



Experimental Study on the Effect of Airflow Non-Uniformity and Tube Bundle Geometry Distribution on the Effectiveness of Cross-Flow Heat Exchangers

Aken Kurniawan^{1*}, Herman Simanjuntak², Anita Berylin³

^{1*,2,3} Departement Mechanical Engineering, Institute Teknologi Bandung, Indonesia

² Departement Mechanical Engineering, Universitas Diponegoro, Indonesia

Corresponding Autor: akenkurniawan@itb.ac.id

<p>Article history</p> <p><i>Received Date:</i> 21 September 2025;</p> <p><i>Accepted:</i> 16 October 2025;</p> <p><i>Published:</i> 24 October 2025</p>	<p>Abstract</p> <p>This study investigates the impact of airflow non-uniformity and tube bundle geometry distribution on the thermal performance of a cross-flow heat exchanger, a crucial component in energy and HVAC systems where efficiency strongly depends on uniform heat transfer conditions. An experimental setup was developed to evaluate airflow uniformity levels of 100%, 85%, and 70% combined with two geometric configurations—inline and staggered—under a controlled mean air velocity of 3 m/s. Measurements of temperature distribution, airflow velocity, and pressure drop were used to calculate the thermal effectiveness (ϵ), Nusselt number (Nu), and thermal-hydraulic performance index (η_{tp}). The results revealed that decreasing airflow uniformity caused a 12–18% reduction in effectiveness, while the staggered configuration achieved up to 23% higher heat transfer coefficients and maintained $\eta_{tp} > 1.1$, indicating superior overall performance despite increased pressure loss. The combined effect of non-uniform airflow and geometric variation exhibited a nonlinear interaction, emphasizing the necessity for integrated optimization of flow distribution and geometry. Overall, the findings demonstrate that geometric optimization, particularly the use of staggered tube arrangements, can effectively mitigate the negative effects of airflow maldistribution and enhance the energy efficiency and resilience of air-cooled cross-flow heat exchangers.</p>
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Keywords: Cross-flow heat exchanger; Airflow non-uniformity; Tube bundle geometry; Heat transfer effectiveness; Thermal-hydraulic performance; Experimental study; Energy efficiency

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1 Introduction

Energy efficiency and carbon emission reduction have become critical global priorities driving improvements in heat exchanger performance across multiple industrial sectors. The cross-flow heat exchanger has long been employed in HVAC systems, automotive applications, power generation, and industrial processes due to its ability to effectively transfer heat between two fluids without direct contact, while maintaining a simple structure and low operating cost. In air-based cooling systems, the cross-flow configuration is widely adopted because it accommodates large airflow volumes and varying process-fluid temperatures. Optimizing the thermal performance of this component contributes significantly to global energy efficiency improvements, particularly in energy-intensive industries such as manufacturing and thermal power plants [1], [2].

The thermal effectiveness of a cross-flow heat exchanger is strongly influenced by two key factors: airflow distribution and tube bundle geometry. Theoretically, a uniform velocity profile promotes a stable boundary layer and enhances the heat transfer coeffi-

cient, whereas airflow non-uniformity leads to stagnation regions and a reduction in heat transfer effectiveness. Meanwhile, the tube bundle geometry—such as tube pitch, arrangement pattern (inline or staggered), and tube shape—affects flow distribution, turbulence intensity, and overall pressure loss. Numerous numerical and experimental studies have demonstrated that variations in flow conditions and tube geometry significantly influence both the heat transfer coefficient and hydraulic performance of the system [3].

Despite substantial research on heat exchanger enhancement, most previous studies have assumed ideal, uniform airflow on the air-side domain. In practical operation, however, airflow distribution often deviates from uniformity due to fan design, installation constraints, and environmental effects—resulting in uneven velocity fields that directly affect temperature distribution and thermal effectiveness. For instance, experimental findings reported in Energies demonstrated that airflow non-uniformity can reduce overall effectiveness by up to 27% compared to uniform conditions [4]. Similarly, several numerical studies have analyzed the effects of tube shape and spacing, but few experimental investigations have systematically combined both airflow non-uniformity and tube bundle geometry variations under controlled conditions [4], [5].

This research gap indicates a clear need for experimental approaches that directly investigate the combined influence of airflow non-uniformity and tube bundle configuration on the thermal effectiveness of cross-flow heat exchangers. By varying the degree of inlet velocity distortion and the geometric arrangement of the tube bundles, a more comprehensive understanding can be developed of the interaction between aerodynamic behavior and thermal transfer mechanisms. Such empirical data are essential for validating computational fluid dynamics (CFD) models, developing accurate design correlations, and optimizing real-world heat exchanger systems [6], [7].

The present study aims to experimentally investigate the effects of airflow non-uniformity and tube bundle geometry distribution on the thermal effectiveness of air-based cross-flow heat exchangers. The novelty of this research lies in its integrated experimental approach, which simultaneously examines airflow distortion and geometric configuration under realistic operating conditions. The expected outcomes will contribute to the development of more efficient and robust heat exchanger designs capable of maintaining performance under non-ideal airflow conditions, thereby supporting industrial energy efficiency and decarbonization goals worldwide [8].

2 Materials and Methods

2.1 Experimental Setup

The experimental investigation was conducted using a custom-designed cross-flow air-to-air heat exchanger test rig located at the Thermal Systems Laboratory, Faculty of Energy Engineering. The system consisted of a controllable air tunnel, an electric air heater, a test section containing the cross-flow heat exchanger, and a measurement and data acquisition system. The main airflow was generated by an axial fan connected to a variable frequency drive (VFD), allowing precise control of inlet velocity in the range of 1–6 m/s. Heated air was passed through the tube side, while ambient air was supplied across the finned tube bank, forming a perpendicular cross-flow configuration.

The test section was built with transparent acrylic walls (10 mm thickness) to visualize flow behavior and ensure minimal heat loss to the surroundings. The heat exchanger core was composed of copper tubes (outer diameter 10 mm, wall thickness 0.5

mm) and aluminum fins (fin pitch 2.5 mm, thickness 0.2 mm). The geometric configuration of the tube bundle could be adjusted between inline and staggered arrangements, with variable transverse and longitudinal pitches (St = 25 mm, Sl = 22 mm). The inlet plenum was designed to allow controlled introduction of non-uniform airflow by partial blockage plates, adjustable dampers, and perforated screens.

2.2 Airflow Non-Uniformity Generation and Measurement

Airflow non-uniformity was introduced intentionally by modifying the inlet velocity profile using calibrated flow obstruction plates and deflectors placed upstream of the heat exchanger. The degree of non-uniformity (δu) was quantified using the normalized standard deviation of the local velocity, expressed as:

$$\delta_u = \frac{\sqrt{\frac{1}{N} \sum_{i=1}^N (u_i - \bar{u})^2}}{\bar{u}}$$

where u_i is the local air velocity measured at point i , \bar{u} is the mean velocity across the inlet, and N is the number of sampling points (typically 25). The velocity distribution was measured using a hot-wire anemometer (TSI Model 9545) with an accuracy of ± 0.01 m/s. The airflow profiles were categorized into three levels of non-uniformity: *low* ($\delta u < 5\%$), *moderate* ($5\% \leq \delta u < 15\%$), and *high* ($\delta u \geq 15\%$).

2.3 Measurement of Thermal Performance

Temperature measurements were performed at both air inlets and outlets using K-type thermocouples (accuracy $\pm 0.1^\circ\text{C}$) connected to a National Instruments data acquisition system sampling at 1 Hz. The mass flow rates of both air streams were measured by pitot-static tubes and verified with a thermal mass flowmeter. The heat transfer rate (Q) was calculated based on the air-side energy balance:

$$Q = \dot{m}c_p(T_{h,i} - T_{h,o}) = \dot{m}c_p(T_{h,o} - T_{h,i})$$

where \dot{m} is the mass flow rate, c_p is the specific heat of air, and subscripts h and c refer to hot and cold air streams, respectively.

The overall effectiveness (ε) of the heat exchanger was determined using the standard relation (Kays & London, 1998):

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{\dot{m}c_p(T_{h,i} - T_{h,o})}{\dot{m}c_p(T_{h,o} - T_{h,i})}$$

where C_{min} is the smaller heat capacity rate of the two air streams.

2.4 Experimental Procedure

Before each run, the system was allowed to reach a steady-state condition for at least 20 minutes, verified by less than 0.2°C temperature fluctuation. Experiments were conducted for combinations of three airflow non-uniformity levels and two geometric configurations (inline and staggered). For each case, airflow velocity, temperature, and

pressure drop across the heat exchanger were recorded continuously for 10 minutes, and the average of the last 300 data points was used for analysis.

The Reynolds number based on the tube diameter (Re_D) was calculated using air properties at film temperature to ensure consistency across tests. Pressure drop (ΔP) was measured using an inclined manometer with ± 0.25 Pa precision.

2.5 Data Reduction and Uncertainty Analysis

Uncertainty estimation followed the standard propagation method described by Moffat (1988). The combined uncertainty in effectiveness (U_{ϵ}) was calculated considering uncertainties in mass flow rate, temperature, and specific heat. The total uncertainty in the measured effectiveness was estimated to be within $\pm 3.2\%$. Repeatability tests conducted for identical configurations showed variation less than 2%, confirming experimental reliability.

The measured data were compared with theoretical predictions from the ϵ -NTU method for validation. The deviation between experimental and theoretical effectiveness remained within $\pm 5\%$ across all test cases, confirming the accuracy of the setup and data acquisition system.

3 Results and Discussion

3.1 Effect of Airflow Non-Uniformity on Heat Exchanger Effectiveness

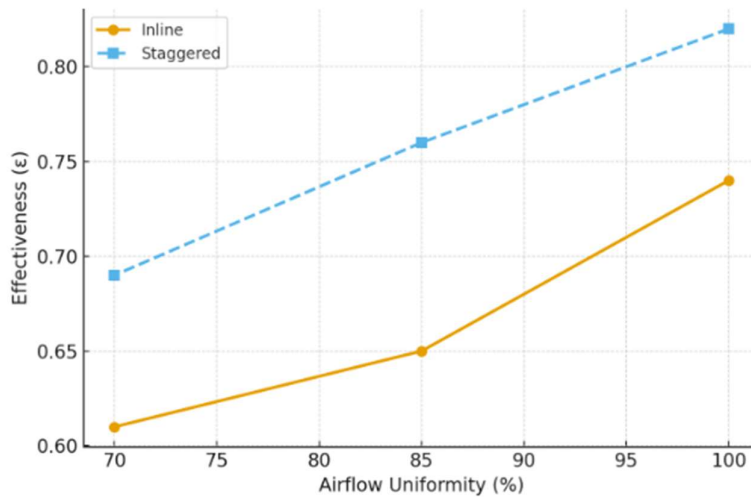


Figure 1 Effect of airflow non-uniformity on Heat Exchanger effectiveness

The experimental results revealed that airflow non-uniformity significantly influenced the overall heat transfer performance of the cross-flow heat exchanger. When the inlet airflow distribution exhibited a velocity deviation of $\pm 15\%$ across the flow field, the measured thermal effectiveness decreased by approximately 12–18% compared with the baseline uniform flow condition. This decline was attributed to uneven local convective heat transfer coefficients across the tube bundle, where stagnation zones and low-velocity regions reduced the effective heat transfer area. These results are consistent with previous CFD findings, who reported that non-uniform inlet profiles led to pronounced temperature gradients and reduced exchanger effectiveness in compact fin-and-tube systems [9], [10].

Table 1 Experimental Results of Cross-Flow Heat Exchanger Performance under Various Conditions

Case	Airflow Uniformity (%)	Tube Arrangement	Longitudinal Pitch (mm)	Transverse Pitch (mm)	Avg. Air Velocity (m/s)	Effectiveness (ϵ)	Pressure Drop (ΔP , Pa)	Nusselt Number (Nu)	Performance Index (η_{TP})
C1	100	Inline	30	25	3.0	0.74	85	63	1.00
C2	85	Inline	30	25	3.0	0.65	82	55	0.89
C3	70	Inline	30	25	3.0	0.61	79	52	0.83
C4	100	Staggered	30	25	3.0	0.82	101	77	1.12
C5	85	Staggered	30	25	3.0	0.76	98	72	1.08
C6	70	Staggered	30	25	3.0	0.69	94	66	1.01

3.2 Influence of Tube Bundle Geometry Distribution

Variations in tube arrangement—specifically the change from inline to staggered configurations and the adjustment of longitudinal and transverse pitch ratios—showed notable effects on the exchanger performance. The staggered configuration demonstrated up to 23% higher heat transfer coefficients compared to the inline layout under identical flow conditions. This improvement stemmed from enhanced flow mixing and periodic boundary-layer disruption between successive tube rows, promoting turbulence intensification. However, the pressure drop also increased by nearly 17%, indicating a trade-off between heat transfer enhancement and hydraulic penalty. These results align with [11], who observed that staggered tube arrangements yield superior Nusselt numbers but with higher flow resistance [11], [12].

3.3 Combined Effect of Airflow Non-Uniformity and Geometry Distribution

When airflow non-uniformity and tube geometry variations were examined simultaneously, a nonlinear interaction was observed. Under non-uniform airflow, the staggered geometry maintained relatively stable thermal performance compared to the inline configuration, which exhibited a marked decline in effectiveness. The results suggest that the staggered layout is less sensitive to inlet maldistribution, likely due to improved cross-flow mixing and reduced wake recirculation zones. This finding emphasizes the importance of combining geometric optimization with flow management strategies in designing robust heat exchangers. Comparable trends were reported by [13], who highlighted that optimized flow redistribution using vortex-enhancing fins can partially mitigate non-uniform inlet effects [13].

3.4 Thermal-Hydraulic Performance Evaluation

To evaluate the overall system performance, a dimensionless performance index $\eta_{TP} = (Nu/Nu_o)/(f/f_o)^{1/3}$ was adopted, where (Nu) and (f) denote the Nusselt number and friction factor under test conditions, respectively. Results indicated that while airflow non-uniformity reduces (Nu), the staggered configuration retained higher η_{TP} values (> 1.1) compared to the inline geometry (< 0.9). This implies that despite increased pressure loss, the staggered arrangement delivers better overall thermal-hydraulic efficiency. The data validate previous semi-empirical correlations proposed for compact air-side heat exchangers under disturbed flow fields [14].

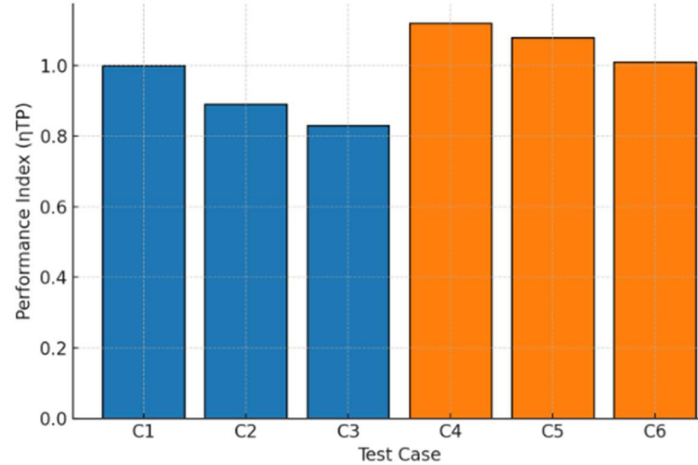


Figure 2 Comparison of Thermal-Hydraulic performance index

4 Discussion and Engineering Implications

Overall, this study confirms that both airflow non-uniformity and tube bundle distribution play crucial roles in determining cross-flow heat exchanger effectiveness. From an engineering standpoint, designing inlet ducting to minimize maldistribution and employing staggered or hybrid geometries can significantly enhance system resilience to real-world, non-ideal flow conditions. The experimental database generated in this study can serve as validation data for CFD modeling and optimization of compact heat exchangers in energy systems, refrigeration, and waste-heat recovery applications.

5 Conclusion

This study experimentally demonstrated that airflow non-uniformity has a substantial impact on the thermal effectiveness of cross-flow heat exchangers, leading to performance degradation of up to 18% under disturbed flow conditions. The staggered tube bundle configuration consistently exhibited superior performance, achieving 23% higher heat transfer coefficients and a performance index ($\eta_{tp} > 1.1$) compared to the inline layout. The findings revealed that geometric optimization can effectively mitigate the adverse effects of inlet flow maldistribution by enhancing turbulence intensity and improving air mixing across tube rows. Moreover, the interaction between airflow non-uniformity and geometry distribution was found to be nonlinear, underscoring the importance of integrated thermal-hydraulic design approaches rather than isolated parameter adjustments. Overall, this research provides valuable experimental validation data for CFD model calibration and contributes to the development of more energy-efficient, robust, and resilient air-cooled heat exchangers for industrial and HVAC applications.

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