\mu}$$

3. Nusselt number(Nu_D)

4. Convective heat transfer coefficient: $h = Nu_D k/D$.

$$Nu_D = C Re_D^m Pr^n Pr/Pr_s)^{0.25}$$

5. Heat-transfer surface area: $As = N\pi DL$, where N is the number of tubes, and L is the tube length.

$$h = \frac{Nu_{D,N_L} k}{D}$$

6. Air mass flow rate: $\dot{m} = \rho AV$.

$$As = N\pi DL, (m^2)$$

7. Temperature difference: report all terminal temperature differences used for LMTD.

$$\dot{m} = \rho_1 V N_T S_T L$$

8. Heat transfer rate: state and verify the selected energy-balance equation and unit.

$$\Delta T_{ln} = \frac{(T_s - T_e) - (T_s - T_i)}{\ln(T_s - T_e) - (T_s - T_i)}$$

9. Thermal efficiency: state the exact numerator, denominator, system boundary, and energy-input basis.

$$\dot{Q} = \dot{m} C_p (T_e - T_i), W$$

9. Heat Transfer Efficiency

$$\eta = \frac{\dot{Q}_{aktual}}{\dot{Q}_{max}} \times 100\% \\ = \frac{\dot{m} C_p (T_e - T_i)}{\dot{m} C_p (T_s - T_i)} \times$$

CONCLUSION

This study evaluated the effect of airflow velocity on the thermal performance of an inline-tube heat exchanger integrated into a biomass-fired fish-drying system. Increasing the airflow velocity from 0.8 to 1.3 m/s increased the Reynolds number, air mass-flow rate, Nusselt number, convective heat-transfer coefficient, and heat-transfer rate. Among the

tested conditions, an airflow velocity of 1.3 m/s produced the highest convective heat-transfer coefficient (approximately 33.17 W/m²·K) and the highest calculated heat-transfer rate (9.86 W). These results indicate that higher airflow velocity promotes stronger forced convection and improves heat transfer around the tube bank.

Conversely, the highest calculated thermal efficiency (54.18%) occurred at an airflow velocity of 0.8 m/s. The reduction in efficiency at higher velocities indicates a trade-off between heat-transfer capacity and the temperature rise of the flowing air. Lower airflow velocity provides a longer interaction time between the air and the heated tube surfaces, whereas higher velocity increases the convective heat-transfer coefficient and total heat-transfer rate. Therefore, select the airflow velocity based on the required drying temperature, heat-transfer capacity, and energy efficiency.

Overall, the inline-tube heat exchanger demonstrated potential for transferring heat from biomass combustion to the drying air without direct contact between the combustion gases and the dried product. Nevertheless, the heat-transfer-rate unit and thermal-efficiency calculation should be verified using the heat-exchanger geometry, measured air mass-flow rate, temperature data, biomass consumption, and fuel calorific value. Further studies should include uncertainty analysis, repeated experiments, pressure-drop measurements, and actual fish-drying performance—including drying time, moisture reduction, specific energy

consumption, and product quality—to provide a more comprehensive assessment of the system.

REFERENCES

- Afrouzi, H. N., Hassan, A., Wimalaratna, Y. P., et al. (2021). Sizing and economic analysis of stand-alone hybrid photovoltaic-wind system for rural electrification: A case study Lundu, Sarawak. *Cleaner Engineering and Technology*, 4, 100191.
<https://doi.org/10.1016/j.clet.2021.100191>
- Babatunde, O. M., Munda, J. L., & Hamam, Y. (2019). Selection of a hybrid renewable energy system for a low-income household. *Sustainability*, 11(16), 4282.
<https://doi.org/10.3390/su11164282>
- Bezrukikh, P. P., Bezrukikh, P. P. Jr., & Gribkov, S. V. (2014). Wind energy: Reference-methodological publication (P. P. Bezrukikh, Ed.). Intekhenergo-Izdat: Teploenergetik.
- Bhutta, M. M. A., Hayat, N., Farooq, A. U., Ali, Z., Jamil, S. R., & Hussain, Z. (2020). Vertical axis wind turbine – A review of various configurations and design techniques. *Renewable and Sustainable Energy Reviews*, 116, 109404.
<https://doi.org/10.1016/j.rser.2021.12.004>
- Faradanesh, B., & Schuff, A. (2003). Dynamic studies of the NYS transmission system with the Marcy CSC in the UPFC and IPFC configurations. *IEEE*

- Conference on Transmission and Distribution, 3, 1175–1179.
- Frimpong, S. O., Millham, R. C., & Agbehadji, I. E. (2021). A comprehensive review of nature-inspired search techniques used in estimating optimal configuration size, cost, and reliability of a mini-grid HRES: A systemic review. *Lecture Notes in Computer Science*, 12957, 492–507.
- Howell, R., Qin, N., Edwards, J., & Durrani, N. (2010). Wind tunnel and numerical study of a small vertical axis wind turbine. *Renewable Energy*, 35(2), 412–422.
<https://doi.org/10.1016/j.renene.2009.07.025>
- Katkov, V. V. (2024). Vertical-axis wind turbine (Patent RU 230843). Russian Federation Patent Office.
- Krishnan, A., Al-Obaidi, A. S. M., & Hao, L. C. (2023). A comprehensive review of innovative wind turbine airfoil and blade designs: Toward enhanced efficiency and sustainability. *Sustainable Energy Technologies and Assessments*, 60, 103511.
<https://doi.org/10.1016/j.seta.2023.103511>
- Kumar, R., Raahemifar, K., & Fung, A. S. (2018). A critical review of vertical axis wind turbines for urban applications. *Renewable and Sustainable Energy Reviews*, 89, 281–291.
<https://doi.org/10.1016/j.rser.2018.03.033>
- Kvitko, A. V., & Azaryan, A. A. (2022). Design features of wind power plants. *Kuban State Agrarian University named after I. T. Trubilin*.
- Luis Silva-Llanca, & Inostroza-Lagos, S. (2021). Optimum power generation assessment in an H-Darrieus vertical axis wind turbine via exergy destruction minimization. *Energy Conversion and Management*, 243, 114312.
<https://doi.org/10.1016/j.enconman.2021.114312>
- Saryyev, K. A., Nazarov, S. B., & Matyakubov, A. A. (2022). Scientific and technical basis for the implementation of combined technologies using renewable energy sources. *Syktyvkar University Bulletin. Series 2: Biology. Geology. Chemistry. Ecology*, 1(21), 78–86.
<https://doi.org/10.34130/2233-1277-2021-4-78>
- Sharma, K. K., Gupta, A., Kumar, R., et al. (2021). Economic evaluation of a hybrid renewable energy system (HRES) using hybrid optimization model for electric renewable (HOMER) software: A case study of rural India. *International Journal of Low Carbon Technologies*, 16(3), 814–821.
<https://doi.org/10.1093/ijlct/ctab012>
- Smirnov, A. V., & Kozlov, D. I. (2020). Advantages and disadvantages of vertical-axis wind turbines. *Renewable Energy*, 3(45), 34–41.
- Solomin, E. V. (2011). Iterative optimization of parameters and operation modes of vertical-axis wind turbines. *Bulletin of South Ural State University. Series*

- "Power Engineering", 15(232), 73.
- Solomin, E. V., Kirpichnikova, I. M., Amerkhanov, R. A., Korbatov, D. V., Lutavac, M., & Maryanov, A. S. (2018). Application of a wind-hydrogen uninterruptible power supply complex in various climatic conditions. *Alternative Energy and Ecology*, 30–54.
- Saburova, E. A., & Tsytzelskaya, V. A. (2019). Wind energy, working principles of wind turbines, and development prospects. In *Proceedings of the XXI International Scientific-Practical Conference* (pp. 460–465). Tyumen Industrial University.
- Sun, M., Peng, L., Lei, H., Zhang, J., Zhang, Z., Chen, Q., Zhang, G., & Li, J. (2025). A novel small-scale H-type Darrieus vertical axis wind turbine manufactured of carbon fiber reinforced 2.2. Experimental procedures